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FINAL REPORT

SEALS DESIGN GUIDE

STUDY OF DYNAMIC AND STATIC SEALS FOR LIQUID ROCKET ENGINES

for Period July 1968 to September 1969

NASA Contract No. NAS 7-434

Phase III

Prepared for

Chief, Liquid Propulsion Technology, RPL
National Aeronautics and Space Administration
Washington, D.C. 20546

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FINAL REPORT

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January 1970

For Period July 1968 to September 1969

TO:	Chief, Liquid Propulsion Technology, RPL Office of Advanced Research and Technology NASA Headquarters Washington, D. C. 20546
NASA CONTRACT NO.:	NAS 7-434 Phase III
PREPARED BY:	Research and Development Center General Electric Company Schenectady, New York 12305
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FOREWORD

This SEALS DESIGN GUIDE was prepared in fulfillment of the requirements for Contract NAS 7-434, Phase III. Contributors for the writing and editing of the GUIDE are as follows:

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The content of the GUIDE draws heavily on the previous work done under NASA Contracts NAS 7-434 and NAS 7-102. A great debt is owed to the many contributors to this work. There are too many to name here but throughout the GUIDE their names appear along with their contributions.

Special thanks is given to the NASA Technical Manager, Mr. R. S. Weiner, Liquid Propulsion Section, Jet Propulsion Laboratory.

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INTRODUCTION

A. Summary of Design Guide

Purpose

The objective of the work under NAS 7-102 and NAS 7-434 was to study the fundamentals of the sealing process in order to provide the understanding necessary to formulate design criteria for seals in liquid rocket engines. Since such an objective is extremely broad in scope, it was necessary to limit the contract study work to certain selected areas of seal technology.

The purpose of this design guide is to present the status of seal design, with particular emphasis on rotating seals. It is intended to be a source of information and give reviews on selected subjects. One of the most useful parts of the guide should be the bibliography.

Contents

Part II of the Design Guide contains design reviews of various seal types as obtained from studying the open literature. Parts III through X cover special seal topics that were studied during previous phases of contracts NAS 7-102 and NAS 7-434.

Part II is an attempt at reviewing the more important types of rotary seals and try to summarize the current design information. It was found that a complete review of the available literature was impossible in the time available. Therefore, more time was spent on the seals judged to be the most important, such as face seals.

Section H of Part II is a special study on labyrinth seals for explosive squibs that was carried out on this contract. This seal is for a very special application and not of as general an interest as the other sections.

Considerable work has been done on face seals under contracts NAS 7-102 and 7-434. The main part of this work is included in Part III. Through these studies and others in the literature, a more thorough understanding of how face seals work is being approached. Therefore, it was felt that these studies should be published here in order to give seal designers a better understanding of face seals and to what is being accomplished in face seal research.

Some of the face seal studies were carried out with the help of computer programs. These programs are given in Part IV. They are based on the solution of Reynolds equation for face seal geometry. One uses the short bearing approximation and cavitation boundary conditions to solve for a face seal lubricant film that contains gas cavities. The second program uses the finite difference approach to solve for various face seal film profiles and includes the effects of radial eccentricity, which can produce an "inward" pumping effect.

Extensive studies were made to see if a viscoelastic additive could be used effectively to reduce leakage in face seals. This work is presented in Part V. It is aimed at applications in the field of liquid rocket propulsion, however, it also gives some insight into face seal performance with non-Newtonian liquids.

Part VI is a brief review of the problems in cryogenic sealing. Material and fluid properties are an important factor here.

Perhaps the single most important factor in seal design is material selection. Part VII is devoted to the topic of materials for seal applications

Part VIII presents the combined seal data collected and published under NAS 7-102. This gives information on seal designs and should be helpful.

Part IX includes the bibliographies. This is perhaps the most important contribution of the Design Guide. It gives some appreciation for the extent of the literature on seals. Unfortunately, however, there are very few publications directed at comprehensive seal design.

Source Material

Much of the source material used for this document was developed under sponsorship of the National Aeronautics and Space Administration for contracts NAS 7-102 and NAS 7-434. The balance of the source material came from the open literature. References are given throughout the Guide and of course the Bibliography gives an exhaustive list of seal references.

In addition to the Seals Design Guide, the following references are recommended:

NAS 7-102 Final Reports (listed below)

NAS 7-434 Final Reports (listed below)

First International Conference on Fluid Sealing, BHRA, 1961

Second International Conference on Fluid Sealing, BHRA, 1964

Third International Conference on Fluid Sealing, BHRA, 1967

Fourth International Conference on Fluid Sea[illegible], BHRA/ASME/ASLE, 1969

Seal Symposium Issue, Trans. ASME Jo[illegible]ub. Tech., April 1968

AFRPL-TR-65-61, "Analytical Techniques for the Design of Seals for Use in Rocket Propulsion Systems," Vol. I Static Seals, Vol. II Dynamic Seals, IIT Res. Inst. for U.S. Air Force, May 1965

NAS 7-396, "Leakage Testing Handbook" Gen. Elec. for NASA-JPL, June 1967.

NAS 8-11523, "Zero Leakage Design for Ducts and Tube Connections for Deep Space Travel," Gen. Elec. for NASA-Marshall Space Flight Center, March 1967

NAS 8-4012, "Design Criteria for Zero Leakage Connectors for Launch Vehicles," Gen. Elec. for NASA-Marshall, Oct. 1967.

The work under contracts NAS 7-102 and NAS 7-434 was performed by personnel of the Mechanical Laboratory of the Research and Development Center, General Electric Company, Schenectady, New York. Primary emphasis was on a broad fundamental study aimed at establishing a basic understanding of the sealing function. During the initial phases of the study, attention was directed toward the development of some measure of order in the broad technology of seals and sealing mechanisms. Two extensive bibliographies plus several detailed studies covering the "state of the art" were produced in the early study work. As the study developed, the various "weak spots" in the technology became rather self-evident. Unfortunately, a large number of such "weak spots" in the technology were detected. Since it was not practical nor desirable, to devote detail attention to all such areas simultaneously, the contract effort was directed toward certain selected avenues of investigations. The results of the preliminary study work and the subsequent investigations is contained in the final reports. A list of the contract phase reports with appropriate identification numbers along with a brief description of the contents of each is given below. Final reports were issued at the conclusion of each contract phase.

"Study of Dynamic and Static Seals for Liquid Rocket Engines"

Conducted for NASA Headquarters and Jet Propulsion Laboratory under Contracts NAS 7-102 and NAS 7-434. Where the reports are available through the NASA Scientific and Technical Information Facility, the appropriate report numbers are given.

1. Feb. 26, 1962 to Feb. 25, 1963

   Volume 1 (N63-19595)

   Categorization System for Seals

   Tabulation of Seal Performance Data

   Description of NASA Seal Program

This report gives an overall description of the study in terms of the status of the technology of seals and the general strategy to be employed. Also included are 21 tables of detailed experimental data taken from published reports on seals.

Volume 2 (N63-19498)

Gas Permeation Through Solids

Analysis of Fluid Flow in Thin Films

Labyrinth Seals

Cryogenic Sealing

Two Phase Phenomena In Dynamic Face Seals

Seal Materials

Leak Detection and Measurement

Seven individual reports make-up this volume, each covering a specific subject pertinent to the technology of fluid sealing. Preliminary work on the General Seal Analysis is included in addition to initial studies in Leak Detection and Measurement.

Volume 3A (N63-19596)

Bibliography of ASTIA Literature on Seals with Subject Index

This is a bibliography of unclassified abstracts on dynamic and static seals as funished by the Defense Documentation Center, formerly the Armed Services Technical Information Agency. This volume contains 682 abstracts selected from the 1100 abstracts supplied by ASTIA. A subject index is included which may be used as a tool to obtain literature on seals from ASTIA. The ASTIA bibliography number assigned to this group of references is ARB2021.

Volume 3B (N63-19597)

Bibliography of Open Literature on Seals with Subject Index

This volume consists of 1482 abstracts of documents available from the open literature. The data and information was obtained from more than twenty-two major sources traced to their initial publication. This search intended to abstract every English language documents that it was feasible and worthwhile to obtain.

2. Feb. 26, 1963 to Nov. 30, 1963

Volume 1 (X64-16398)

Tabulation of Seal Performance Data Continued

Updating of Bibliographies

A brief review of the accomplishments of the Phase 2 effort is presented. The main portion of the material is devoted to an analysis of seal categories

by presenting 84 charts on experimental data on typical seal applications. These charts are useful as a ready reference for evaluation of particular seal designs in terms of known seal performance data.

Volume 2 (X64-16399)

Dynamic Face Seals - A Review

Seal Material Study

Fluid Surface Tension Sealing

Leak Detection in Gas Systems

Leak Detection Using Nuclear Techniques

An extensive presentation of the work in leak detection and measurement is included in this report. Results of an experment on sealing by fluid surface tension techniques, seals materials study and a review of dynamic face seal theory are also included.

Volume 3 (X64-16400)

Analysis of Fluids in Thin Films Continued

Streaming Birefringence

Concentric Cylinder Experiment (Non-Newtonian)

Parallel Disk Experiment (Non-Newtonian)

The report contains the material developed on the General Seal Analysis covering the behavior of fluids in thin films. The analysis is based on various non-Newtonian fluid models for which the appropriate equations of state are developed. Some experimental work on a modified coaxial cylinder and modified parallel plate arrangements are presented. Also included is a section on streaming birefringence, which is a technique useful in fluid flow studies.

3. April 1, 1964 to Oct. 1, 1964 (X65-10582)

Visco-Elastic Effects in Dynamic Seals

Normal Stress Measurement With a Jet Viscometer

Non-Contacting (Viscous and Centrifugal) Seal Study

Review Foreign Literature on Seals

Updating Bibliography on Open Literature

Liquid-Gas Leakage Correlation Experiments

Leak Detection Techniques Continued

This report contains a critical review of the available knowledge of viscoelastic effects and their relationship to dynamic seals. Data from some preliminary experiments at moderate shear rates are presented to give some estimates of the normal stress that may be developed in a high speed face seal. The subject on non-contacting seals of the viscous (screw) and centrifugal (slinger) type are treated rather extensively both in the laminar and turbulent flow modes. A report on the initial work toward establishing a liquid to gas leakage correlation is presented.

4. January 1, 1965 to September 1, 1965 (X66-13001)

   Visco Seal Turbulent Analysis

   Centrifugal Seal Experiments

   Visco-Elastic Jet Experiments Continued

   Liquid-Gas Leakage Correlation Continued

   Updating of Bibliography on Open Literature

   Preliminary Seal Design Guide

This report extends the Visco Seal theory to the turbulent regime and presents a semi-empirical correlation for Centrifugal (Slinger) Seal design.

5. January 1, 1966 to February 14, 1967

   Volume 1

   Visco-Elastic Jet Experiments Continued

   Non-Newtonian Seal Experiments (Face Seal)

   Newtonian Seal Study (Face Seal)

   Gas Leakage Correlation Experiments

   Updating of Bibliography

In this report the visco-elastic experiments were extended to the face seal geometry in order to verify predictions. Newtonian face seal theories were developed to study the effects of film cavitation.

   Volume 2

   Effects of Geometry and Inertia on Face Seal Performance

   (Laminar, misaligned, precessing with tangential waviness.

   Turbulent, misaligned precessing. Turbulent shaft seal.

   Cavitation due to centrifugal inertia.)

6. April 1, 1967 to April 1, 1968

   Analysis of Dynamic Face Seals

   (Cavitation and Gas Cavity Formation, Eccentricity and Misalignment,

   Inward Pumping or Reverse Flow Effects, Two-Fluids)

   Non-Newtonian Experiments (Torque and Leakage)

   Non-Newtonian Correlative Model

   Updating of Bibliography

This report contains extended face seal theories that cover turbulent flows with centrifugal inertia effects and explains the laminar flow inward

pumping phenomena due to radial eccentricity and misalignment. Face seal experiments confirm the cavitation theory, the inward pumping theory and the centrifugal inertia theory for laminar flow. Non-Newtonian experiments and correlation for face seals shows a significant effect on leakage, but not as much reduction as predicted from the Jet Viscometer experiments

B. Categorization of Seals

To facilitate presentation of the information on seals it is helpful to categorize seals into various generic classes. One convenient system is shown in Table 1. It will be noted that this system considers various types of motion as well as physical forces and geometric configurations normally used in seal designs.

As seen from Table 1, the rotary motion seal includes a broad category of contacting and non-contacting shaft seals for use on high speed and low speed rotating machinery. The turbopump shaft seal used on most liquid rocket engines is an example of a high speed rotary motion seal of the positive contact dynamic type. For low speed applications the lip and ring type may be employed. ~~The~~ labyrinth and ~~bushing~~, which are fixed clearance seals, have frequently been employed as pressure breakdown seals on rotating shafts.

Of recent interest are the centrifugal (slinger) seal and the viscous (screw) seal. These are non-contacting type seals which depend on speed to produce the sealing function. At the proper speed conditions the seal function is accomplished by the fluid centrifugal or viscous forces present in the sealing device. These can be a very effective seal systems capable of providing high reliability.

C. Seal Performance Requirements

For any seal application it is important that the performance requirements be established as completely as possible. The amount of allowable leakage, the expected seal reliability, the start-up and shutdown transients and the total expected life are parameters which must enter into the initial seal design.

These performance requirements must be reviewed in terms of know capabilities of various seals systems. This is especially true in the case of dynamic seals where performance is influenced by pressure and speed. For example, for low pressure systems (below 200 psi), the positive contact seal can be a low leakage device capable of attaining high reliability. However,

SEAL CATEGORIZATION SYSTEM

I. DYNAMIC SEALS

A. Rotary Motion

1. Selection & Control of Geometry
   - a. Positive Contact
     - (1) Face
     - (2) Packings
     - (3) Lip
     - (4) Rings
   - b. Controlled Clearance
     - (1) Labyrinth
     - (2) Bushing
2. Control of Fluid Properties
   - a. Freeze
   - b. Magnetic
3. Control of Fluid Forces
   - a. Centrifugal (Slinger)
   - b. Screw (Viscoseal)
   - c. Magnetic
   - d. Hydrostatic Face Seals
   - e. Hydrodynamic Face Seals

B. Oscillatory Motion

1. Selection & Control of Geometry
   - a. Positive Contact
     - (1) Face
     - (2) Packings
     - (3) Lip
     - (4) Rings
     - (5) Diaphragms
   - b. Controlled Clearance
     - (1) Labyrinth
     - (2) Bushing
2. Control of Fluid Properties
   - a. Freeze
   - b. Magnetic
3. Control of Fluid Forces

C. Reciprocating Motion

1. Selection & Control of Geometry
   - a. Positive Contact
     - (1) Diaphragm
     - (2) Packings
     - (3) Lip
     - (4) Rings
   - b. Controlled Clearance
     - (1) Labyrinth
     - (2) Bushing
2. Control of Fluid Properties
   - a. Freeze
   - b. Magnetic
3. Control of Fluid Forces

II. STATIC SEALS

A. Selection & Control of Geometry

1. Positive Contact
   - a. Elastic Deformation
     - (1) Gaskets
     - (2) Rings
     - (3) Diaphragms
   - b. Plastic Deformation
     - (1) Gaskets
     - (2) Rings
     - (3) Packing
   - c. Hermetic Sealing
     - (1) Welding
     - (2) Brazing
     - (3) Soldering
     - (4) Cementing
     - (5) Mold in place seals (metal injection etc.,) luting, caulking, depositing, plating
   - d. Other - Lapped
2. Controlled Clearance
   - (1) Labyrinth
   - (2) Bushing

B. Control of Fluid Properties

1. Freeze
2. Magnetic

C. Control of Fluid Forces

TABLE 1

for high speed, high pressure applications the positive contact seal will provide acceptable leakage only if adequate leakage flow is permissible. Non-contacting dynamic seals will provide essentially zero leakage in both high and low speed application, however, it may be necessary to use an auxiliary device to provide "zero speed" sealing.

In all cases, start and stop transients are very critical. This is particularly true for positive contact dynamic seals which are designed for high speed and high pressure operation. Special provision is required if the start transients are slow or if extended operation at off design conditions is required. The non-contacting dynamic seal may also require special provision for extended operation at off design conditions.

In many positive contact dynamic seal applications it is possible to increase reliability by allowing increased seal leakage or by decreasing face loading. The increased leakage serves to cool and lubricate the seal interface. The increased flow also allows the removal of wear debris which may otherwise have a very deleterious effect on seal performance. The decrease in seal interface loading reduces the amount of heat generated at the interface and thus reduces wear.

D. Status of Sealing Technology

In any fundamental discussion of seals and sealing mechanisms it is instructive to review briefly the status of the current technology. As presently known, the sealing technology is not general enough to be covered by a single unified theory for all seal concepts. Recent developments indicate that there are many similarities between dynamic seals and bearings.

For static seals, as outlined in the categorization chart, acceptable design criteria and procedures have been formulated. The basic phenomena are well understood and a satisfactory design may be evolved if the system requirements are not extreme. The basic considerations for a successful stationary seal can be summarized as follows:

1. Provision for suitable supporting structure to sustain imposed loads.
2. Provision for elastic or plastic deformation at the seal interface when necessary.
3. Smoothness and flatness of mating surfaces determines amount of deformation needed for effective sealing.
4. Random surface damage can preclude the possibility of obtaining a successful seal.

5. Suitable provision must be made to meet re-assembly requirements.

For the rotary type positive contact dynamic face seals no rational design basis has been evolved. Certain broad design criteria are available, but in general the frequently anomalous behavior of such a seal has made it an exceptionally difficult device to evaluate analytically. The pressure-velocity criteria and the pressure balancing techniques are useful relationships; however, they represent only an indication of the current status of the technology and not an understanding of the sealing mechanisms involved. In the last ten years a considerable amount of basic work has been done to understand the operation of the face seal. There is perhaps more literature on this seal type than on any other. The difficulty in developing a design analysis for face seals results from the following: (a) a long life seal depends upon good interface lubrication, (b) the interface irregularities which promote good lubrication are random and unplanned, i.e. not designed into the seal.

Some guide lines for positive contact dynamic face seals are listed below. These guide-lines represent a minimum condition and apply only within the known ~~limits~~ of applicable pressure-velocity relationships.

1. The mating surfaces of the primary seal elements form a friction couple. Therefore, the material combination used must be carefully selected, including considerations for the fluid to be sealed and the operating temperature.
2. Maximum dimensions of surface asperities and distortions due to temperature and pressure must be controlled to avoid damage to seal elements.
3. Contaminates in the working fluid or in the environment must be closed out from the seal interface.
4. For high reliability, lubrication in the form of additional leakage, past the seal interface, may be required.
5. Debris caused by seal face wear may be detrimental to seal life.

In order to overcome some of the shortcomings of the face seal, in recent years designers have developed the hydrodynamic and hydrostatic face seals. These seals use hydrodynamic and hydrostatic thrust bearing principles to develop a known and controlled thrust load and maintain a

finite clearance between the sealing surfaces. As such they are non-contacting seals. They are more expensive and usually applied to special situations where reliability is needed.

Two other types of non-contacting seals that have been developed recently are the viscoseal and the centrifugal slinger seal. These also are normally used in special applications only.

The viscoseal has received a great deal of attention and the analytical design methods are well developed. Extensive experimental work has substantiated most of the theory; however, it has also demonstrated the major shortcoming of this seal, gas ingestion. Gas ingestion has been observed over a wide speed range but is most significant at the higher speeds. The gas mixes with the sealed fluid in the seal clearance and reduces the effectiveness of the seal. At higher speeds the gas is pumped through the seal into the high pressure cavity.

The slinger seal has been used successfully in a number of applications. It requires relatively high speeds and fluid densities. It tends to generate heat that must be removed by a through flow of liquid. Theoretical design techniques are fairly good for this seal. However, the problem of liquid interface instability has not been solved analytically and in practice has required special geometries. With some fluids there will be a mixing with air in the clearance space of this seal also.

The advances in lip seal technology have been mainly in the area of materials and development of the helix seal, a lip seal with spiral grooves molded into the lip. Some theoretical design analyses have been attempted for this seal type, but in general there is no underlying theory.

Overall, rotary seal design can be broken down into two general areas. The first one includes the most common and most used seal types such as face seals, lip seals, etc. These seals are designed primarily from experience and only recently has much been done theoretically. The second group of seals are the newer concepts which have a relatively strong theoretical basis but are for special applications, are more costly and less experience exists.

E. Recommendations

Seal technology is in a state of rapid growth. There is much yet to be learned, yet there exists already a significant body of information. The problem is that this information has not been gathered together into a unified document. This design guide is an attempt to do this. However, the seal literature is so extensive that it is not possible to cover it adequately or to even discuss certain types of seals. Therefore, the first recommendation would be to extend this review and make it even more complete. In addition the following should be done:

1. Additional basic analyses and experiments are needed to establish a rational design basis for contacting type seals such as face seals, lip seals, circumferential seals and packings.
2. Design analyses for these seals should be developed that would calculate seal temperatures, thermal and mechanical distortions of the seal parts and dynamic tracking of seals under conditions of vibration and misalignment. Heat transfer in seals is important to good performance. It has received some attention, but not nearly enough.
3. Further work should be done on contacting seals to determine their performance when they are subjected to different combinations of sealed fluids, i.e., gas-to-gas, liquid-to-gas, and liquid-to-liquid.
4. Further design information is needed for the various types of hydrostatic and hydrodynamic face seals. This is especially true for applications where the seal separates a gas and a liquid, or two different liquids.
5. For viscoseals, the gas ingestion problem needs further study. This is a serious draw-back to this type of seal.
6. A better understanding of the liquid interface stability in centrifugal slinger seals is needed.
7. The problem of two phase, liquid-gas mixture sealing, such as wet steam, needs further study.
8. Existing design data, and materials information needs to be made more readily available to engineers.

CHAPTER II

SEAL DESIGN - ROTARY SEALS

A. Mechanical Face Seals
B. Hydrostatic and Hydrodynamic Face Seals
C. Bushing Seals
D. Viscoseal
E. Lip Seal
F. Slinger Seal
G. Labyrinth Seal
H. Labyrinth Seals for Explosive Squibs

A. MECHANICAL FACE SEALS

I. Introduction

Mechanical face seals have been used in industry since the early nineteen hundreds. They are unique in their ability to seal with negligible leakage under severe conditions, and consume relatively little frictional energy. They are reliable, have long life, and require little maintenance. Their major application is to seal liquids under pressure.

The function of a face seal is to prevent leakage where a rotating shaft pierces a housing or wall. Cross-sections of several typical types of face seals are given in Figure 1. The basic concept involved appears simple at first thought. Two smooth faces, one a rotating ring affixed to the shaft, the other a stationary ring secured to the housing, are pressed together. This decreases the clearance between the faces, thus decreasing the size of the leakage path to limit or prevent fluid leakage.

Essential Physical Features of Mechanical Face Seals

Empirical experience has produced features common to all mechanical face seals. These features are:

1. Sealing is accomplished by holding two mating, annular faces together.
2. The faces are pressed together by light spring loading, and by fluid pressure acting on the rear of one of the sealing faces. The amount of fluid pressure so employed varies according to the "balance" of the seal.
3. The two annular surfaces in contact are made from dissimilar materials capable of operation under boundary lubrication conditions. Low coefficients of friction, good heat conductivity, are sought to prevent high temperatures and breakdown of face materials.
4. The face seal mating surfaces are lapped flat to within a 2-5 light bands of flatness (24 to 60 microinches); the resultant finish (surface roughness) is approximately 10 to 20 microinches.
5. A certain minimum amount of roughness and/or waviness of the seal faces should be considered an essential feature of a seal. Excessively rough surfaces can make for high coefficients of friction, excessive wear and for short life. However, exceedingly smooth surfaces have resulted in the failure of seals and bearings. The ability to provide load bearing is lost. Such failure has been attributed to the lack of hydrodynamic pressure generation that variations in surface height would have produced, or the inability of the surfaces to retain a lubricant [3].

6. The shaft mounted face is flexibly mounted and sealed to the shaft so that the mating faces are held in contact in spite of shaft vibration or eccentricity.
7. Auxiliary "static" seals, usually "O" rings, are provided to seal the mating faces to the shaft or machinery housing.

Performance Limits

The practical operating limits of seals are generally stated to be as follows:

Pressure - Figure 2 is a compilation from the published data of various seal manufacturers of the pressures that can be sealed readily with a face seal. Important is the fact that higher pressures can be sealed with balanced seals than with unbalanced seals. Typical maximum pressures for unbalanced seals are in the order of one to two hundred psi. Balanced seals are used up to pressures in the order of 1000 psi or more.

This originally raised the question why relatively small changes in balancing can alter the maximum sealing pressure so [4] . This difference in duty can be ascribed to the different lubricating regimes that can exist between the seal faces.

Face Speed - Manufacturers' recommendations are for maximum shaft speeds in the 6000 rpm to 15,000 rpm range. The maximum speed noted for any face seal is about 30,000 rpm.

Life - In conservative refinery practice, typical seal life is noted as being about 2000-8000 hr [5] . In domestic water pumps, life is probably in the order of 15,000 hr, i.e., 2 years of continuous service. For very high face velocities and poor lubrication, such as for a cryogenic liquid seal in a rocket engine turbopump, life may be only a matter of minutes.

Leakage - Existing data is unreliable since in practice a liquid may often evaporate before it can be measured. For a "properly" designed seal, an ticipated leakage can range from "zero" to about 10 cc/min. Leakage is generally reported to increase with speed. Some find the converse: a stationary seal leaks badly, but improves in sealing with increasing speed.

Principle of Operation - The realization that this seal is in many respects a complex device; that boundary lubrication, cavitation, or a full fluid film can exist between the faces depending upon conditions; that conceivably various different mechanisms can act to provide load bearing between the faces (as well as prevent leakage); are thoughts of relatively recent origin. Indeed, the past decade has seen considerable exploration of the face seal as

a fluid film device, motivated by the hopeful expectation of improving the operation of the seal. Nevertheless, the theory of face seals has not in toto progressed to the point of providing a general, useable, "textbook" type of rational design approach based on analytical foundations. The scope of existing, comprehensive studies is somewhat limited; many important areas still lie unexplored, both analytically and experimentally. The range of major studies is blocked out in Figure

Thus, even though certain mechanistic approaches are well advanced, it cannot as yet be said whether the face seal as it basically stands today--evolved largely by trial and error testing--has resulted in about the best seal possible when used properly; or whether advances such as longer life, lower leakage, operation at greater extremes of rotational speed or fluid pressure, can ultimately be obtained.

Undoubtedly a primary reason for this slow advance is the small dimensions of a face seal making direct examination of a lubricating or sealing mechanism exceedingly difficult. Furthermore, the wide range of conditions possible as to seal materials, dimensions, alignment, fluid pressures, bearing loads, rotational speeds; as well as the difficulty in assuming what may or may not be a dominant mechanism (or mechanisms) confuse the issue. Small differences in dimensions as well as manufacturing tolerances may produce considerable variations in performance. Judging by progress over the past decade, one may expect continued exploration and increasing knowledge, leading to a slow evolution of design techniques.

A seal is also a thrust bearing. Thus it has the same basic problems as a bearing, as well as additional requirements stemming from its sealing function. A review of theory must include as evidence the performance of a seal, show how this is related to physical form, and the lubrication regimes that exist. Boundary lubrication fundamentals are important to a class of seals; practice here is well established. Mixed and fluid film lubrication are also found in seals. For these, the hydrodynamic generation of pressure for load bearing and sealing between faces is an interesting, as well as speculative subject.

Initially, the theories or approaches applied to face seal design were simple. For example:

1. It is an intuitively obvious concept that seal faces can be forced together to prevent leakage. Thus it seemed reasonable to design by experimentally increasing the external load on

the faces until leakage became tolerable, seeking a compromise versus increasing wear and shorter life.

2. Surface tension acting at the interface between liquid and air, at the low-pressure edge of the seal faces, creates a pressure differential to contain the liquid. The sealing pressure so created can be computed knowing the surface tension, and assuming that the maximum gap width between seal faces is equal to the sum of the waviness of the two face surfaces [6]. For common fluids, sealing pressure in the order of 25 to 50 psi can thus be inferred.

3. In high speed seals, centrifugal force acting on fluid present between seal faces that are not in physical contact can be used to oppose the hydrostatic pressure causing inward radial leakage [7].

In contrast, much of the recent literature is concerned with how the physical features of face seals possessing liquid films between the faces are related to seal performance. How important are these features? Should they be modified? Thus, for example, are better or more poorly finished surfaces desirable? Can distortion, vibration, and eccentricity and misalignment be put to advantageous use?

In summation, practical knowledge of boundary lubrication in a seal is far more advanced than knowledge of hydrodynamic effects. That is, it is possible to design a seal and select suitable face materials from the standpoint of <u>friction</u> <u>and</u> <u>wear</u>. It is not possible on a broad basis to select or isolate specific hydrodynamic pressure generating mechanisms to be used, and thereby deliberately create load bearing pressure with pressure variations to oppose leakage. An exception to this would be the use of centrifugal force in a high speed seal.

Another exception to this is the use of hydrodynamic and hydrostatic face seals. These are not conventional mechanical face seals, rather, they are designed partly as a thrust bearing and partly as a face seal. These types of face seals will be discussed in a following chapter.

II. Seal Design

A. Seal Specifications

Before the design or selection of a mechanical face seal can be accomplished, it is essential that the conditions under which the seal must operate are known and accounted for. Some of these factors are listed in the design specification sheet of TABLE 1. After each of these aspects are factored into the design appropriately, the face seal should operate reliably for a satisfactory period of time. This "satisfactory" period of time may range from three

months for hot concentrated mixed acids to seven years for a simple water pump [8].

There is no dependable shortcut for a good seal design. The seal is often a critical component of a machine design and it is not sound judgment to assume that because of its small size and relative cost that it does not deserve a first rate engineering design effort.

It may be found that several different seal designs can be used for a specific application. In this case cost or convenience may determine the final selection. There is no all-purpose seal suitable for all applications. Each seal application must be treated as an individual case.

Before proceeding to discuss the various design factors it may be helpful to record here the opinions of several seal engineers. In summarizing the fundamentals of face seals Reference [9] makes the following statement: "Seal engineering is relatively young. It is emerging from a state-of-the-art knowledge to a science. No adequate textbook on the subject has been written. Handbooks treat the subject like an orphan. Much must be accomplished yet, but the future looks bright." From an applications point of view a manufacturer made the following statement [10]: "As a seal manufacturer I must point out that one of the major problems that we have to face today is a surprising ignorance on the part of seal users as to the exact operating conditions for which a seal is required and reputable seal manufacturers today are quite capable of providing a mechanical seal which will work under a very comprehensive range of operating conditions if they can be accurately specified.

In my own experience failure due to shortcomings in design or lack of understanding of operating conditions and requirements are less than 1% of the failures that we investigate. The highest proportion were due simply to inadequate cooling of the seal ..... The majority of the remaining failures were due to the seal being called upon to operate under conditions other than those as originally specified, for example, abrasives in suspension not mentioned ...., acid carry-over in certain types of chemical process, dry running through automatic controls shutting off pump suction or cleaning processes such as steaming out."

TABLE 1

Design Specifications for Mechanical Face Seals

Specifications	Design Considerations
Application:	
machine; cryogenic turbo pump etc.	choice of face seal type and cost
function;	"
duty cycle	"
performance	"
environment	" ,materials
reliability	perhaps consider non-contacting type
life	" , wear, matl.
leakage (containment)	" compound seal with buffer fluid
assembly	internal vs. external seal, care in assembly
maintenance	"
vibrations	mechanical design, natural frequencies, stresses
shock	"
mechanical factors	"
electrical factors	materials
chemical factors	"
radiation factors	"
Fluid to be Sealed:	
type; gas, liquid	
specification, identification	
concentrations, additives, cleaning operations	materials used
properties; viscosity	lubricating quality of fluid
density	fluid centrifugal forces, leakage, $\Delta\rho$
vapor pressure	flashing in the interfacial film, loss of lub. film
chemical properties; corrosive	materials, need for containment, buffer fluid, special seal
flammable	" " " "
toxic	" " " "
decompose	increased wear, need for flushing, filtering, separating
crystalize	" " " "
abrasives; dirt, rust, slurry, cleanliness	" " " "

TABLE 1 (continued)

Specifications	Design Considerations
flooded, dry, splash	need for positive lubrication, dry operation
long periods of idleness	" " "
fluid separation; liquid/liquid	inward pumping of fluid or gas, special design
liquid/gas	" " " cavitation
gas/gas	" " "need for lul
Temperature:	
maximum	material, thermal stresses, distortion, uneven wear
minimum	" " "
normal operation	" " "
variation, cycling	" " "
thermal shock	" " "
ambient	" " "
temp. rise from friction in seal	" " " and need for cooling seal. For high temp. consider bellows seal.
Pressure:	
maximum	materials, stresses, distortion, uneven wear, face load, balance
minimum	
normal	
variation, cycling	
sudden change	
Speed: (shaft)	
maximum	face speed, face wear, fluid centrifugal forces, centrifugal forces and stresses in seal parts, rotating balance, friction heating, power loss
minimum	
normal	
variation, cycling	
continuous, intermittent, idle periods	
Size: (dimensions)	
shaft diameter	face speed, stresses, distortion, alignments, space for compound seals, cooling lines, separators, filters, etc.
max. o. d.	
length	

TABLE 1 (continued)

Specifications	Design Considerations
tolerances	
misalignment	face finishes, materials, imperfections
end play	
shaft deflection	
shaft run out and eccentricity	inward pumping

B. Seal Components

Primary Seal Faces

Seal Head: May be either stationary or rotating.
Choice usually depends upon the direction of pressure.
For high speed shafts the head is usually the stationary member because of the dynamic balance required.
Head includes a secondary seal and spring, or bellows.
Usually is the replaceable member of the seal couple.
Often made of iron, steel or ceramic.
Must be aligned with shaft.
Stress in parts important design factor.

Seal Seat: Also requires some sort of secondary seal.
Secondary seal does not have to slide axially.
No spring needed.
Seat is fixed either to the housing or to the shaft.
Often made of graphite.

Secondary Seals

For Head

Pusher-Type Seal (Sliding Seal - more susceptible to dirt, which may increase its sliding friction, than is the non-pusher type)

O-Ring: Sealing effect provided by the hydraulic pressure and mechanical preloading.

V-Ring: Sealing effect provided by spring force and hydraulic pressure force in the seal causing a mechanical preload to the shaft. Seals at the shaft surface and at the housing mating surface by the action of the spreader element forcing the wings of the seal out.

u-Cup: (same as V-Ring)

Wedge: Mechanical and hydraulic forces provide sealing force.

Modified Wedge: (same)

Piston Rings: (same)

Non-Pusher Type Seal (Mechanical design important so as to give the right hydraulic load (mean effective diameter) and spring rate. Must be strong enough to withstand pressure distortion).

Metal Bellows: Used primarily where an all-metal seal is necessary because of high temperatures or corrosive liquids. Consideration should be given to incorporating a vibration damper in the design and avoiding seal

natural frequencies near the machine natural frequencies or running speeds.
May be formed or welded discs. They form a seal with practically no sliding friction and also provide a spring force for the seal head. This force may be augmented by a separate spring. Static seal provided by a welded joint.
For more information on the hydraulic forces and the mean effective diameter, see the discussion on seal balancing. High pressures may deflect the bellows radially enough to alter its effective diameter.

Molded Elastomer: Must be preloaded to shaft to provide static sealing. Adhesive used as well.

For Seat

Static seals listed under pusher type, above, may be used even though axial motion is not present with the seat.
Any other type of static seal such as gaskets, integral seal, ground surface, etc.

Springs (To provide a mechanical axial force that presses the primary seal surfaces together. The spring is mounted to the seal head.)

Single Spring: Design the end of the spring so that it presses with a force which is uniformly distributed around the circumference of the seal head.
Consider corrosion of the metal in choosing materials and selecting size of cross-section. In high speed applications take into account the centrifugal force on the spring for rotating head designs. This centrifugal force will tend to change the applied axial force.

Multiple Springs: In this design the circumferential distribution of the axial force can be made symmetric and relatively uniform by the placement of the springs. The spring rate can be varied by changing the number of springs used. The effect of centrifugal forces on these springs is less than for the single spring.

Spring Washers: These elements take up a minimum of axial length but usually have higher spring rates which may load the seal faces more than desired. Material selection tends to be more limited.

C. Seal Arrangements

Tandem seal arrangement for handling high pressures.

The total pressure differential is broken down into smaller pressure differentials, using as many tandems as necessary. In this way each seal sees only a fraction of the total pressure differential.

Externally Mounted Seal [11]

1. Simplify installation and removal, and adjustment.

2. Pressure normally on the inside of the seal surfaces placing the seal rings in tension.
3. Can be designed to blow open at a specific pressure, thus acting like a relief valve. This may be a disadvantage to this design if the sealed fluid is hazardous.
4. Determining the cause for seal leakage may be easier because the seal can be seen.
5. Usually contain a blind cavity in the rotating element that can act as a centrifuge and fill with solids from the liquid. This can lead to loss of axial motion and eventual seal leakage.
6. More difficult to cool seal faces.

Internally Mounted Seal [11]

1. Tends to keep itself clear of solids in the liquid.
2. Pressure on outside of seal faces.
3. Will tend to contain over-pressure if not overbalanced.

Split Seals

Where necessary for installation and replacement.

Primary seal rings and secondary seals are both split.

Rotating Head

Usually require fairly close shaft tolerances.

Take into account the direction of the pressure stresses on the seal members. If pressure is on the outside of the seal faces (internally mounted seal) then the seal rings are under compression. They are under tension if the pressure is on the inside of the faces (externally mounted seal).

Stationary Heads

Used in high speed applications to simplify the dynamic balancing and to reduce the centrifugal forces and stresses on the moving parts. These forces could act on the spring to move the head axially.

May, however, require closer tolerances on the housing bore and finish.

Stationary Seat

This is the most common arrangement. Quite often hard, brittle materials are used for the seat and a stationary seat mounted in the housing places this material under compression.

Rotating Seat

More limited and critical design, and choice of materials.

Trouble Shooting [11]

Leaks: 1. If the seal is tight while stationary, but leaks when rotating, the faces are flat but probably not square, or the shaft is oscillating.

2. If the leak is steady, it may be from non-flat faces, secondary seal, or gaskets.
3. Intermittent leakage may indicate thermal distortion of the faces.

D. Balance

Axial Forces on a Seal

It is necessary to consider the axial forces acting on a seal face in order to define the "balance" of a seal and relate balance to seal performance. Two external forces act to press the seal faces together. They are:

1. A hydrostatic force F_p, proportional to the pressure being sealed P_p. Most of the load on the faces is produced in this way by the fluid being sealed.
2. A spring force F_s. This force is relatively small. It is used to ensure contact between the faces before the pressure being sealed has been built up. The spring force must be large enough to overcome the static friction F_f of the "O" ring.

Fluid pressure between the seal faces P_h (when fluid is present between the faces to exert pressure) opposes these external forces, and acts to separate the seal faces. This pressure can be due primarily to leakage flow across the faces, and be determined by the characteristics of the flow process acting to produce a pressure drop. If the faces are close enough, and speeds high enough, hydrodynamic pressure generation caused by surface irregularities or other causes should be superposed on this pressure drop. If however, the faces are pressed tightly together, fluid should be excluded, as evidenced by a lack of fluid pressure as reported by Mayer [12].

For the first case, suppose that the seal faces are parallel and separated by fluid. Then laminar leakage for a continuous liquid film between the faces, results in a linear pressure distribution between inlet and outlet (Figure 5). This follows when the width of the seal face is small with respect to the radius of the seal, and a constant viscosity is assumed. The net force F_h acting on the rotating seal face due to this leakage pressure is $(P_p/2)A_h$, where A_h is the contact area of a face.

Unbalanced Seal

The unbalanced seal is the simplest form of seal (Figure 1a). It is fitted to a shaft of constant diameter. As a consequence, the contact area A_h between the sealing faces is equal to the area A_p on the rear of the sealing ring upon which the hydrostatic pressure of the fluid being sealed acts (Figure 1b Presumably, the pressure limit for sealing is reached when the forces on the

seal head are sufficient to either: 1. Blow the seal open or, 2. press the faces together tight enough to produce physical contact and boundary lubrication; with attendant high friction, heat release and wear.

Balanced Seal

To overcome the pressure limitation of a few hundred psi, a seal is "balanced" by using a stepped shaft (Figure 1b). This reduces the loading on the seal faces resulting from the hydrostatic pressure P_p. The usual limit to balancing is about one third. One third of the pressure is balanced out, leaving two-thirds of the hydrostatic pressure (P_p) to act on the seal face. That is:

$$F_p = (1 - \beta)\, P_p\, A_p = 2/3 P_p\, A_p,$$

where β is termed the amount of balance. Alternately, $(1 - \beta) = \eta$ is also used to designate the extent of balance [13].

Should the balancing limit of $\beta = 1/3$ be much exceeded, it has been found that the seal faces tend to be in an unstable condition [5]. Balancing can only be carried so far, i.e., up to $\beta = 1/2$ as an ideal limit. For a β larger than this, once the faces are slightly forced apart by motion, misalignment, flashing of liquid into vapor, dirt, etc., the faces will open further by themselves, and subsequently be unable to close.

To see why this should be so, let us consider the axial forces that act on the seal. For the seal gap (h) not to open, the sum of the axial forces must act to press the faces together. That is:

$$(F_s + F_p) > F_h$$

For a linear pressure distribution between the seal faces:

$$\left[F_s + (1-\beta)\, P_p\, A_p \right] > \frac{P_p}{2} \quad A_h$$

Neglecting a relatively small F_s:

$$(1 - \beta)\, P_p\, A_p > \frac{P_p}{2} A_h$$

For $\beta = 1/3$:

$$(1 - \beta) = \frac{2}{3} > \frac{1}{2} \frac{A_h}{A_p}$$

$$\frac{2}{3} > \frac{1}{2}$$

A_h in practice is only slightly larger than A_p. This shows therefore, that a 1/3 balance acts to provide a net force keeping the faces together, and that a much greater degree of balance cannot do so. If the balance is too close and the spring force is relied upon to keep the faces in contact, the seal will be too sensitive to accuracy of spring setting and changes in spring rate.

Actually, to keep the faces together, the net static forces acting on the seal ($F_s + F_p - F_h$) must be greater than zero for various reasons. There must be a margin sufficient to overcome vibration, part misalignment, and "O" ring friction. In addition, the leakage pressure term ($P_p'/2$) may increase (or decrease) because the seal faces may cease to be parallel under operating conditions because of wear, thermal gradients and deformation under load.

Suppose it were possible to press the seal faces tightly together, leaving no gap ($h = 0$), so that full contact with no leakage were achieved. Then the net fluid pressure between the faces due to the leakage of fluid between them would approach zero (Figure 5). Or, suppose that the leakage clearance diverged in the direction of flow to reduce the net fluid pressure. Under these conditions a balance larger than $\beta = 1/3$ would be possible. The fact that a $\beta = 1/3$ has been stated to work satisfactorily until the faces move apart [4, 5] implies that the average fluid pressure between faces can be less than $P_p/2$.

Balance Diameter

It is convenient to speak of pressure balancing in terms of the "balance diameter." This is a physical dimension on the balance seal head. It is shown in Figure 6(b), 7 (b), 8 (b). The hydraulic forces on the seal head are determined by this diameter and the face diameters d_1 and d_2.

When the seal pressure acts on the outside diameter of the seal, the hydraulic seal area is

$$A_p = (1-\beta)\, A_h = \frac{\pi}{4}\,(d_2^2 - d_B^2).$$

Therefore, when an interface film exists, the net hydraulic force on the seal head is

$$\begin{aligned}(F_p - F_h) &= P_p\,(1-\beta)A_h - P_p\,A_h/2 = P_p\,A_h\,(.5-\beta)\\ &= P_p\,\frac{\pi}{4}\left[(d_2^2 - d_B^2) - \frac{1}{2}\,(d_2^2 - d_1^2)\right]\\ &= P_p\,\frac{\pi}{4}\left[\frac{d_2^2 + d_1^2}{2} - d_B^2\right] \approx P_p\,\frac{\pi}{4}\,(\bar{d}^2 - d_B^2)\end{aligned}$$

where

$$P_p = (P_s - P_o)$$

$$\bar{d} = (d_2 + d_1)/2.$$

The seal balance is, then,

$$b = \frac{F_p}{P_p A_h} = (1-\beta) = \frac{d_2^2 - d_B^2}{d_2^2 - d_1^2} = \frac{(d_2 - d_B)}{(d_2 - d_1)} \cdot \frac{(d_2 + d_B)}{(d_2 + d_1)} = \begin{cases} 1.0 \text{ unbalanced seal} \\ .67 \text{ balanced seal} \end{cases}$$

$$\approx \frac{d_2 - d_B}{d_2 - d_1}\;.$$

The unbalance ratio for the seal is

$$B = \frac{(F_p - F_h)}{P_p\;A_h} = (.5 - \beta) = \left(\frac{d_1^2 + d_2^2}{2} - d_B^2\right) \Big/ (d_2^2 - d_1^2) = \begin{cases} .5 \text{ unbalanced} \\ .17 \text{ balanced design} \end{cases}$$

$$\approx \frac{\bar{d}^2 - d_B^2}{d_2^2 - d_1^2} \approx \frac{\bar{d} - d_B}{d_2 - d_1}\;.$$

The unbalance ratio, B, is convenient to use since it is a measure of the unbalanced hydraulic force tending to close the seal surfaces together. That is, if

(a) $B = .5$; no balance and large hydraulic closing force

(b) $B = .17$; balance design and hydraulic closing force is reduced

(c) $B = 0$; no hydraulic closing force and unstable seal operation may result

(d) $B < 0$; hydraulic force tends to open the seal faces

(e) If $P_p < 0$, i.e. $P_o > P_s$ then the seal pressure is on the inside diameter of the seal and the seal balance, b, and seal unbalance ratio, B, must be redefined as follows;

$$b = \frac{d_B^2 - d_1^2}{d_2^2 - d_1^2} \approx \frac{d_B - d_1}{d_2 - d_1}$$

$$B \approx \frac{d_B - \bar{d}}{d_2 - d_1}$$

Influence of Static Seal on Balance Diameter [14]

In the previous discussion the friction of the static seal and its influence on the balance diameter has been neglected. For the unbalanced seals shown in Figure 6-8 it was assumed that the effective area over which the sealed pressure P_s acted was equal to A_h. We see from Figure 9 that this area may be slightly larger, depending upon the clearance between the seal head and the shaft. This area, A_c, may not be negligible, especially in high pressure seals. Actually, the hydrostatic pressure on the static seal (O-Ring, etc.) is partially transmitted to the shaft. From Figure 10a and 10b it can be seen that an O-Ring may either increase or decrease the sealing diameter (sealing area A_p). Test results indicate that 5 to 10% of the cross sectional diameter of the O-Ring is active on the opposite surface.

Bellows Mean Effective Diameter

When balancing a bellows type seal it is recommended that the bellows and seal faces be so constructed that the average seal face diameter, $\bar{d}$, and the bellows mean effective diameter, d_m, are approximately equal. The bellows mean effective diameter is approximately the average diameter of the bellows. Usually the bellows mean effective diameter is determined by testing when it is desired to know it accurately. Deflections can cause the mean effective diameter to shift at high pressures.

With the bellows seal, then, balancing is accomplished by adjusting the mean effective diameter and not by stepping the shaft and adjusting the various seal hydraulic areas as was done with the pusher type seal. In Figure 11 a formed metal bellows seal is shown. The mean effective diameter of the bellows, d_m, is the diameter at which the net pressure force acts on the bellows. It is given by the following expression

$$d_m = \frac{2\,(d_o^3 - d_i^3)}{3\,(d_o^2 - d_i^2)} = \frac{(d_o + d_i)}{2} + \frac{1}{6} \cdot \frac{(d_o - d_i)^2}{(d_o + d_i)}$$

$$\cong (d_o + d_i)/2 \text{ for } (d_o - d_i) \ll d_o,\ d_i.$$

From this expression the net hydraulic closing force can be calculated. Note that the larger the seal average diameter, $\bar{d}$, is in relation to the bellows mean effective diameter, d_m, the greater the closing force, for the sealed pressure on the outside diameter the seal. If the pressures are reversed this will be an opening force. For this case, then, the relative diameter sizes should be reversed, i.e. $d_m > \bar{d}$.

How is this balance equation related to the balance equation for the pusher type seal? The net closing force for the pusher type seal is

$$(F_p - F_h) = P_p\,(1-\beta)\,A_h - P_p\,A_h/2$$

$$= P_p\,A_h\,(1-\beta - .5)$$

$$= P_p\,A_h\,(.5-\beta)$$

where $0 \leq \beta \leq .5$ to prevent blowing open and $\beta \approx .33$ for design. Relating this to the bellows seal can be accomplished by calculating the unbalance ratio,

$$B = \frac{F_p' - F_h}{P_p\,A_h} = (.5 - \beta) = \frac{\bar{d}^2 - d_m^2}{d_2^2 - d_1^2} \cong \frac{\bar{d} - d_m}{d_2 - d_1}\ .$$

Comparing this expression to the unbalance ratio for the pusher type seal shows that the mean effective diameter of the bellows is analogous to the balance diameter.

It should be noted that the bellows seal does not have as much frictional damping as the pusher type seals and, therefore, is more susceptible to vibrations. This is particularly true if $\bar{d} \cong d_m$ and only the spring force acts on the seal head.

Seal Interface Film Effects

There is a very important implication in the face seal balance inequality

$$F_s + F_p > F_h = P_p\,A_h/2.$$

It is this: "If $F_s + F_p > F_h$, then, $F_h \neq P_p\,A_h/2$".

This follows from the fact that the balance expression is an inequality which means that the closing force exceeds the hydraulic force on the seal face, F_h, The force in the wall of the bellows convolution, due to the pressure, may be found by taking moments about d_m; i.e.

$$F_c \ (d_m - d_i) = F_b \ (d_o - d_m)$$

or

$$F_c = F_b$$

since d_m is half way between d_o and d_i. Therefore, the pressure force exerted by the bellows is

$$F_b = \frac{1}{2} (P_s - P_o) \cdot \frac{\pi}{4} \ (d_o^2 - d_i^2).$$

The hydraulic force is

$$F_p = F_b + \frac{\pi}{4} \left[P_s \ (d_b^2 - d_o^2) - P_s \ (d_b^2 - d_2^2) + P_o \ (d_o^2 - d_a^2) - P_o (d_1^2 - d_a^2) \right]$$

$$= \frac{\pi}{4} \left[\frac{1}{2} (P_s - P_o) \ (d_o^2 - d_i^2) + P_s \ (d_2^2 - d_o^2) + P_o \ (d_o^2 - d_1^2) \right]$$

$$= \frac{\pi}{4} \left[\frac{1}{2} (P_s - P_o) \ (d_o^2 - d_i^2) + P_s \ d_2^2 - P_o \ d_1^2 - (P_s - P_o) \ d_o^2 \right]$$

$$= \frac{\pi}{4} \left[-\frac{1}{2} (P_s - P_o) \ (d_o^2 + d_i^2) + P_s \ d_2^2 - P_o \ d_1^2 \right].$$

In order to examine to possibility of the seal blowing open, the net hydraulic closing force on the seal surfaces must be considered,

$$(F_p - F_h) = F_p - \frac{P_p}{2} \cdot \frac{\pi}{4} (d_2^2 - d_1^2)$$

$$= \frac{\pi}{4} (P_s - P_o) \frac{1}{2} \left[(d_2^2 + d_1^2) - (d_o^2 + d_i^2) \right] .$$

$$= \frac{\pi}{4} (P_s - P_o) \left\{ \left[(d_2 + d_1)^2 + (d_2 - d_1)^2 \right] - \left[(d_o + d_i)^2 + (d_o - d_i)^2 \right] \right\}$$

$$= \frac{\pi}{4} (P_s - P_o) \left\{ \left[\bar{d}^2 + \frac{(d_2 - d_1)^2}{4} \right] - \left[d_m^2 + \frac{(d_o - d_i)^2}{4} \right] \right\}$$

$$\simeq P_p \frac{\pi}{4} (\bar{d}^2 - d_m^2) \text{ for narrow bellows and seal.}$$

and, therefore, there must be an additional force acting on the seal face. This leads to the following alternatives:

1. The faces touch and no film exists or only a partial film exists, or:
2. $F_h > P_p \ A_h/2$ and a film exists.

From experience it is certain that face seals operate under both of the above conditions. Under condition 1 the materials used for the faces becomes very important and this is the reason that carbon graphite is such a popular face seal material, i.e. it provides dry lubrication of the faces and controlled wear. Because under condition 1 the wear rate is greater and friction (heat generation and power loss) higher, the seal will usually last longer and operate more successfully under condition 2. The condition whereby $F_h > P_p \ A_h/2$, exists due to hydrostatic and/or hydrodynamic pressure build-up between the faces. Hydrostatically this occurs when the film thickness between the faces is convergent in the direction of leakage. Also this gives a positive "spring constant" to the film, i.e. the hydrostatic face force F_h increases as the average film thickness gets smaller. Thus a "controlled clearance" is established with its magnitude depending upon $(F_s + F_p)$ and the radial contour of the faces.

If this radial contour is such that the film thickness is <u>divergent</u> in the direction of flow, then, $F_h < P_p \ A_h/2$ and tends to draw the faces even more tightly together. Normally this condition is to be <u>avoided</u>.

Hydrodynamic pressures are generated in the interface film because of surface "waviness" and variations in the circumferential direction. Fortunately this force also results in a positive film "spring constant." Its magnitude, of course, also depends upon the seal rotational speed.

These hydrostatic and hydrodynamic forces usually occur due to unplanned irregularities which are small enough to be somewhat beyond the control of the designer. They are, however, not entirely beyond his control and he should exercise this control for best performance. Particularly, he should make the radial contour slightly convergent and avoid divergence. Also he should not go to great pains to make the surfaces completely flat. A small amount of waviness can be advantageous.

Because of the importance of these effects, they are discussed further in section 2B and in Chapter 3 .

Chapter 2B is on hydrostatic and hydrodynamic face seals. It should be evident now why they are of interest, i.e. to maintain a fluid film between face surfaces.

E. Operating Parameters

1. Mechanical Effects

High Fluctuating Pressures

The fluid pressure acts on the seal in two ways, axially and radially.

As discussed previously, the detrimental effects of excessive axial pressure forces can be overcome by seal balance. If this is not done properly the result is:

(a) For unbalanced seals the face load becomes excessive, squeezing out the fluid film, bring the surfaces in contact, resulting in friction, overheating and high wear rates.

(b) For seals in neutral balance the spring force is relied upon to keep the surfaces in contact and an unstable condition may result where the seal head oscillates.

(c) For over balanced seals the hydraulic force becomes an opening force and when it exceeds the spring force the seal blows open.

Pressures also exert forces in the radial direction, tending to distort the various parts of the seal non-uniformly. This is critical because the seal faces are lapped within light-bands of flatness. The cross-section of the seal parts should be large enough to avoid over distortion. The design will depend upon the pressure level and the elastic modulus of the materials used.

The pressure can cause an angular deflection of the seal ring if the secondary seal acts as a pivot point. This condition is termed "coning". If this condition is maintained for some time the surfaces will wear-in and may function properly. After the pressure is reduced the distortion is relieved and a change in face pattern has resulted. While distorted the seal balance may be altered and in the undistorted condition the face pattern may have changed in such a way as to alter the interface film pressures in a deleterious fashion, i.e. give a radially diverging film thickness. If the pressure fluctuates the condition is even more troublesome. The face wear patterns will continually change with a resultant loss of flatness and increased leakage.

In addition to using heavier cross-sections or materials with a high modulus of elasticity, the secondary seal may be placed so as not to act as a pivot point. Complex geometric shapes are also designed with the idea of counteracting distortions so that the final effect is a flat, parallel face [9] .

Shaft Deflection

Shaft deflection at the seal may result in seal leakage especially if the deflection changes after the seal faces have been grooved by wear [9] . Shafts should be designed with adequate stiffness so as to minimize the deflections caused by radial loads.

Misalignment

If the plane of the seal faces is not perpendicular to the shaft, the flexible seal member must wobble in order to maintain contact with the fixed face [9] . Of course, if contact is not maintained leakage will result. High speeds may accentuate this problem.

Horizontally split seals can be especially troublesome because the two halves must be aligned with each other as well as with the shaft.

Axial oscillations caused by misalignment may result in formation of a groove around the shaft due to abrasion with the secondary seal.

End Play [9]

End play will cause excessive axial motion of the secondary seal similar to misalignment. Again shaft grooving and abrasion will result, especially if dirt is present. Leakage ultimately results. An O-ring secondary seal can tolerate some end play, and a bellows is insensitive to it.

End play can be prevented through use of an adequate thrust bearing such as a double-row ball-thrust bearing.

Distortion

Seal distortion and breakage may result from excessive or uneven mechanical stresses applied by fasteners (bolts, etc.), shrink fits, clamp joints and other mechanical means for holding the seal parts in place. Carbon is easily distorted or broken and must be handled carefully.

Imperfections [9]

Burrs, nicks and scratches on a shaft may cause trouble by gouging the secondary seal during installation. To avoid damage, leakage and subsequent replacement of the secondary seal, careful handling and minute inspection of the parts during installation is necessary.

Corners which O-rings are exposed to should be chamferred to avoid. Cut O-rings may go unnoticed and cause leaks that are not self-correcting as as a face that "wears-in" might be.

2. Friction

Introduction

The friction between seal faces is important since it limits the maximum pressure and speed at which a seal can operate. Friction results in a power loss to be dissipated in the form of heat. Thus friction resulting from the face materials in contact (dry or boundary lubrication), or from the friction in a fluid film, must be considered depending upon the lubrication regimes in a seal. An improvement in various friction and wear properties is directly translatable into improved seal performance. Practice and knowledge relating to materials for boundary lubrication is comparatively well established.

One expects friction in a seal to follow the trends found in other bearing elements, where different lubricating regimes exist depending upon conditions. These regimes are customarily termed boundary lubrication, mixed film lubrication, and full film lubrication. Different coefficients of friction are found in each regime. This is sketched in Figure 12 versus the "duty parameter" (ZN/P).

Boon, Honingh and Van Rijssen [15] state that they have tested may seals. They consider Figure 13 to be typical of the frictional characteristics of the seals they have tested when sealing a liquid. Figure gives dry friction for the same seal. It is quite different in character. The "wet" seal friction in Figure 13 initially decreases with load, then increases greatly as the load is further increased. The friction values obtained are representative of the transition between full film, mixed and boundary lubrication. The fact that lower friction values occur at about $\eta = 0.5$ to 0.8 ($\eta = 0.7$ for $\beta = 0.3$ balance), and that higher friction is found at $\eta = 1.0$ ($\eta = 1$ for an unbalanced seal, i.e., $\beta = 0$), ties in nicely with the need to use a balanced seal when sealing against high pressures. This points out that balancing a seal decreases friction by changing dry friction to lubrication with a liquid film.

If a fluid film has been obtained by balancing, the gap width should increase. This is apparently confirmed since the authors note: "For $\eta = 0.5$ the friction is quite low, but the seal leaks a little." The implication is that the improved ability of a balanced seal to handle higher pressure and obtain a lower effective coefficient of friction is attained at the cost of leakage, and that this leakage then brings in to play the following effects:

1. Fluid pressure between faces due to leakage.

2. Hydrodynamic bearing action, because of the presence of liquid, to enable higher unit loads to be borne by the seal faces.

Fogg, in tests on a parallel thrust bearing [16], obtained data given a relation between the coefficient of friction (f) and high values of (ZN/P) in the full fluid regime of

$$f = c\ (ZN/P)^{1/2}$$

where c is a constant.

It is possible to show that the exponent for (ZN/P) should be approximately 1/2 if the hydrodynamic pressure generating mechanism is reducible to the Reynolds equation without side leakage; i.e., if pressure generation were due primarily to fluid viscosity acting in a converging clearance. This relation should hold for large values of (ZN/P) when the ratio of clearance to variations in surface height is large. As the bearing surfaces approach at low valuesof (ZN/P), the exponent should decrease.

Nau [17, 18] has compared face clearance and friction/duty parameter data from tests of various seal, and seallike geometries. He notes that in some cases [19] and [20] the exponent is one-half, whereas in others [21] and [22] an entirely different relation holds. Thus it would appear that different mechanisms have been observed to act in the fluid lubrication regime.

P V Relations

It is often customary to characterize the load carrying ability of materials in sliding contact by using a "P V" value. This has been applied to dry or boundary lubrication, as well as fluid film lubrication. For dry rubbing, assuming that the coefficient of friction (f) remains constant, the heat release rate is then proportional to the product of the pressure applied to the faces (P), and the sliding velocity (V). The transfer of heat away from the rubbing faces then determines the temperature increase at the faces. If the temperature is high, it can cause the surfaces to fail or wear rapidly.

In a seal P is proportional to P_p, the pressure being sealed. Thus the friction between surfaces directly determines the maximum pressure and speed that a seal can function at. For an unbalanced seal, P equals or exceeds P_p. For a balanced seal P is less than P_p. When using material combinations with a low coefficient of friction and good heat transfer, the upper P V limit for an unbalanced seal has been quoted as being about 200,000;

with the limit for a balanced seal at about 2,250,000 (23). (These P V values are in psi-ft/min.) Again, this checks the presumption that a balanced seal possesses a fluid film. That is, the heat released between faces with a liqui film should be an order of magnitude less than when boundary lubrication and physical contact occurs.

Materials for Boundary Lubrication

A high P V value can be obtained in boundary lubrication by:

1. A low coefficient of friction.
2. A high conductivity heat transfer path; or by providing coolin;
3. Materials capable of operating at high temperatures.
4. Materials not prone to wear.

Mayer, for instance, believes that it is possible to build unbalanced seals to reach 1000 psi, or increase speed limitations by 10:1; provided proper "friction" materials are used, and good overall heat transfer is obtained (4).

Various material combinations are tabulated in Table 1 to illustrate the effects of friction (f) and conductivity (k). To provide a means of comparison, it is assumed that two plates of equal thickness, one of each material, are in contact and rubbed together under a constant unit load. Then $f/(K_1 + K_2)$ is proportional to the temperature rise at the rubbing face, if heat transfer away from the external faces is neglected by assuming a constant temperature at the external faces. A low value of $f/(K_1 + K_2)$ is of course desirable. To achieve this in a seal, it is typical to use a relatively soft material bearing against a hard material. Thus PTF[1]/steel, carbon-graphite/steel, or graphite/CO-Cr-W alloy, carbon/ tungsten carbide combinations are frequently used. Metallic additives are, however, often added to PTF or carbon to increase thermal conductivity.

As a point of interest, steel/steel is compared with brass/steel in Table 1 to show the effect of the better conductivity of the brass. Graphite/graphite shows a potential gain because of a low coefficient of friction and high conductivity. Glass/glass shows the bad effects of a high coefficient of friction and the poor conductivity typical of ceramics. However, ceramics are used extensively as seal materials because they are able to withstand the high temperatures they generate.

PTF[1] = polytetrafluoroethylene compound

Table 2 - FRICTION AND THERMAL CONDUCTIVITY IN SEAL MATERIALS

Materials Couple Material [23] /Material [13]	Lubricant (Dry or Wet With Water)	Coefficient of Friction f	Thermal Conductivities K_1/K_2 BTU/HR-FT-°F	$\frac{f}{K_1+K_2}$ X 10^4	$\frac{f}{K_1}$ X 10^4	REF.
A - Common Materials						
Steel/Steel	D	0.4	26/26	77		
Brass/Steel	D	0.35	60/26	47		
Graphite/Graphite	D	0.1	80/80	6.25		67
Glass/Glass	D	1.0	0.6/0.6	8330		67
B - Carbon-Graphite/Steel	D	0.12-0.20[a]	10.9/26	32.5-54		87
	W	0.05-0.08[a]	10.9/26	13.5-21.7		87
C - PTF/Steel PTF	D	0.12[a]	0.14/26	53.6	8600	88
PTF+15% Graphite (load=2psi V=20fpm)	D	0.12[a]	0.26/26	53.3	4600	88
PTF (load = 1000 psi)	D	0.026[a]	0.14/26	9.9	1860	89
D - High Temperature Ceramic (Al_2O_3 + Cr + Mo)/SiC	D	0.65	209/290	13.0		90, 91,92

[a]Indicates dynamic friction coefficients

TABLE 3 - VISCOSITY AND HEAT TRANSFER PROPERTIES OF LIQUIDS

	μ CENTI-POISE	SP. GR.	BOILING POINT °F	K_1 BTU/HR-FT-F	$\frac{u}{K_1}$
Water	1	1.0	212	.343	3.32
Hydrocarbon oils					
Kerosene	1	.8	350/525	.0693	1.87
Light Oil	25	.8		.07	35.
Liquid gases					
O_2	.19	1.14	− 297.5	.0867	2.19
H_2	0.013	1.07	− 423	.0693	1.87

Variations in Friction Under Conditions of Use

The coefficient of friction for two given materials in contact may vary with speed and load. The coefficient of friction usually decreases with increasing speed. The friction coefficient also decreases with increasing load for the softer, more readily deformable materials. Hence, the softer material of a seal may be given a narrow face to reduce the friction heat generated [23, 4]. See Figure 14 for friction versus load for a carbon, and Figure 15 for a PTF.

Materials at Low Temperatures

In sealing liquified gases, the thermal conductivities of seal materials may be radically different than at "ordinary" temperatures. In some cases thermal conductivities are as much as 100:1 lower. Copper is different than other materials; its conductivity increases approximately 10:1. Thermal conductivity data to illustrate this is given in Figure 16.

Not many frictional measurements have been made at cryogenic temperatures. However, available test data implies that the coefficient of friction is not greatly affected by temperature [24, 25].

Heating in a Fluid Film

Heat generation in a fluid film also limits the performance of a seal should excessive evaporation or boiling of the liquid result. Then the viscosity and heat conductivity of the liquid become important primary variables as well as heat transfer through the seal faces. Conversion of the liquid to gas, so that part of the faces are no longer separated or lubricated by liquid should reduce the permissible face loading. Also it is noted that seals used with boiling liquids tend to be erratic in performance. The liquid flashes into vapor, causing the seal to vibrate and leak [23].

For a given face temperature, the maximum increase in temperature in a liquid film bounded by parallel faces is a function only of face velocity (V), fluid viscosity (μ), and the thermal conductivity of the fluid (K_1). Assuming constant fluid properties, the maximum temperature obtained is not affected by the spacing between seal faces. The maximum temperature increase (Δt) for heat conducted equally through both faces, neglecting leakage flow and end effects is

$$\Delta t = \frac{\mu V^2}{8K_1 J}$$

where J is the mechanical equivalent of heat.

The temperature increase in the liquid between seal faces was calculated for a few liquids including the case of one face non-conducting. The results are plotted in Figure 17 as a function of face velocity. The temperature differences for the two cases differ by a factor of 4. An actual seal should fall between these two extremes. Appropriate viscosities (μ) and conductivities (K_1) are given in Table 3. A relative ranking of the various liquids is given as μ/K_1; i.e., the ratio of heat generation to conductivity.

3. Wear

The rate at which a seal wears determines its life. Failure need not necessarily be catastrophic, but can be slow rate of wear over the lifetime of the seal. Wear can smooth a rough seal face to make it useable, also conceivably roughen a smooth surface to make it capable of hydrodynamic load bearing.

Roughness: Summers-Smith's Data

Summers-Smith [19] has reported on the effects of surface roughness on seal friction and wear. He wanted to determine the limiting conditions of operation versus life and leakage for typical seal construction. The fluid sealed was water, and/or water/glycerine mixtures. This provided as easy way to vary viscosity (0.9 to 1350 cp at 20C), yet use a liquid processing poor "lubricity" under boundary lubrication conditions. This undoubtedly made the effects of surface roughness more critical and obvious. Pressures sealed were from 0 to 50 psi; speeds 500 to 2800 rpm. The leakage path was radially inwards.

The rate of wear was measured versus duty parameter (Figure 18a), using seals wherein the roughness of the hard face was in the range of 2-6 micro-in. c.l.a.[2] such as would be expected in a commercial seal. At $\frac{\mu\omega}{2\pi P_p}$ values of 5×10^{-8} and above, wear rates of carbon and PTF faces were low, ranging from 1 to 4 micro inches per hour. This an acceptable rate of wear for a seal for long life. At low $\frac{\mu\omega}{2\pi P_p}$ values below 5×10^{-8} very high wear rates were obtained. This implies that fluid films tended to exist at high values of the duty parameter, and contact and wear at low values. A plot of the coefficient of friction versus duty parameter substantiates this (Figure 18b), giving the pattern of mixed lubrication and its transition to a full fluid film.

To obtain the effects of surface roughness on friction, rough surfaces (16-20, and 30 micro-in.) were run at low values of the duty parameter.

As one might expect, this increased the coefficient of friction (Figure 18a). The important conclusion is that surface roughness of 10 micro-in. and less produce desirably low coefficients in the order of 0.01 to 0.05 when in the mixed lubrication regime.

The roughened surfaces were run for quite some time. The surfaces tended to "break-in" and the surface roughness decreased (Figure 18b). Roughness then remained constant at from 0.5 to 12 micro-in., depending upon the initial surface roughness. This means that a rough or damaged surface has a good chance of "healing". Also, provided that initial break-in-periods are permitted, relatively high surface roughnesses can be tolerated. Thus, exceedingly fine initial finishes are not needed.

Others have also noted this in seals [4]. Coffin [26] has shown that break-in periods with light loads can in certain cases enable parallel thrust bearings to subsequently sustain high loadings. Somewhat similar results have been obtained for journal bearings [27]. They studied the effects of the surface roughness of steel shafts rotating at various loads and speeds in a journal made of Babbit. When operated in the mixed lubrication regime, wear decreased from an initially high value to cease after a break-in period.

Materials: Mayer's Data

Additional information on friction and wear for unbalanced seals is given [4] for a surface speed of 1000 ft/min, covering a pressure range up to 250 psi. This information is reproduced in Figure 20. Companion information on the coefficient of friction is given in Figure 20. This clearly shows that the maximum loading on an unbalanced seal is determined by the rate of wear deemed permissible. Schoenherr also gives wear rates versus load [13].

Debris in Fluid

Probably the wear obtained at high duty parameters (Figure 18a) was caused by debris in the liquid being sealed. The presence of excessive quantities of abrasive debris can result in short seal life, unless removed by some auxillary device or flushed out by clean fluid injected at the seal.

The solid contaminants permitted in "clean" rocket propulsion fluids and clean aerospace hydraulic fluids range from 20 particles per liter to zero [28]. This is low compared to the contaminants found in typical "unused" oils, fuels, hydraulic fluids, etc. Closed circulating systems containing pumps, gears, and bearings, unless specially filtering precautions are taken, are often exceedingly "dirty" and can be expected to have many thousands of particles per liter due to the normal wear of these components. Thus it would

not be unreasonable to expect coefficients of friction and wear to be affected by the debris present in a specific fluid or system being sealed.

In extensive studies on slide valves [29], wherein the slide valves possessed small clearances, relative motion between mating surfaces, and leakage between the surfaces; it was found that particles approximately equal to the clearance dimensions caused rapid wear if the walls forming the clearance space moved in and out to "squeeze" the particles. Debris larger than the clearance could not enter to affect friction and wear. However, it was found that large radii on the inlet side of the leakage clearance acted to break down large particles so that they could then enter and increase friction. A seal undoubtedly possesses eccentric and angular misalignment to produce relative motion between faces, and so should be subject to this type of wear. For instance, Denny [21] found wear on the carbon face on the inlet side of the leakage path while maintaining a fluid film between faces.

Production of Surface Roughness (Micropads)

Originally the Battelle Memorial Institute noted that many surface asperities were to be found on the carbon faces of seals, and that these asperities were probably capable of moderate load bearing [30]. These asperities have been termed "micropads".

Recently, used carbon face seals taken from sealing propeller shafts in marine service were examined [31]. They reported that about 30% to 40% of the worn area of the carbon face was covered with micropads of a somewhat different type. These micropads were not present in new seals. The resultant loadbearing has not been determined. Also, the means by which wear occurs to produce micropads has not yet been investigated. This is an interesting aspect that undoubtably will be pursued.

Effect of Fluid on Seal [32, 11]

The fluid being sealed has two effects on the seal: (1) as a lubricant and, (2) as a corrodant.

Acting as a lubricant, the fluid film between the primary sealing elements plays a vital role in successful operation. If the fluid film cannot support loads well, i.e. cannot serve as a good lubricant, the face materials themselves have to bear the contact stress. This condition results in shortened life.

Corrosive action by the fluid on the seal elements or the system in which the seal is located causes a number of problems. Materials must be selected with care in order to prevent easy seal failure. Attention must be paid to the possibilities of galvanic, intergranular and stress cracking corrosion for those parts subjected to cold working without annealing or high stresses, such as springs. Corrosion of the system by the sealed fluid could lead to deposits or suspended oxides which are generally abrasive in nature and will cause rapid wearing of the faces and early seal failure. Also, <u>the selection of the secondary seal is mainly dependent on the sealed fluids chemical aggressiveness</u>.

<u>Dry operation</u> is a common cause of seal failure. A seal may run dry due to flashing of the fluid at the seal interface, vacuum condition in the seal chamber, or especially in a vertical shaft arrangement, because proper venting of gas or air is lacking. Flashing commonly occurs when sealing a liquid that is being pumped near its boiling point. The additional frictional heat developed by the seal may quickly boil the liquid out of the seal chamber unless the temperature is controlled by a jacket, by flushing with a cool pumpage, or another liquid not near its boiling point. Recirculation or flushing from an external source can also build up the pressure in the seal chamber when operating at a vacuum. Otherwise, air may be drawn across the seal faces which will then run dry.

If it is possible for a seal to run dry, or the seal is sealing a vapor space, a double seal with seal fluid for lubrication and cooling, or a lubricated (grooved face) single seal may be used. There are gas seals available that are designed for continuous dry operation. They are carefully balanced and are usually more exotic in design and, thus, more expensive.

<u>Abrasives (Dirt)</u> [9, 11]

Dirt, or solids in the liquid to be sealed, can be a troublesome hazard to seal operation.

This problem can be summarized as follows: (a) trouble caused, (b) source or type and size of abrasive, and (c) solution to the problem.

(a) <u>Problem</u>:

(1.) The very fine particles may wedge between the seal faces and cause wear and leakage due to their abrasive cutting action. Since the primary seal faces move relative to each other at clearance from 40 to 180 microinches, particles of this size (1 to 5 microns) can get between them.

(2.) The particles may deposit in the seal chamber and around the seal parts, eventually interfering with movement and cooling of the parts. External seals using an O-ring or V-ring have a cavity in the rotary element that may act as a centrifuge and gradually fill with solids preventing proper seal function.

(b) <u>Source of Abrasive</u>

(1.) Inherent: The abrasive is inherent in the liquid like sandy water and slurries.

(2.) Thermal: Certain liquids form crystals when subjected to thermal conditions. Ammonium nitrate, for instance, will form crystals that are abrasive at the faces if proper temperature is not maintained to keep them in solution.

(3.) Atmospheric contact: Some liquids (like sugar solutions) when evaporating leave crystals. Another example is water treated with chromates which when seeping through the faces vaporizes in the atmosphere leaving behind an abrasive residue.

(4.) Stagnation and Degradation: Over a period of time solids could be formed under these conditions.

(c) <u>Solution to Problem</u>

(1.) Inherent: Keep the abrasives away from the seal faces by injecting a clean neutral liquid over the seal faces, or use double seals.

Injection will result in the clean fluid intermixing with, and diluting, the process fluid being sealed.

In order to reduce this intermixing, a throttle bushing is placed in the seal chamber throat. Thus the injected fluid is supplied at a pressure above that of the process and the flow rate is controlled by the throttle bushing (Figure 21).

The clean fluid for injection may be obtained: (a) from and outside source, (b) by filtering the process fluid if the solids are small in size and quantity, or (c) by using a cyclone separator on the recirculation line if the solids are free-flowing, heavier than the liquid and at least 30 psi pressure differential is available.

Cyclone or centrifugal separators are very effective within certain limitations, such as low viscosity fluids, and solids considerably heavier than the liquid. To remove the fine particles the cyclone must be small in order to have a sufficient pressure differential at low flow. Cyclone separators are rated as to the fraction of particles of a given size and density that are removed for a specified pressure differential. The following are a few typical ratings:

particle size microns	% removal	density ratio	pressure differential, psi	viscosity
8	95	2.7/1	30	water
5	"	"	40	"
3.3	"	"	200	"
3.1	"	5.4/1	40	"
20	"	2.7/1	40	No.5 fuel o

Cyclone separators remove abrasive particles that are inherent or suspended in the liquid and are not effective on thermal or atmospheric abrasive conditions. Adequate flow to the seal, 3 gpm or more, should be allowed. The small orifice holes in the separator may become plugged by the particles and should be checked periodically.

Double seals may be used if no dilution is permissible.

(2.) Thermal: If this condition exists then the temperature should be kept high enough to prevent crystals from forming. This may be accomplished by using a heated water jacket, steam or electrical heating elements. It is desirable, however, to operate the seal as cool as possible to slow down the corrosion rate and maintain material integrity.

(3.) Atmospheric Contact: To remedy this condition the atmosphere is kept away from the seal by creating a buffer zone through use of either a quench gland, tandem seal or double seal arrangement. (See Figure 21)

(4.) Stagnation and Degradation: A simple recirculation to keep the fluid fresh in the seal area is satisfactory.

Speed [11]

For a given face pressure and coefficient of friction, the amount of heat generated at the seal faces is directly dependent on the relative velocity between the surfaces. As the velocity increases the rate of heat input to the faces (power loss) increases and consequently the temperature rises at the faces until a balance is established. The rise in temperature affects the mechanical and physical properties of the face material and the fluid film, reducing the ability of both to support loads.

The heat generated at the seal surfaces, i.e. the power loss is

$$\text{Power} = \text{Torque} \times \text{speed}$$
$$= \left[(F_p+F_s)\ f\ \frac{\bar{d}}{2},\ \text{ft-lb}\right] \times (2\pi\ N,\ \text{rad/min})$$

where

$(F_p + F_s)$ = Face Load, lb

f = Coefficient of Friction (depends upon lubrication regime)

$\bar{d}$ = Average Seal Face Diameter, ft

N = Shaft Speed, rpm.

Thermal Effects [33, 11, 34]

(a) Temperature Level: Temperature conditions at the seal faces can be several hundred degrees above the ambient due to friction at the rubbing surfaces. These conditions may dictate the success or failure of the seal. This temperature should be considered when designing and selecting materials, especially when the fluids handled are prone to flashing or boiling due to the increased temperature. The engineer should have complete information on the liquid to be sealed in order to avoid failure from this cause.

The mechanical and physical properties of both the materials of construction and the sealed fluid are changed by temperature. The combined effects on materials and fluids due to increased temperature tend to reduce the ability of the seal to withstand pressure and velocity. In the broad range of applications where face seals are used, there are many instances of temperatures exceeding the useful limits of elastomers, plastics, and some metals. Cryogenic temperatures can be as much a problem as high temperatures.

When conditions of high temperature and limited lubrication exist, heat checking of the metal faces may occur. This may be overcome by using materials less prone to checking but still provide good wearing surfaces. Materials with high thermal conductivities are good. Examples of face materials commonly used are:

Tungsten Carbide: Good thermal conductivity, high resistance to heat checking. A good choice.

Stellite: Very susceptible to heat checking. Make a good mating surface for carbon when properly used and where adequate lubrication is available. It is equal to most stainless steels in corrosion resistance and finishes to a smooth hard surface.

Cast Iron: Good face material but more susceptible to corrosion and does not have as high a thermal conductivity as tungsten carbide.

Ni-resist: Same as cast iron.

Flushing or cooling the seal faces with flow from a by-pass line will usually improve the high temperature condition. A positive flush of the seal area is usually desired.

(b) Thermal Shock: Rapid changes in temperature may produce high temperature gradients with resultant thermal stresses. Materials having low thermal conductivities, low tensile strength, high coefficient of thermal expansion and high modulus of elasticity are particularly susceptible.

(c) Thermal Distortion: Friction at the seal faces tends to make them hotter than the rest of the seal. Since the seal is essentially a tube with a heat source at one end and cooling along its length and at the opposite end, a temperature gradient is established axially along its length. The tube diameter is larger, therefore, at the hotter end. The size differential is proportional to the temperature difference, seal diameter and coefficient of thermal expansion.

This growth occurs after the ring has been lapped and as a result the faces are no longer parallel. This growth effect is called "coning" alters the pressure gradient across the faces and can change the seal balance so that during the first start up the seal tends to blow open.

This critical balance phenomenon happens only after the first start-up of the seal. If the seal passes through this break-in phase without excessive leakage, the face wears parallel to the mating surface at operating temperature. Thereafter the seal is balanced at operating conditions (parallel faces) and overbalanced (larger hydraulic closing force) during startup when the faces are no longer parallel.

This problem can be compensated for by cooling of the seal to reduce the temperature differential, or by calculating the amount of "coning" and tapering the seal nose enough to compensate for the distortion. The seal face then starts with contact at the outside diameter and maintains this con-

tact after thermal distortion, assuring an overbalance (for seal pressure on the O.D.) until the sealing surfaces wear in. This eliminates the over-balanced design otherwise needed to over come the initial tendency for the seal to blow open.

Chemical Effects [33]

When chemicals come in contact with materials, they may set up a chemical action that becomes a form of corrosion. This damage can cause loss of the seal or reduced life. It is essential, therefore, to have complete knowledge of the liquid being sealed. Some seal materials that are used are:

Carbon vs. ceramic: Good for acid service while attacked by alkalines.

Filled teflon: high corrosion resistance.

Tungsten carbide: " "

Boron carbide: " "

When considering materials for the wearing face of the seal, consideration must be given not only to corrosion resistance, but also how this resistance takes place. Chemical action on some metals form a coating that protects the base metal. If this coating is removed by surface friction, deeper corrosion can set in, increasing the face wear.

Radiation Effects [33]

Many materials such as elastomers, plastics and some carbons and ceramics are affected by gamma rays. Metals, in most cases, are not seriously affected except where cost or sintered metals contain fillers or binders that may be attacked.

Maintenance [9]

In general after a face seal is installed it does not require further attention or adjustment until complete replacement is necessary. The concept of package replacement is generally practiced.

Thorough training and understanding of the basic face seal principles is essential to assure recognition of causes for failure and the necessity to be very careful when handling and installing seals.

When a shaft is removed for any reason other than seal failure, it is not necessary to replace the primary seal faces. A light lapping or polishing will recondition them. The carbon can be reused until the wearing nose is used up. The seat wear is usually significantly less and it has a long life. Many seats have both sides lapped by the seal supplier and by turning it around it is ready for use again.

F. Seal Materials [33, 9]

There are three functional categories of seal components and each category has its own material selection problem. These categories are:

1. Hardware, consisting mainly of spring and retainer.
2. The secondary seals which may be of the pusher type (O-ring, V-ring, etc.) or the non-pusher bellows type.
3. The primary seal consisting of the seat and head (washer or nose piece).

Selection of Seal Material

With the appropriate knowledge of chemical and mechanical action, and material properites, it is possible to select the proper materials for best operating and most economical seal.

First consideration is the selection of the best wearing faces for the primary seal rings plus corrosion resistance.

Next the secondary sealing elements are chosen to provide long life considering the adverse conditions of corrosion, temperature, pressure, etc.

Metals should be carefully selected to avoid breakdown of the mechanical features built into the seal. Thin wall sections and small cross section springs will rapidly lose their effectiveness if corrosion is present.

1. Hardware; spring, retainer and miscellaneous hardware

The main consideration in the selection of materials for these components is corrosion. Handbooks on chemical compatability are available which will help narrow down the choice. It is important to remember that although the fluid being sealed may not be corrosive, contaminants or breakdown products of the fluid can be quite corrosive.

Some of the materials used for seal hardware are:

Brass: Common materials used for hardware where the conditions are not too severe.

Steel: " " "

Stainless Steel: " "

Nickel-Copper Alloys: Used for more severe corrosive conditions.

Nickel-Molybdenum Alloys: " "

Nickel-Molybdenum-Chromium Alloys: " "

Nickel-Chromium Alloys: " "

Titanium: Used for very severe corrosion problems. Technical difficulties and economics usually make it one of the last choices.

Tantalum: " "

Zirconium: " "

Springs require special attention to proper selection of the material because of their small cross-section coupled with the hazard of stress corrosion. For example, for springs subjected to chloride solutions, austenitic stainless steels should not be used.

2. Secondary Seals

The selection of the material for secondary seals is based mainly on corrosion or chemical attack by the sealed fluid, temperature and pressure.

Pusher Type Seals: Elastomers and plastics are commonly used. There must be some flexibility to allow the seal to compensate for normal tolerances, run-out, and wear. The designer must consider any chemical or mechanical action that would cause set or cold flow of the secondary seal element that no longer lets it perform properly when motion of the seal head takes place. Chemical attack may also cause the elastomer to swell resulting in a secondary seal hang up and an inoperative face seal.

Buna N: -40°F to 250°F Used where the fluid is non-aggressive.

Neoprene: -40°F to 250°F "

Butyl: -60°F to 200°F "

Fluorocarbons: -30°F to 450°F "

Silicone Elastomers: -80°F to 550° F Used for extremes of temperature where general fluid resistance is not needed.

Polytetrafluoroethylene (TFE, Teflon): -450°F to 500°F Best all around material for pusher type seals because of its temperature range. Excellent material in either the "V" ring or wedge form. Used where chemical attack is a problem. TFE is one of the most inert plastics available and has proven itself in every chemical environment with the exception of molten alkali metals and atomic radiation. It has been used successfully with liquid oxygen at -310°F and heat transfer oils at 500°F. TFE has a very low coefficient of friction and will not adhere to metals, making it ideal for use as a sliding seal element.

TFE alloyed with glass and asbestos: Proven useful.

Asbestos : Use of a special asbestos wedge has extended the upper limit for pusher type seals to 650°F. This material has proven successful in the refining industry.

Bellows Seals: Metal bellows used for high temperatures. Important over 450°F. The thin sections of a metal bellows are required to give a low spring rate so that the load on the seal faces will not be excessive. These thin sections must be protected from corrosion which can reduce the spring rate and produce high stresses. The high stresses developed at relatively low loads also limits the usefulness of metal bellows.

Austenitic Stainless Steels: Up to 700°F.

Precipitation Hardening Stainless Steels: 700° F to 900°F.

Inconel X: Above 900°F Exhibits low relaxation rates necessary to avoid permanent set and resultant loss of bellows load.

Hastelloy Alloys: Used in more corrosive conditions.

Carpenter 20 Cb: "

Titanium: Superior corrosion resistance. Protects the relatively thin sections of metal bellows.

Elastomers and Plastics (Bunas, neoprenes, vitons, and some silicones): May also be used in bellows configurations for fluids and temperatures within their range. Care must be taken in selection of materials since excessive swell or shrinkage can cause seal failure.

TFE: Machined to the form of a bellows it is used where the pressure is low and an extremely corrosive fluid is being handled.

Gaskets: Since the secondary seals on the stationary seal face are not required to move, any type of secondary seal used on the seal head can be used here as well as other types of gaskets including metal ones.

3. Primary Seal Rings

The seal head (washer or nose piece) and seat are the most important parts of the seal. The material selection for these parts is the most critical and complex. Improper choice of materials for one or both of these surfaces will substantially reduce the seal life and usefulness. The important factors in selecting materials for theprimary seal rings are:

1. Compatability: The mating surfaces must form a good friction couple, giving favorable wear qualities. That is, the seal rings must maintain their integrity at the interface under conditions of positive contact pressure and relative velocity. Selection

1. Compatability: (continued) of the best wearing faces for given operating conditions is the first consideration. It is common practice to run dissimilar material combinations for the seal faces. Seldom can a material be run against itself and accomplish the results obtained by using a careful choice of dissimilar materials. There are some exceptions to this. It is desirable to have at least one hard surface when choosing mating face materials.

2. Corrosion: The primary elements must resist corrosion or chemical attack by the sealed fluid, that might weaken them and deteriorate their wearing surfaces.

3. Strength: Materials must be chosen that will resist destruction by forces other than wear such as:

(a) Radial and axial pressure forces.

(b) Centrifugal forces which are a function of shaft speed.

(c) High temperature weakening of the material.

(d) Thermal distortion due to temperature gradients.

(e) Thermal shock due to rapid changes in temperature.

(f) Radiation damage resulting in a weakening of the material or dimensional changes.

Typical materials for the primary seal elements are:

a. Carbon-Graphite: Carbon-graphite itself or impregnated with resins, inorganic salts, polymers or metals such as lead, babbitt, antimony or silver is used extensively as one of the seal rings mated against various metal alloys, alumina ceramics and tungsten carbides

Carbon-graphite's unique combination of compatibility with a wide range of alloys, plus its good thermal conductivity, chemical inertness and temperature resistance accounts for it being the most widely used primary seal ring material.

1. Compatibility: Has an inherent quality of self lubrication not found in similar materials.

2. Corrosion: One of the most chemically inert materials.

3. Strength: Low tensile strength.
Low modulus of elasticity.
These factors are a disadvantage because of deflection problems.
In all cases the carbon is subject to wear and replacement.
Little difference if it is used as the rotating or stationary element.
Usually used as the simplest construction part.
Rapidly dissipates friction heat from its surfaces because it is black.

b. Alternates to Carbon-Graphite

Leaded Bronzes: Used where carbon's low tensile strength and elastic modulus are a disadvantage.

Nickel-Silvers: Have even higher strength properties than the bronzes.
Show promise where the pressure is high and the velocities are low.

Plastic filled with Graphite: For high production seals when the conditions are mild.

TFE, glass filled: Used integral with a virgin TFE bellows for severe corrosion conditions.

c. Mating Materials:

Metals: Should be carefully chosen for wear resistance and corrosion resistance.

Cast Iron: Mild conditions of pressure and velocity.
Little if any corrosiveness due to the fluid.

Ni-Resist: Same

Hardened Tool Steel: Same

Hardened Stainless Steel: Mild corrosion.

Stoody: same

Stellite: Same
Prove to heat checking when lubrication is limited.

Ceramics: Usually accepted as best for corrosion resistance

Alumina Ceramics: Appreciably corrosive fluid, high resistance to corrosion.
Moderate pressure and velocity.
Excellent wearing qualities.
Used either as a coating or a solid piece.
Extremely hard, next to diamond.
Relatively low tensile strength.
Low thermal conductivity.
High modulus of elasticity, brittle.
Poor thermal shock resistance.

High Purity Alumina Ceramic (99.5% Aluminum Oxide): Severe corrosive conditions.

Carbides: Excellent mating face.

Tungsten Carbide: Highest pressure and velocity conditions are best handled with carbon-graphite running against tungsten carbide.
Used under corrosive conditions.
Sometimes used where ceramics cannot be.
Also available in a high purity form which has exceptional corrosion resistance.
Thermal conductivity 8.3 times that of alumina ceramic.
Thermal conductivity 4.0 times that of tool steel.
" 7.5 Ni-resist.
High tensile strength.
Low coefficient of thermal expansion.
Good thermal shock resistance.

Future Materials and Environmental Conditions

New materials will be needed to extend the usefulness of mechanical face seals. New and extended applications and new process fluids are creating more stringent environmental conditions for seals.

Current seal limits are ranging up to 1200 psi, 8000 ft/min, and 700°F. Future applications probably will require operation at pressures up to 5000 psi, velocities to 20,000 ft/min and temperatures to 1000°F.

G. Hydrodynamic Pressure Generation

When a fluid is present in substantial quantities between the faces of a seal, it is desirable to create pressure within the film to:

1. Separate the faces, support load, and prevent physical contact.
2. Create internal pressures to oppose and prevent leakage, to make the seal "seal".

Table 4 lists various fluid film theories applicable to seals that consider pressure generation due to hydrodynamic action, i.e., relative motion between the faces and radial leakage flow between the faces.

Denny [21] in a series of experiments with a face seal like geometry, first reported face pressures many times higher than could be due to radial leakage flow, and also noted the existence of inward pumping of 45 to 75 psi against the pressure being sealed. As best as could be determined from capacitance and friction-torque measurements, there appeared to be a liquid film between the faces. The face pressure increased as the faces were placed closer together. The pumping changed whenever the seal faces were relapped or reassembled, implying that small differences in the face dimensions or alignment were probably responsible for the inward pumping. These findings were the forerunner of increasing interest in hydrodynamic effects in seals.

Table 4

FLUID FILM SEAL THEORIES

Assumptions, Analytical Procedures

Unless otherwise mentioned, the standard lubrication assumptions have been made: (1) Liquid exhibits no tensile strength, (2) No side leakage, (3) Viscosity constant, (4) Use of Reynolds' equation, (5) Inertia neglected.

Solution for	Configuration	Assumptions	Analytical Procedures	References
Non Parallel Surfaces (pressure generation due to relative motion)				
*Pressure Leakage	Angular face misalignment Sine wave	Side leakage	Power series perturbation technique	39
Pressure Friction	Surface waviness Simple wedge		DuBois-Ocvirk short bearing approximations	20
*Pressure	Angular face misalignment Simple wedge		1 - short bearing approximation	40
		Radial and circumferential variations	2 - Finite difference equations, computer	
Parallel Faces (pressure generation due to relative motion)				
Surface Roughness				
Pressure Friction	Sine wave		Standard integration in closed form	41
Pressure	Symmetrical saw tooth	Includes tooth deformation	Order of magnitude analysis	42
Viscosity				
Pressure	Simple slider	Viscosity as f (temp.)	Graphical evaluation	43
Compressibility				
*Pressure	Angular face misalignment Sine wave	Compressible fluid	Linearization, power series perturbation	42,44
Grooves, slots				
Pressure	Spiral slots in face	Compressible fluid		45, 46
Clearance varies in radial direction, constant in tangential. Face pressure produced by pressure sealed against, less pressure loss due to laminar flow in radial direction.				
Leakage, Pressure, Clearance	**6 Contours: parallel, divergent, convergent, etc.	Axial and radial deformation proportional to pressure	Integrals in closed form	47
Leakage Pressure		Deformation proportional to pressure	Order of magnitude analysis	48

* Includes case of axial vibration.

** Also leakage for sinusoidal variation in tangential direction.

Load Bearing

Parallel Faces

Fogg's experiments [16] on simple, parallel face thrust bearings showed that considerable load can be carried by flat faces without physical contact. The action of a "thermal wedge" was advanced to explain this action. The lubricant was presumed to heat and expand as a consequency of friction. Load bearing so produced has been approximately analyzed [35] and shown to be an orde of magnitude less than if a wedge type of thrust bearing were used. Currie, Brockley, Dvorak [36] have recently obtained a numerical solution using relaxation techniques and confirmed this.

For some configurations, thermal and pressure deformations of the bearing surface produce a convergent film to provide load bearing [37, 38]. Viscosity variations as a function of temperature as proposed [43] may be of importance; however, a decrease in load bearing was obtained for a simple inclined slider bearing in a later paper [8].

Surface Waviness (Parallel Surfaces)

Surface roughness produced by machining was investigated [41], and shown to be an important pressure producing mechanism. Salama used a sinusoidal surface profile on one bearing face as a model, and assuming cavitation, obtained expressions for the load bearing capacity and friction between parallel faces. Cavitation is assumed so that only positive pressure are created between the seal faces. Negative pressures, if they existed, would be symmetrical and equal (on the average) to positive pressures, thus resulting in a net film pressure of zero.

A check with experiment was obtained by testing with different profile amplitudes (H) and different distances (L) between successive projections. In much of the range, friction checked closely with the assumption that the fluid film is discontinuous. But, beyond a certain H/L value, Salama found a rapid decrease in load bearing capacity compared to theory. Friction also increased. From Salama's plots, it appears that the mixed film region had been entered. Salama suggested that oil was squeezed out between successive projections. The significance of this appears to be that there is a limit to the amount of waviness that can be used, considering the fluid supply and seal dimensions.

Non-Parallel Surfaces

Lyman and Saibel [39] have analytically fitted Denny's data on leakage and pressure generation; assuming varying amounts of waviness; also misalignment, i.e., the rotating face is not normal to the axis of rotation. A fair match was obtained for leakage, face separation, and external face loading, assuming a waviness with face eccentricities of about 1.0 to 0.57 micro in. Correlation was not obtained between inward flow and misalignment. A basic inconsistency is that Denny concluded he had a full fluid film, whereas the existence of a cavitating, incomplete film was a fundamental postulate of Lyman and Saibel's derivation.

Nahavandi and Osterle [40] also evaluated the effects of face misalignment, using a different analytical technique than Lyman and Saibel. A numerical computer solution was obtained of load bearing capacity. Axial virbration increased or decreased load bearing ability depending upon the particular example.

Ishiwata and Hirabayashi [20] varied the waviness of seals from 40 to 210 micro-in. In extensive experiments, they found this to have considerable effect upon friction and the ability to seal. Increasing waviness prevented leakage, but also increased friction. Friction was always lower when leaking than when sealing. Ishiwata and Hirabayashi were able to plot their friction data as smooth curves versus surface waviness and the duty parameter $Z\mu/P$, and thus show that much of their data is in the hydrodynamic fluid film region.

In their analysis, Ishiwata and Hirabayashi assumed the waviness of the seal to be approximated by a series of wedges occurring as one of the seal faces. They assumed cavitation, and using the short bearing approximation, solved the Reynolds equation to obtain the pressure generated (P) as

$$P = \frac{\mu V b^2}{2 \pi D}\left(\frac{1}{h_1^2} - \frac{1}{h_0^2}\right)$$

where h_1 is the minimum clearance to the top of a wedge, h_0 the maximum clearance to the bottom of a wedge.

This relation shows that extremely high pressures are generated as seal faces approach. For example assume:

1" dia. seal, 1000 RPM, 0.1" seal width.

Kerosene (μ = 1 centipoise)

h_1 = 20 micro in.

h_0 = 40 micro in.

Then, p = 20 psi. But, for h_1 = 2 micro in., p = 3500 psi. Thus, changing the clearance in the radial direction should provide a means of controlling pressure

generation across the face of a seal, beyond the amount obtained from waviness or eccentricity.

However, an important point is that small variations of the order that Lyman and Saibel postulate for face eccentricity (1 micro in.) cannot be readily manufactured or controlled. However, variations of 10 micro in. and greater can be practicable, as for example in Ishiwata's and Hirabayashi's use of face waviness of from 10 to 210 micro in.

Micropads

The Battelle Memorial Institute [30] repeatedly photographed cavitation trails in the liquid film between the surfaces of face seals. One of the faces was made of glass to permit direct observation. The cavitation is generally in the form of long, thin striations; i.e., elongated voids separating thin channels of liquid. The void and liquid paths are concentric to the center of rotation, describe circular trails, and start at definite points where surface projections exist. Battelle has concentrated on the role of these projections in producing load bearing. Again the essential elements are in effect a fluid wedge to produce a positive pressure, and the invoking of cavitation to prevent a negative pressure. Load bearing in the order of 10 psi has been obtained for small micropads, 100 psi for larger micropads, dimensioned as follows.

To provide suitable micropads for study and improve load bearing, numerous micropads werephotoetched in a rectangular grid pattern on a seal face [49]. The micropads were no larger than 6×10^{-3} in. diameter by 100 micro in. high. Ideal, simplified analyses of load bearing were found to agree order-of-magnitude-wise with test results. Subsequently, tests on larger micropads 12×10^{-6} in. in diameter showed load bearing an order of magnitude larger than predicted [50]. It was postulated that this was due to a slight slant in the top surfaces of the micropads. Deliberate attempts were then made to manufacture a slanted micropad, resulting in further increases in load bearing.

Of interest is the finding of micropads on worn carbon surfaces, as previously mentioned [31]. These micropads were essentially small plateaus at the same level, set off by valleys of from 10 to as much as 100 to 300 micro-in. depth.

Surface Deformations

It is customary to consider surface deformations in rolling machine elements where high Hertzian stresses are developed. Since film pressures are comparatively low in seals, deformation should only be of importance when the seal faces consist of relatively light rings or when using face materials possessing a low modulus of elasticity, such as PTF or carbon. Davies [51, 52] has examined the case of a symmetrical saw-tooth type of surface roughness, and shows that moderate load bearing is possible if the saw-tooth is deformed by the fluid pressure generated. The resultant pressure distribution is not symmetrical. The importance of this is that a net pressure in the fluid film can be obtained without the need of cavitation.

Grooves, Slots

Grooves and slots have been used in the faces of seals for improved cooling; and to generate fluid pressure to maintain a full fluid film, thereby supporting load and minimizing friction. Mayer [4, 12] used grooves in face seals handling liquid to reach high speeds and loads. He gives empirical relations between the grooves he uses and seal friction. He finds that the number, size and location of the grooves have to be held to a fairly narrow specification for satisfactory results. Whitley and Williams [53, 54] have applied spiral grooves to gas face seals and thrust bearings, and compare experiment with theory, including thermal effects and face distortion. Spiral grooves have also been used for face seals handling liquid metals [55].

Fluid Compressibility

Taylor and Saffman [42] analyzed compressible flow between disks. They found that a centripetal head (inward pumping) exists as a function of misalignment or axial vibration. Pressure variations in an incompressible fluid film average out for a net pressure generation of zero. A compressible fluid produces a non-symmetrical pressure variation, thus yielding a net film pressure. This argument applies to any non-symmetrical variation, i.e., such as viscosity as a function of pressure [56]. Whiteman [44] made a similar analysis for annular face seals for a liquid assuming a viscous, slightly compressible fluid. The net pressure generation obtained for the conditions chosen by these authors is low, in the order of a few psi. Unless very high shaft speeds are used, the net centripetal head thereby available for sealing purposes is relatively low.

The original cause for Taylor and Saffman's investigation of fluid compressibility was that Reiner believed that "cross-stresses" might exist in many fluids. This would result in a centripetal head. Reiner constructed a vacuum pump operating on air to demonstrate this [57]. Air was pumped radially inwards, between the faces of a seal-like configuration. Taylor and Saffman demonstrated that this effect could be due to misalignment and

vibration in the machine. Reiner [58] thereupon proceeded to obtain a centripetal head using a liquid (toluene), attempting to eliminate the objections posed by Taylor and Saffman. This controversy, and the causes of the centripetal head, have not yet been resolved.

Non-Newtonian Fluids

Weissenberg [59] originally noted that some liquids produced a centripetal head when sheared between the faces of rotating disks. This has since become known as the Weissenberg effect. It is of interest because it might apply to seals in opposing outward radial leakage, and providing load bearing. The Weissenberg effect has been found in polymer solutions, soaps, sugar solutions, gels, etc. [60, 61]. The basic similarity is that all of these fluids possess molecules with high length to diameter ratios. The Weissenberg effect is not to be found with short, low molecular weight molecules. Tanner, [62] using an analytical fluid model proposed by Oldroyd, [63] concluded that the centripetal head should be negligible for kerosene.

Bernd [64], following the general approach developed by [65, 66] has tested polyisobutylene solutions in kerosene at high shear rates. Polyisobutylene is a long chain molecule used to confer desirable temperature/viscosity and shear/viscosity characteristics to an oil. Appreciable tensile stresses (100 to 1000 psi) were obtained parallel to the plane of shear in the liquid at shear rates of from 10^5 to 10^7 reciprocal seconds, such as can occur in seals. These tensile stresses should form hoop stresses in a seal-like geometry to produce sealing. However, the life of long chain molecules at such high shear rates is short; also the tensile stress thus obtained decreases as the clearance between faces becomes small. The indication is that some moderate sealing is obtained with a polymer fortified oil.

Leakage, Sealing

Leakage Data: Faces in Contact

Mayer, in tests on seals [4, 12], noted that the character of the leakage he observed was different for each of the lubrication regimes. The relation he obtained for leakage Q in unbalanced seals (boundary lubrication) is of considerable interest:

$$Q = \frac{\pi\ D\ P_s S h_r{}^2}{P_p{}^2}$$

where P_p is the external pressure applied to the seal face, h_r the surface roughness, P_s the pressure being sealed. Leakage was independent of viscosity. S was an "experimental variable" that increased as a function of speed alone, ultimately reaching a constant value. S, although independent of fluid tested, was 100 times greater for leakage outwards than for leakage inwards. Pressure measurements between the seal faces showed no fluid pressure. Leakage was also stated to be independent of seal width, seal widths were apparently varied by about 25%.

One is somewhat in a quandary to identify the type of flow taking place. Laminar flow for a continuous fluid takes the form:

$$Q = \frac{\pi h^3 DP_s}{12 h_\mu \ln(R_2/R_1)}$$

In order to explain why leakage was independent of viscosity, Mayer suggested that depressions in the surface (i.e., surface roughness) act as a "conveyor" to meter the flow. Also, the dependence of Q on h_r^2 (i.e., proportional to the cross sectional area of flow) might imply a frictionless flow process such as a nozzle.

Since boundary lubrication results in high temperatures being reached at each surface projection as physical contact is made between rubbing surfaces [67], it would appear worthwhile to consider two-phase gas/liquid or gas flow in seeking an explanation. A gas-dominated flow regime would be relatively insensitive to liquid viscosity. That is, the range of viscosities encountered when liquids are in gas form is at least an order of magnitude less than a comparable range for liquids. Also, it is to be noted that Goodman [68], in analyzing viscous leakage flow between stationary surfaces possessing surface roughness, assumed plastic deformation as the surfaces were pressed together, and obtained a computer solution for leakage following the general form:

$$Q = c \frac{1}{\overline{P}_p n}$$

where n went through values ranging from 0.3 to ∞ as the compressive force (σ) increased. Thus it is conceivable that Mayer's case may be reducible to an example of viscous flow, provided deformation of the surfaces under load is considered. The large difference in S between inward and outward flow may be a consequence of fluid pressure deforming or cocking the seal faces to produce a non-uniform spacing between part of the faces, or perhaps a non-uniform stress loading where the faces are in contact.

Deformations Changing Radial Clearance

If there is appreciable leakage flow along the radial direction, the net force of the fluid acting on the faces will be increased if the gap between

faces is larger on the high pressure side. Opening the gap on the low pressure side results in a decrease in the net force. This effect should be quite marked, since for a given leakage rate, $\frac{dp}{dr} \propto \frac{1}{h^3}$, where h is the gap width. The Britich Hydromechanics Research Association has analyzed various cases in several papers, and claim improved seal operation as a consequency [48, 69].

Snapp [47] and Cheng and Snapp [70] determined face loadings for specific seals using different face contours in the radial direction. The face pressure analyzed was produced by the pressure sealed against, less the pressure drop due to laminar flow in the radial direction. A numerical solution was given for pressure profiles include deformation. Significant effects occurred when sealing pressures were above approximately 800 psi. Radial deformation tended to offset the effect of axial deformation, showing the possibility of having one type of deformation compensate for another.

Centrifugal Pumping: Fluid Film

Centrifugal force acts in a high speed seal to oppose radially inward leakage. The essential parameters can be seen by examining the relation for laminar fluid flow between seal faces when centrifugal effects are included:

$$Q = \frac{\pi h^3}{6 \mu b \ln(R2/R_1)} \left[\frac{3\rho\omega^2}{20} (R_2^2 - R_1^1) + P_p \right]$$

TABLE 5 - SEALING PRESSURE DUE TO CENTRIFUGAL FORCE [2]

SPEED - RPM	Seal Dimensions $R_2 = 1.5''$, $R_1 = 1.25''$	Seal Dimensions $R_2 = 2''$, $R_1 = 1''$
12000	15.3 psi	88.8 psi
36000	137 psi	800 psi

Sneck [71] has augmented this equation to cover the case of angular misalignment. The condition for no leakage is when the bracketed term equals zero. The pressure sealed against is then:

$$P_p = 3/20 \quad (R_2^2 - R_1^2) \rho\omega^2$$

To seal against high pressures a large width (b), and a high speed (ω) are desirable. This is shown in Table 5 where the sealing pressure is computed for water.

Viscous Pumping: Fluid Film

Findlay [72], assuming a full fluid film, obtained a solution for viscous pumping in an eccentric seal that also possessed angular misalignment. The pumping thus implied could be either inward or outward, depending upon the phase angle between the eccentricity and the waviness. For the case of inward pumping the minimum eccentricity required to oppose outward leakage to obtain zero leakage occurred at a waviness of $\epsilon = 0.817$. For this waviness the pressure sealed against as a function of eccentricity was found to be:

$$P_p = \frac{3\mu\omega\ (R_2 - R_1)\ e}{2.45\ h^2}$$

Using this relation, Findlay estimated a 5 psi sealing pressure for a 0.001 in. eccentricity (e) in a seal having $R_1 = 1.75$ in., $R_2 = 1.25$ in., operating at 1000 rpm with a 350 centistoke liquid, assuming an average clearance between faces of $h = 0.001$ in.

Grooves

Except for a case involving liquid metals [55], no general attempt has been made to exploit spiral grooves in a face seal to create a radial pressure differential in a liquid to oppose leakage. Parallel work has been done with axial seals, using an advancing screw thread to create a pressure head [73, 74] along the length of the screw.

Cavitation

Many of the load bearing analyses are based on cavitation as a fundamental premise; most of the leakage relations presume a continuous, complete film. Also cavitation in a seal as photographed by different workers [75, 30, 76] is different in appearance. Thus one is brought to the questions: what is cavitation, under what conditions does cavitation occur, and how significant is it. At the present time these are really unanswered for the context of a seal.

In high pressure seals, cavitation should be less likely to occur in the portion of the seal biased at high pressure by the fluid being sealed. Thus there should be a limitation to the use of cavitation. Experiments on seals where cavitation has been postulated to explain the resultant effects have so far been done at relatively low pressures (0-30 psi). This includes the work of Summers-Smith on surface roughness [19] and Ishiwata and Hirabayashi on surface waviness [20].

Under certain conditions, centrifugal force and flow between the seal faces can promote cavitation. Suppose as in Figure 23, centrigual force were used to pump liquid through a seal possessing parallel faces, without an external pressure head applied to the seal. Then the internal pressure in the liquid film would drop below the ambient pressure. That is, the pressure increase in the radial direction due to centrifugal force is $\frac{dp}{dr} \alpha\ r\omega$; whereas the force resisting flow is essentially $\frac{dp}{dr} = c$, where c is a constant. This points out that an "excess" of centrifugal force prone to cavitate. Sneck [71] has solved for the onset of cavitation in a misaligned seal.

Liquids may on occasion exhibit moderate tensile strength. The usual analytical assumption is to assume no tensile strength. It is true that "normal" cavitation and film breakdown has been photographed as a common occurrence in journal bearings [77, 78]. However, Dowson [79] noted downstream displacement of the film rupture point with a slider bearing. That is, the liquid possessed some tensile strength to produce a negative pressure. The tensile strength obtained in water depends upon the gas nuclei content, and the geometry and properties of the solid surface in contact with the water [80]. It is important whether the solid itself possesses gas nucleation sites and/or bonds physically to the liquid. There should be marked differences in these surface properties in the solid materials used in seals. A seal with an extremely small clearance must filter out large gas nuclei, thus making for moderate amounts of tensil strength. Also, the inertia of the fluid surrounding a void limits the rate at which a cavitation void can grow [81]. This could conceivably be important in a high speed seal. As an example, when using acoustic transducers to induce cavitation, the stress level for the inception of cavitation increases above 20 kc/sec [82].

On the other hand, once cavitation has occurred, small gas nuclei remain [83]. That is, cavitation voids contract in a high pressure region. The vapor contained in the void condenses. However, non-condensible gases, such as air initially contained in dissolved form in the liquid that was originally vaporized or gas that has diffused from the liquid into the void, re-dissolve relatively slowly and so remain behind as free gas. Assuming conditions favorable for this action, it is conceivable that free gas so produced can influence the downstream pressure field and subsequent cavitation.

Surface Tension

Brkich [6], Jagger [84], and Summers-Smith [19] have considered surface tension as a means of sealing. As can be seen in Table 4, the maximum pressures that can be sealed by this means are in the 25 to 50 psi range for common fluids, assuming a 20 micro in. gap between seal faces. The pressure is inversely proportional to the gap between seal faces. That is:

$$P = \frac{2\sigma}{h \times 69000}$$

Where

σ = Surface tension-dynes/cm

P = Pressure-psi

h = Gap width-cm

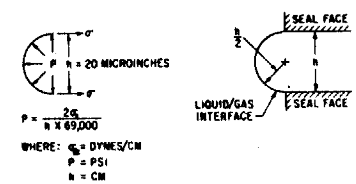
Jagger was able to positively demonstrate the existence of this effect by submerging lip seals in liquid. This eliminated the gas/liquid interface; consequently the seals no longer maintained a pressure differential to seal effectively. Lip seals are related to axial face seals; they possess a sealing lip pressing radially on the outside diameter of the shaft being sealed.

It has been argued that once a seal has leaked, thereby wetting the sides of the seal faces so that a suitably small meniscus does not form at the edge of the seal, sealing due to surface tension should cease. However, test data obtained by Summers-Smith has shown that surface tension can be important in a leaking seal. He decreased the surface tension of water by adding a surfactant to the water. Leakage increased, showing that a pressure differential due to surface tension was acting.

In order to maintain a meniscus of the smallest possible radius (h/2), and prevent spreading of the liquid; sharp 90° corners are desirable at the edge of a seal. In case a sharp edge were not readily obtainable, it should be possible to deposit PTE surfaces seal to prevent the spread of liquid on the seal periphery. Such coatings have been used by Campbell to prevent the leakage of oil from bearings, using the coating techniques he developed [85].

An interesting possibility postulated is that the cavitation trails observed by Battelle can be used to seal via hysteresis of the wetting angle [86]. That is, may trails occur side by side across the face. Given proper material/liquid combinations, the sum of the surface tension effects across the face resisting leakage flow should be appreciably larger than those given in Table 6.

TABLE 6—SURFACE TENSION—PRESSURE DEVELOPED AT A LIQUID/GAS INTERFACE [2]



	Surface Tension dynes/cm	Pressure P—psi
Liquid metals (Na, Li, K)	425/430	246/255
H_2O	72.0	41.7
Hydrocarbon Oils	20/30	15.4/17.4
Liquid Oxygen	13.2	7.6
Liquid Hydrogen	1.91	1.1

NOMENCLATURE

A_h	=	Nominal area of contact between seal faces
A_p	=	Seal area exposed to sealed pressure
B	=	Seal unbalance ratio
b	=	Seal balance parameter $(1-\beta)$
c	=	Constant
d_1	=	Inner diameter of seal face
d_2	=	Outer diameter of seal face
$\bar{d}$	=	Average seal diameter $(d_1+d_2)/2$
d_i	=	Inner bellows diameter
d_o	=	Outer bellows diameter
d_m	=	Mean effective bellows diameter
d_B	=	Balance diameter
F_B	=	Bellows force
F_f	=	Friction force
F_h	=	Force on seal face
F_p	=	Hydrostatic force
F_s	=	Spring force
f	=	Coefficient of friction
h	=	Mean clearance between seal faces
h_o	=	Maximum clearance
h_1	=	Minimum clearance
h_r	=	Surface roughness, peak to valley
J	=	Mechanical equivalent of heat
K	=	Thermal conductivity
N	=	Rotational speed, rpm
n	=	Constant

NOMENCLATURE (con't)

P	=	Pressure
P_o	=	Ambient pressure
P_p	=	Average face pressure
P_s	=	Sealed pressure
Q	=	Leakage
R_1	=	Inside radius of seal
R_2	=	Outside radius of seal
r	=	Radius
S	=	Variable
T	=	Temperature ^{o}R
Δt	=	Temperature rise
V	=	Face velocity
Z	=	Viscosity, centipoise
ZN/P	=	Duty parameter
$\mu V/2\pi P_p$	=	Dimensionless duty parameter
β	=	Seal balance parameter
σ	=	Surface tension
τ	=	Shear stress on seal faces
ρ	=	density
ω	=	Rotational velocity
μ	=	Viscosity, reyns

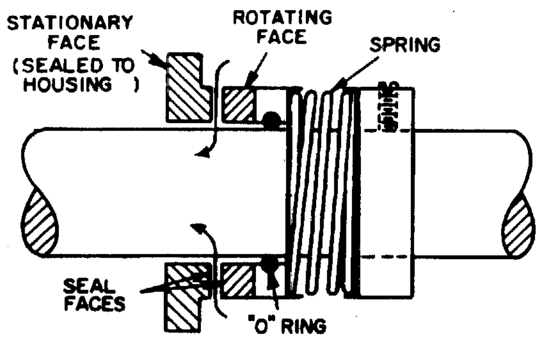


FIGURE 1 a - UNBALANCED SEAL-CROSS SECTION

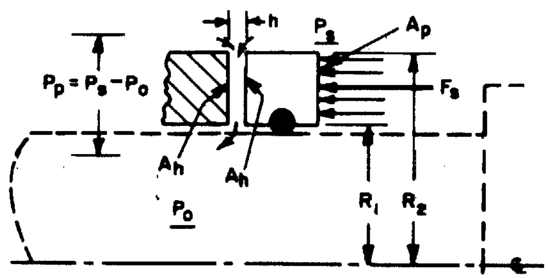


FIGURE 1 b - UNBALANCED SEAL-SCHEMATIC

h = CLEARANCE BETWEEN MATING SEAL FACES

F_s = SPRING FORCE

P_p = NET HYDROSTATIC PRESSURE OF FLUID BEING SEALED - psi ($P_s - P_o$)

P_o = EXTERNAL, AMBIENT PRESSURE

P_s = INTERNAL PRESSURE

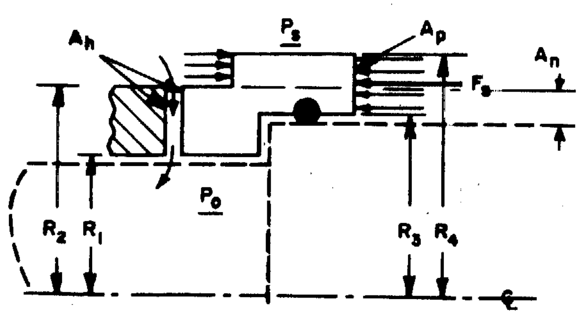


FIGURE 1 c - "BALANCED" SEAL - SCHEMATIC

FIGURE 1 TYPICAL SEALS

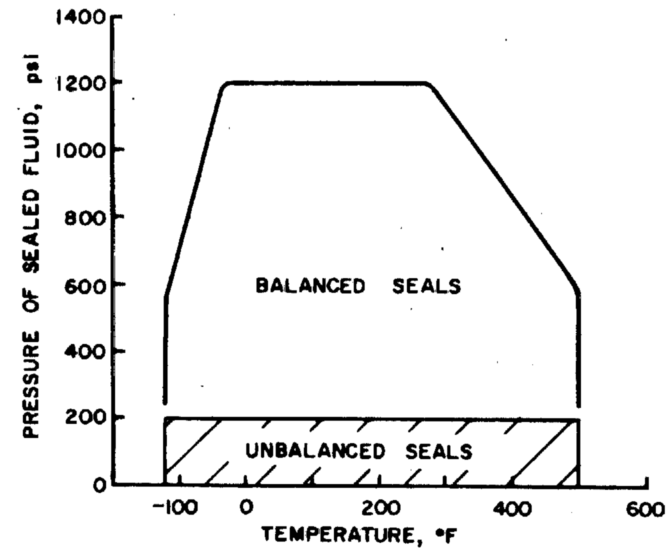


FIGURE 2 APPROXIMATE OUTLINE OF SEAL OPERATING CONDITIONS RECOMMENDED BY MANUFACTURERS

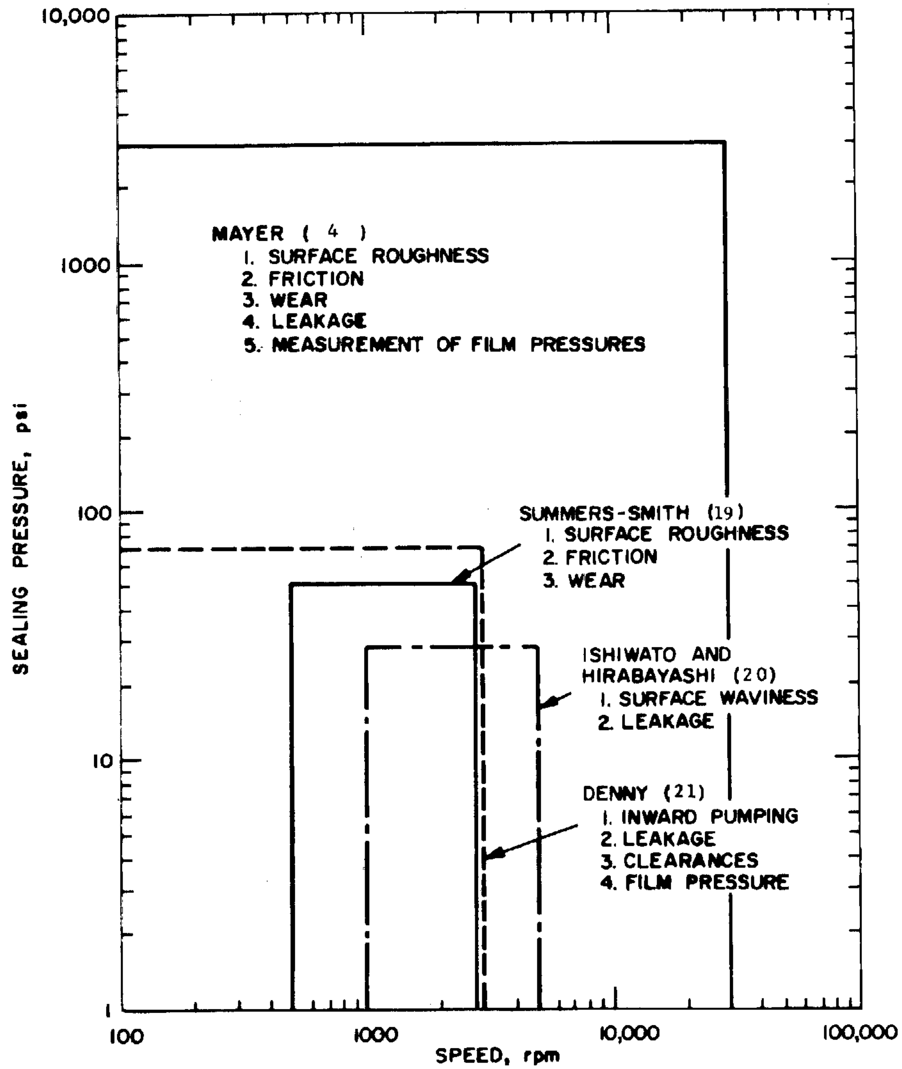


FIGURE 3 CONDITIONS OF SIGNIFICANT TEST INVESTIGATIONS OF SEAL PERFORMANCE (PRESSURE SEALED vs SPEED)

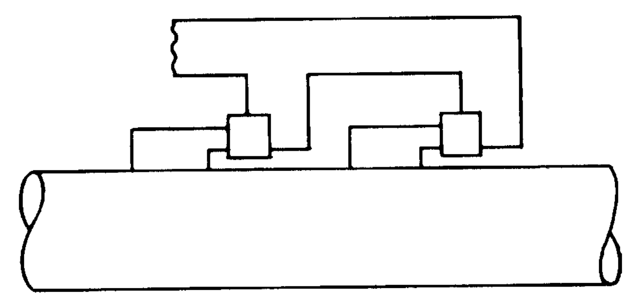


Figure 4

Tandem Seal Arrangement

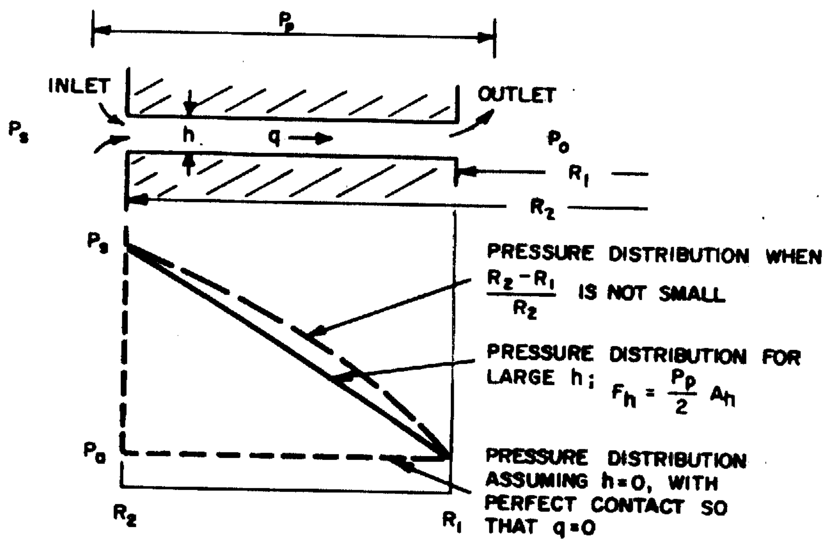
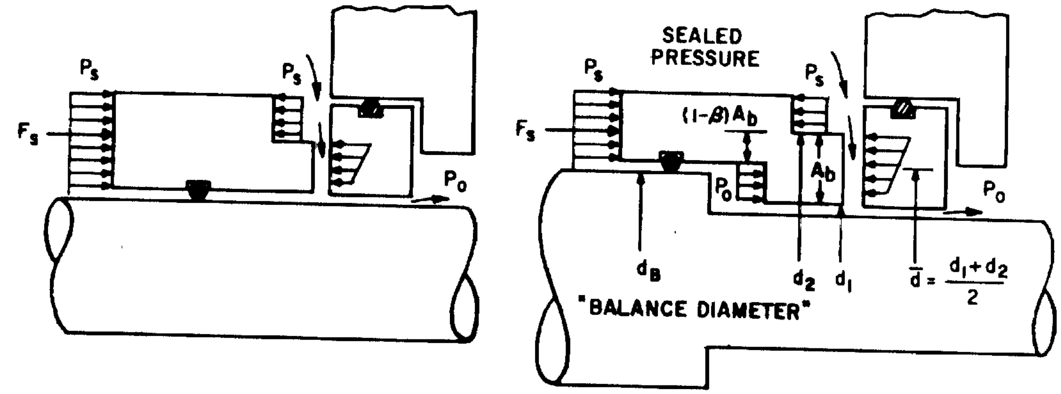


FIGURE 5 PRESSURE BETWEEN PARALLEL SEAL FACES DUE TO LEAKAGE

Figure 6
Rotating Head Seal

Pressure Loaded, Internal if $P_s > P_o$

Pressure Unloaded, External if $P_o > P_s$



(a) Unbalanced

$b = 1.0,\ B = .50,\ \beta = 0$

(b) Pressure Balanced

$b \approx .67,\ B \approx .17,\ \beta \approx .33$

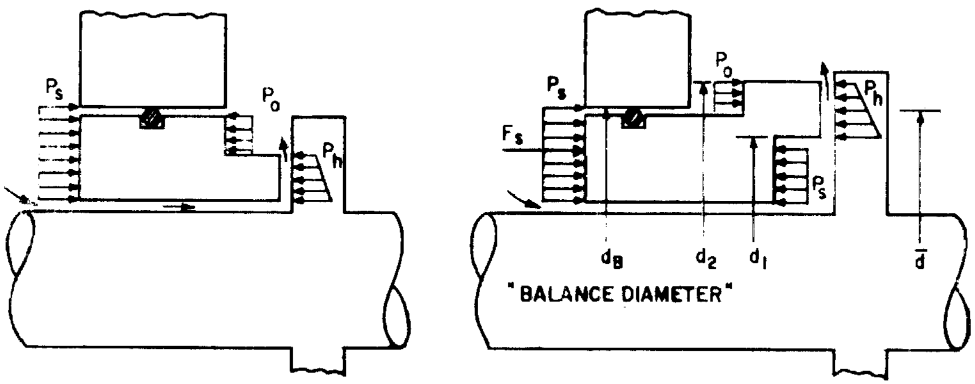
Seal Balance = $b = (1-\beta) \approx \frac{d_2 - d_b}{d_2 - d_1}$; $P_s > P_o$ (Sealed pressure on O.D.)

$\approx \frac{d_B - d_1}{d_2 - d_1}$; $P_o > P_s$ (Sealed pressure on I.D.)

Unbalanced Ratio = $B = (.5-\beta) \approx \frac{\bar{d} - d_B}{d_2 - d_1}$; $P_s > P_o$ (Pressure on O.D.)

$\approx \frac{d_B - \bar{d}}{d_2 - d_1}$; $P_o > P_s$ (Pressure on I.D.)

Figure 7
Stationary Head
Pressure Loaded, Internal if $P_s > P_o$
Pressure Unloaded, External if $P_o > P_s$



(a) **Unbalanced**
b=1.0, B=.50, β=0

(b) Pressure **Balanced**
$b \approx .67$, $B \approx .17$, $\beta \approx .33$

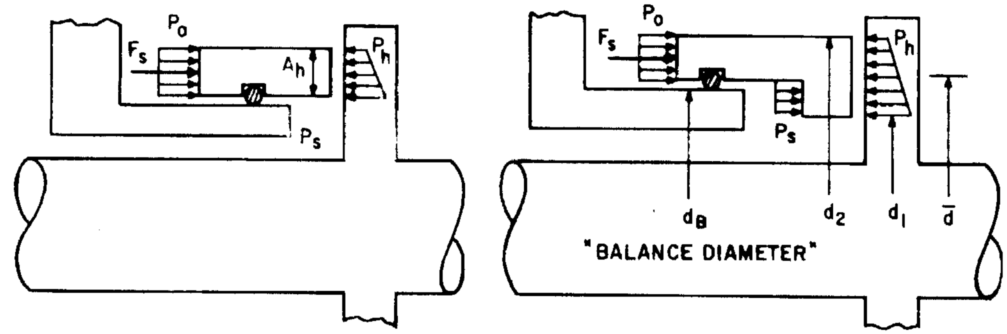
$$b \approx \frac{d_B - d_1}{d_2 - d_1} \; ; \; P_s > P_o \quad \text{(Pressure on I.D.)}$$

$$\approx \frac{d_2 - d_B}{d_2 - d_1} \; ; \; P_o > P_s \quad \text{(Pressure on O.D.)}$$

$$B \approx \frac{d_B - \bar{d}}{d_2 - d_1} \; ; P_s > P_o \quad \text{(Pressure on I.D.)}$$

$$\approx \frac{\bar{d} - d_B}{d_2 - d_1} \; ; \; P_o > P_s \quad \text{(Pressure on O.D.)}$$

Figure 8
Stationary Head
Pressure Unloaded, External if $P_s > P_o$
Pressure Loaded, Internal if $P_o > P_s$



(a) Unbalanced
b=1.0, B=.50, β=0

(b) Pressure Balanced
$b \approx .67$, $B \approx .17$, $\beta \approx .33$

$P_p = (P_s - P_o) < \frac{F_s}{A_{h/2}}$ Or seal will blow open

$$b \approx \frac{d_B - d_1}{d_2 - d_1} \; ; P_s > P_o \quad \text{(Pressure in I.D.)}$$

$$\approx \frac{d_2 - d_B}{d_2 - d_1} \; ; \; P_o > P_s \quad \text{(Pressure on O.D.)}$$

$$B \approx \frac{d_B - \bar{d}}{d_2 - d_1} \; ; \; P_s > P_o$$

$$\approx \frac{\bar{d} - d_B}{d_2 - d_1} \; ; \; P_o > P_s$$

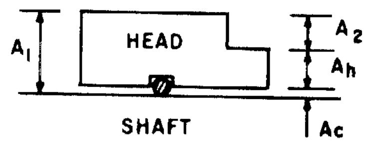
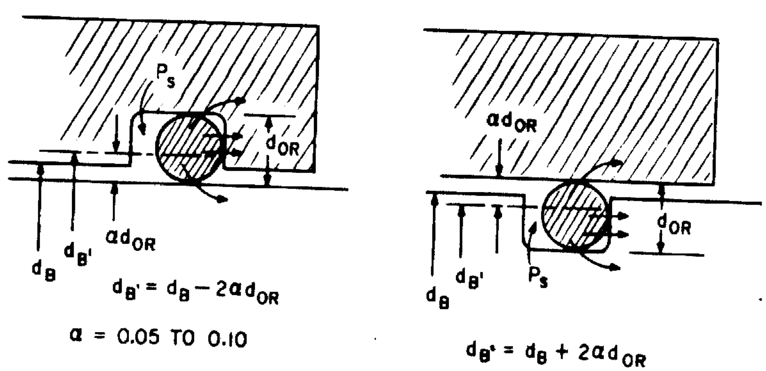


Figure 9

Seal Head Balance Areas

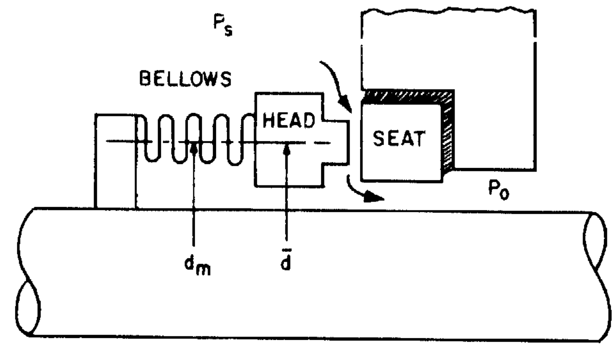


Figures 10a and 10b

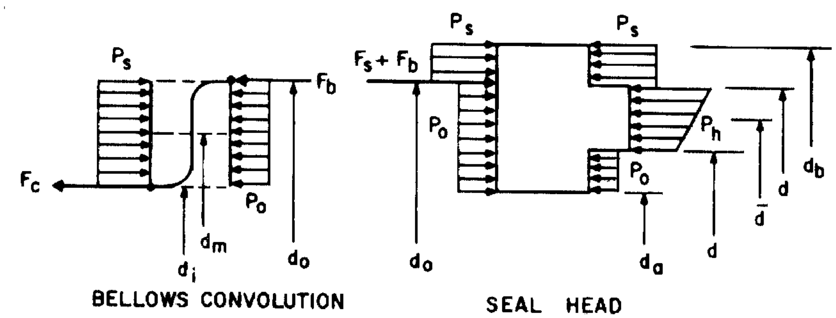
Influence of Static Seals on the

Balance Diameter [14]

Figure 11 Bellows Seal



Seal Balance = $B = \dfrac{d^2 - d_m^2}{d_2^2 - d_1^2} = (.5 - \beta) = \begin{cases} .5 \text{ unbalanced} \\ 0 \text{ to } .17 \text{ balanced design } (P_s > P_o) \\ 0 \text{ to } -.17 \text{ balanced design } (P_s < P_o) \end{cases}$



F_s = spring force in bellows

F_b = pressure force exerted by bellows

F_c = force in bellows convolution due to pressure difference

d_o = outside diameter of bellows

d_i = inside diameter of bellows

d_m = mean effective diameter of bellows $\simeq (d_o+d_i)$

d_1 = inside diameter of seal face

d_2 = outside diameter of seal face

$\bar{d}$ = average diameter of the seal face = $(d_1+d_2)/2$

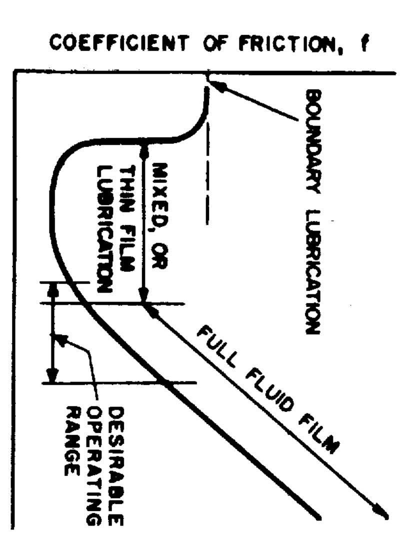
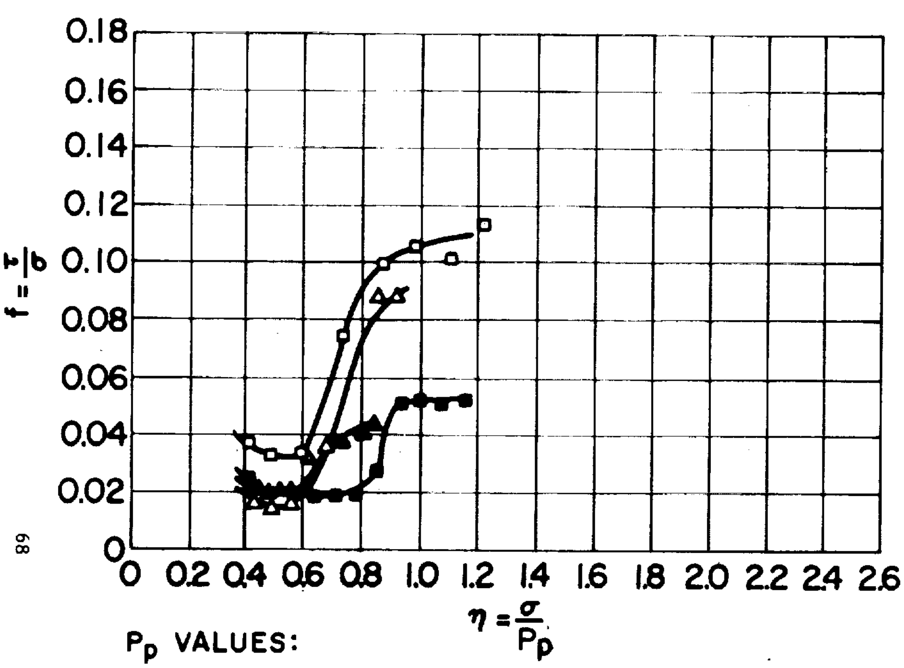


FIGURE 12 LUBRICATING REGIMES FOR A JOURNAL BEARING



f = COEFFICIENT OF FRICTION

η = $1-\beta$

σ = EXTERNAL PRESSURE APPLIED TO SEAL FACE

P_p = PRESSURE SEALED AGAINST

τ = SHEAR STRESS ON SEAL FACES

P_p VALUES:

□ = 59 PSI, FIRST HOUR

■ = 59 PSI, AFTER 20 HOURS

△ = 88 PSI, FIRST HOUR

▲ = 88 PSI, AFTER 20 HOURS

FIGURE 13 TYPICAL WET RUNNING FRICTION OF A FACE SEAL (STELLITE/CARBON)

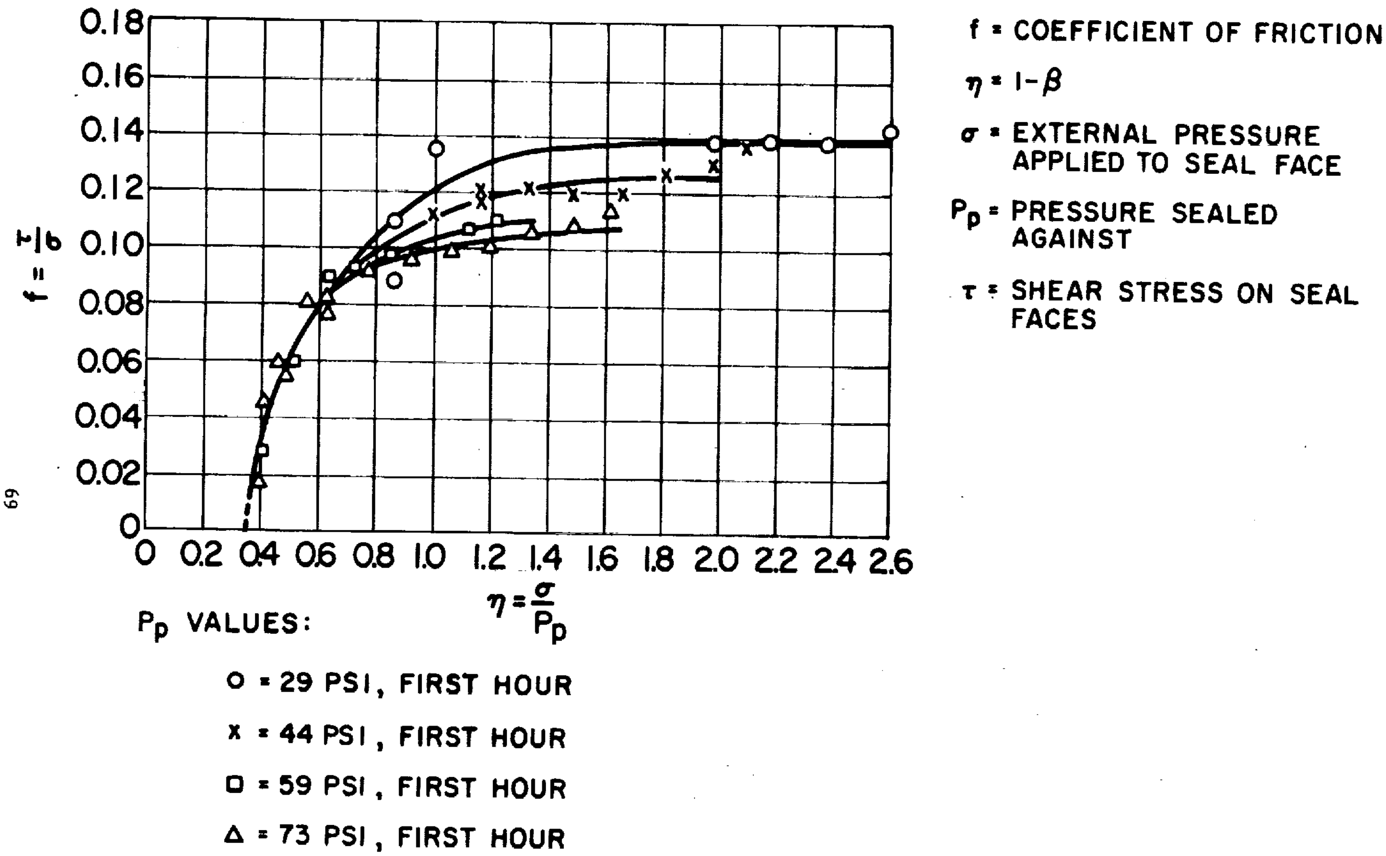


FIGURE 14 TYPICAL DRY FRICTION OF A FACE SEAL (IN AIR)

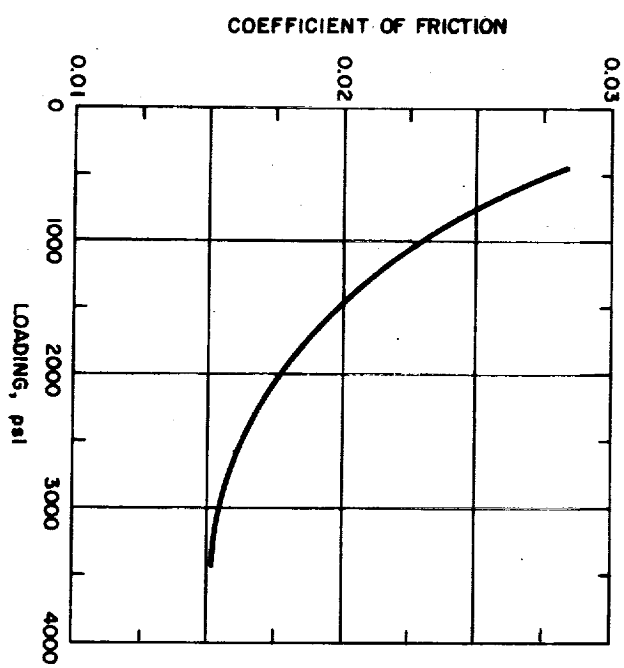


FIGURE 15 COEFFICIENT OF FRICTON FOR TEFLON VERSUS LOADING

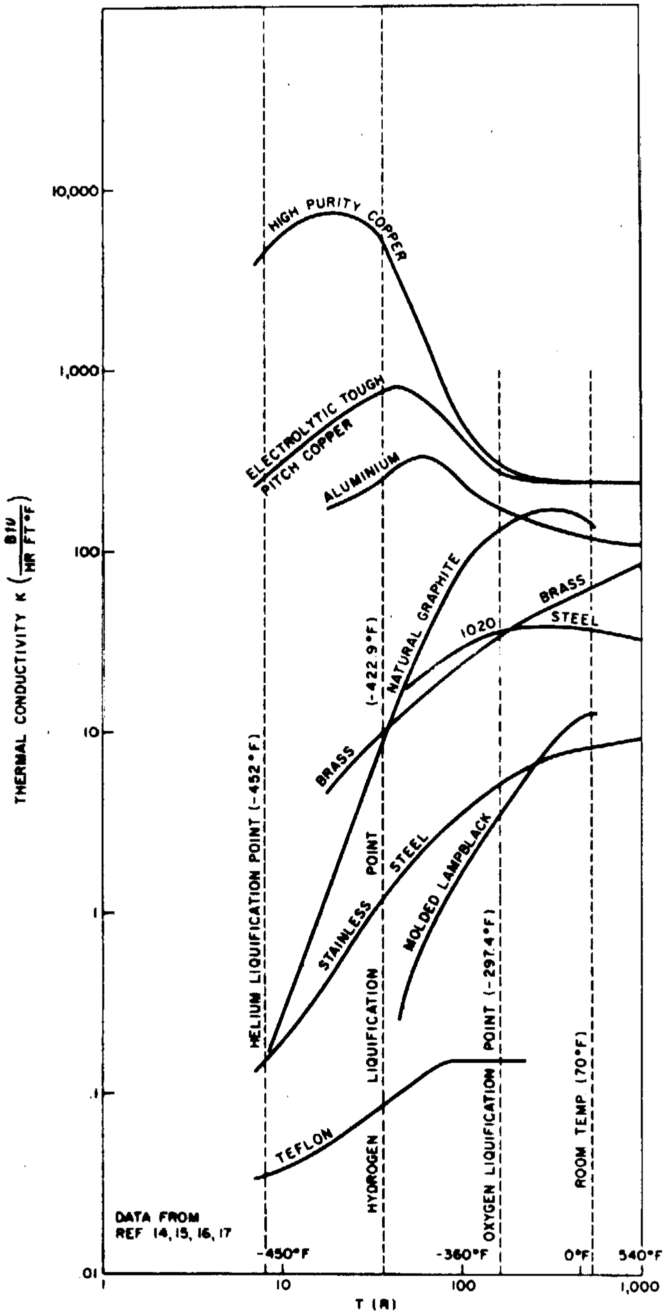
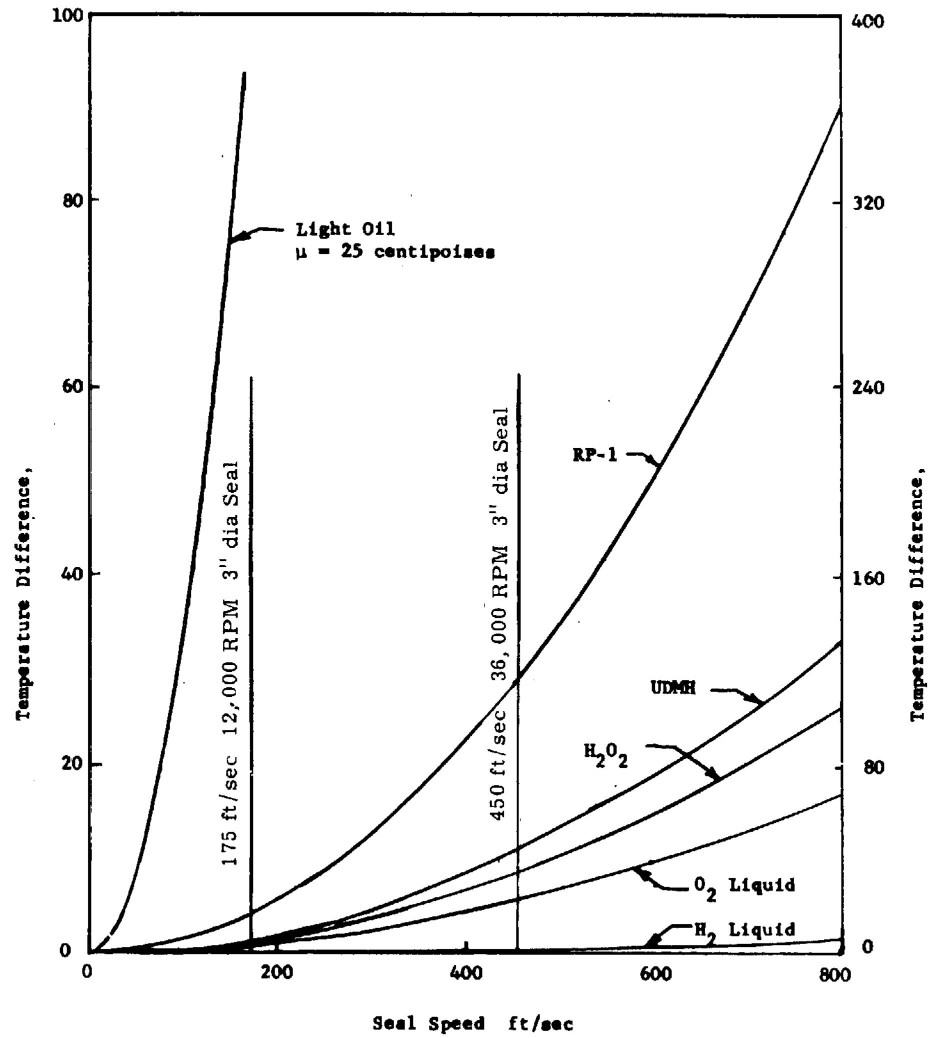


FIG 16 THERMAL CONDUCTIVITY AT CRYOGENIC TEMPERATURES

Figure 17

MAXIMUM TEMPERATURE DIFFERENCE IN SEAL FILM



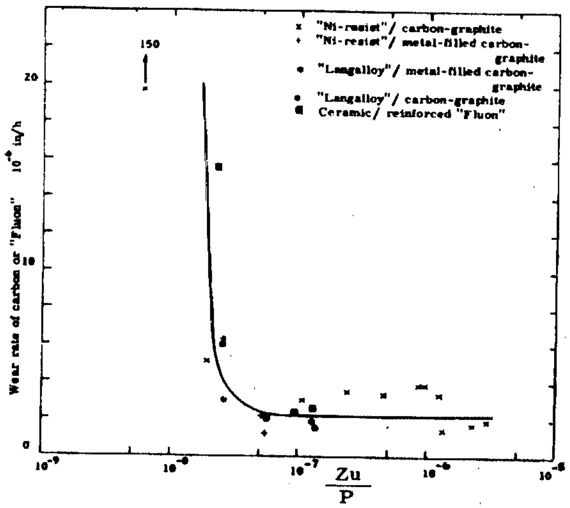


Figure 18a. Variation of Wear Rate with Duty Parameter

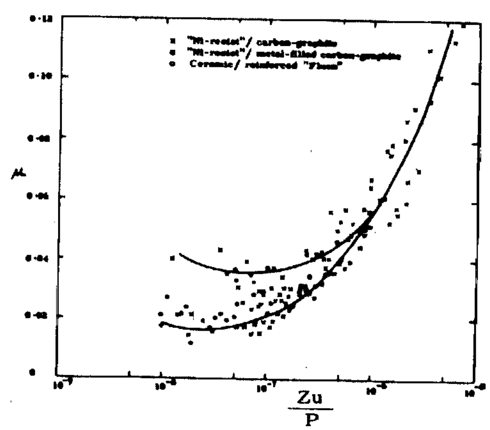


Figure 18b. Variation of Coefficient of Friction with Duty Parameter

Note: Zu/P is dimensionless

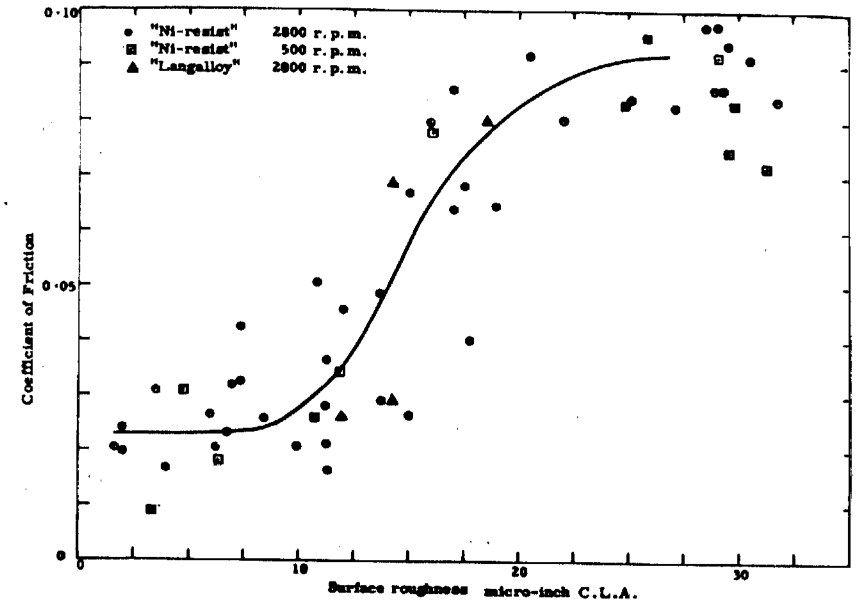


Figure 19a. Effect of Surface Roughness on Coefficient of Friction.

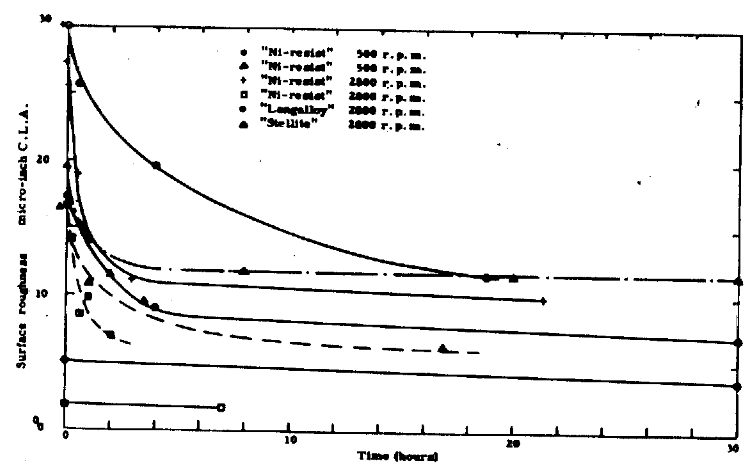


Figure 19b. Variation of Surface Roughness with Time of Running.

Note: Contact pressure 62.5 lb/sq. in counterface, metal-filled carbon-graphite; liquid, water.

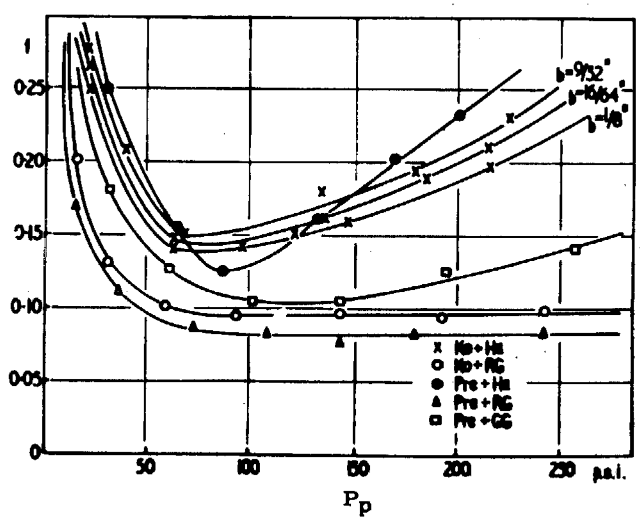


Figure 13. Friction Curves with Water, for Unbalanced Seal.

Note: V=1000 ft/min
b = Seal width = R_2-R_1

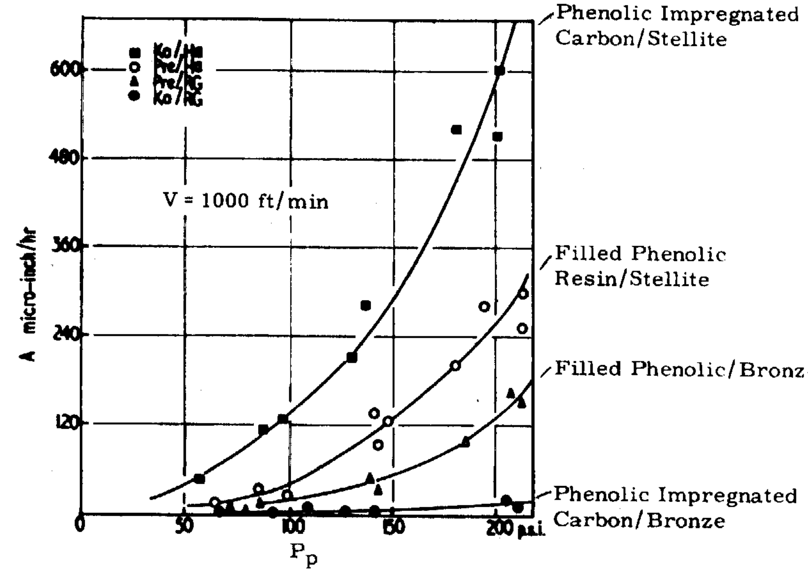


Figure 20 Wear Curves with Water, for Unbalanced Seal.

Material	High Temperature Limit T(^oF)	Thermal Conductivity (BTU/ft. h^oF)
Pre = filled phenolic resin	250	0.45
Ko = phenolic impregnated hard carbon	350	2.7
Ha = stellite	> 1500	6.7
GG = cast iron	> 1000	30.0
RG = bronze	1000	34.0

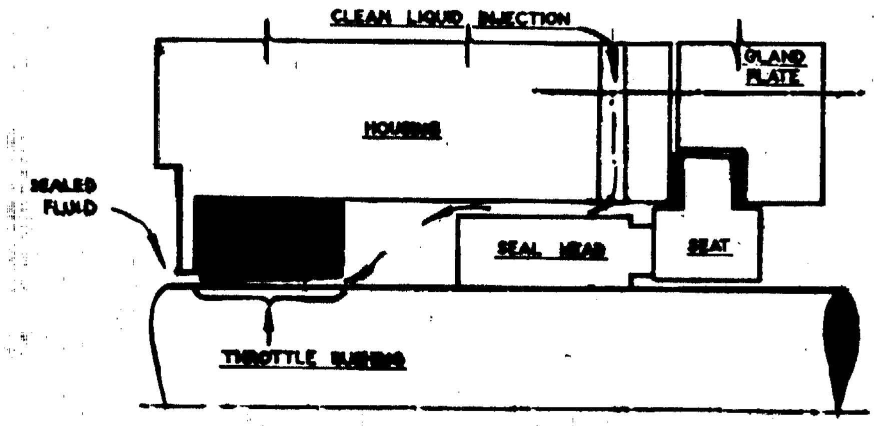


Figure 21 Quench Gland

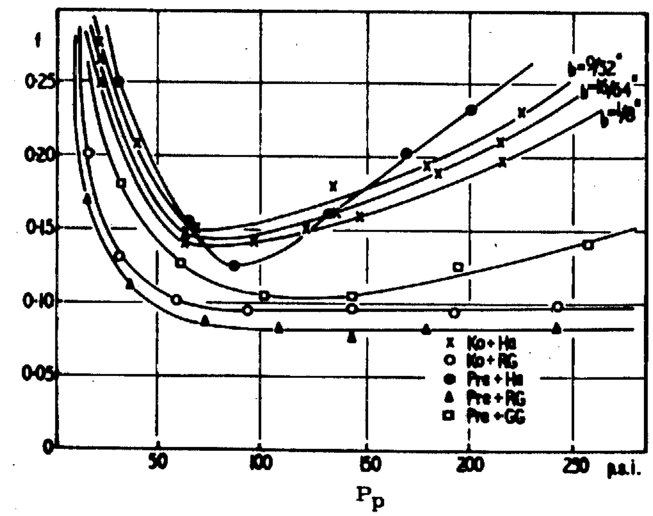


Figure 22 Friction Curves with Water, for Unbalanced Seal.

Note: V=1000 ft/min
b = Seal width = R_2-R_1

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B. HYDROSTATIC AND HYDRODYNAMIC FACE SEALS

HYDROSTATIC AND HYDRODYNAMIC FACE SEALS

I. Introduction

Perhaps one of the most important spin offs of the relatively recent fundamental face seal studies has been the introduction of hydrostatic and hydrodynamic face seals.

Hydrostatic and hydrodynamic effects have been used to explain the operation of ordinary mechanical face seals. These effects, however, were usually found to result from low level, uncontrolled variation in face seal geometry. It was inevitable that these variations be incorporated in a controlled fashion in face seal design. This is now being done at an increasing rate and with increasing interest. An important factor in this development is the broad base technology in hydrostatic and hydrodynamic bearings. This technology is now being applied to seals.

The characteristics and advantages of these types of face seals will now be discussed briefly.

Hydrostatic Face Seals

1. Develop a separating force between the primary seal faces due to either: (a) an external pressure source or, (b) the pressure drop across the seal.
2. Operate at a finite clearance and, therefore, a finite leakage.

Advantages: (a) No rubbing contact and, therefore, a longer life and more reliable operation.

(b) Does not depend directly on rotational speed for its operation.

Disadvantages: (a) Usually a higher leakage rate than an ordinary face seal.

(b) Limited tolerance for rubbing contact since designed for finite clearance.

(c) Usually more expensive because of intricate geometry and complex design.

(d) Seal dynamics are more critical because the clearance must be maintained and the seal must follow surface contours and motions.

Hydrodynamic Face Seals

1. Develop a separating force between the primary seal faces due to hydrodynamic thrust bearing type of action. Some of the designs used are: spiral grooves, Rayleigh step and tapered land.
2. Advantages and disadvantages are the same as for the hydrostatic face seals with the exception that the rotational speed is an essential factor.

II. Design

A. Film Spring Constant

An essential feature of both the hydrostatic and hydrodynamic face seals is that the fluid generated force which tends to separate the primary faces, must increase with decreasing clearance or film thickness. That is, it must be a positive spring constant. This is necessary for finite clearance operation. If the spring constant were negative the primary seal faces would be forced together and the result would be a contacting face seal.

B. Design Parameters

1. Seal Type

a. Hydrostatic

(1.) Is a reliable pressure source available; is there always a minimum pressure drop across the seal.

(2.) Wish to rely on pressure source rather than rotational speed.

(3.) Can a higher spring stiffness be obtained with the hydrostatic design in comparison to the hydrodynamic design.

(4.) Can the minimum film thickness be controlled close enough, i.e., limit the leakage.

(5.) A finite leakage always exists.

b. Hydrodynamic

(1.) If a dependable pressure source is not available and the rotational speed is relatively high, then a hydrodynamic design may be desirable.

(2.) Rubbing between the seal surfaces must be tolerated during start-up and shut-down.

(3.) Leakage may be reduced to zero if a spiral groove pumping design is used.

(4.) Lands are usually wider than ordinary face seals since the hydrodynamic lift geometry must be designed into the surface. Often the lifting part of the land is in addition to the sealing land.

c. Hybrid

(1.) Combined hydrodynamic and hydrostatic lifting forces may be a desirable design feature, or an unavoidable consequence of operating conditions.

(2.) Possible advantage of high hydrostatic film stiffness at low speeds and high hydrodynamic film stiffness at high speeds.

(3.) Hydrostatic feature avoids surface rubbing at start-up and shut-down.

2. Film Stiffness

a. Must be positive, i.e. increasing opening force with decreasing clearance.

b. A high stiffness to enable the stationary seal ring to follow the motion of the rotating seal surface due to axial runout, misalignment and thermal distortion without a severe loss of the minimum film thickness [1].

c. Must have a characteristic such that a minimum clearance and leakage can be obtained over the operating range of pressure and speed.

d. Must not be susceptible to reduction in stiffness due to thermal coning or dishing, i.e. these characteristics should be kept to a minimum.

e. Must provide a stable dynamic behavior.

3. Pressure Differential

a. Determines leakage.

b. Provides hydraulic separating force in the hydrostatic design.

c. If high, may even effect behavior of some hydrodynamic designs.

d. May deform primary seal rings and thus change the film thickness profile.

4. Speed.

a. If speed and fluid viscosity are low, a hydrostatic design may be necessary.

b. Friction heating increases with speed.

c. Dynamic problems, such as vibrations, are usually more severe at high speeds.

d. A segmented design may be required, especially in larger seals, to enable the faces to dynamically track one another and maintain a small but finite clearance.

5. Fluid

a. Viscosity is a very important parameter for hydrodynamic face seals. If it is low, this type of design may be restricted to only high speed seals.

b. Viscosity usually is not a major parameter in hydrostatic seal design as far as clearance control is concerned. It does directly effect the leakage rate, however.

c. The chemical aggressiveness of the sealed fluid effects the design in the same manner as it would an ordinary face seal. The difference here is that quite often the material selection for a hydrostatic/hydrodynamic face seal is somewhat different than that for an ordinary face seal.

C. Fundamentals of Hydrostatic Face Seals

1. Positive Film Spring Constant

As discussed before, it is necessary to have a positive film spring constant in order to insure static stability of the seal. These arguments are

important since they also apply to balancing of ordinary mechanical face seals.

[2] It is well known that the transverse hydrostatic pressure distribution, across the seal land, is strictly a function of film (clearance) profile in the laminar regime. If the film is parallel, then the unit seal pressure or average pressure (separating force) does not vary with film thickness and thus produces no stiffness. This is the case of neutral stability. If the film is convergent, that is, the clearance decreases in the direction of flow, then the average pressure increases when the mean film thickness decreases. This gives a positive stiffness and hence a stable fluid film. On the other hand, a divergent film would cause the average pressure to decrease when the film decreases and thus gives rise to a statically unstable fluid film.

This can be demonstrated with the step geometry, which is a common and easily obtainable hydrostatic geometry, Figure 1a. The variation in pressure distribution and separating force is shown in Figures 1b and 1c. Also shown in Figure 1c is the mechanical spring force which is always applied to the face seal to hold it "closed." The intersection of these two curves gives the operating clearance of the seal. The magnitude of the separating force will also depend upon the balance diameter of the seal (see Seal Balance under Mechanical Face Seals).

For isothermal, compressible, laminar flow (no inertia effects) the load per unit length of seal is [2]

$$W = b\,(P_2 - P_1)\,\bar{W} \tag{1}$$

$$\bar{W} = \frac{1}{1-r_{12}}\left\{\frac{b_2}{b}\left[\frac{2(r_{i2}^2 + r_{i2} + 1)}{3\,(r_{i2} + 1)} - r_{12}\right] + \frac{b_1 r_{i2}}{b}\left[\frac{2(r_{1i}^2 + r_{1i} + 1)}{3\,(r_{1i} + 1)} - r_{1i}\right]\right\} \tag{2}$$

$$r_{12} = P_1/P_2 \tag{3}$$

$$r_{i2} = P_1/P_2 = \left[\frac{(h_1/h_2)^3\,(b_2/b_1)\,(P_1/P_2)^2 + 1}{(h_1/h_2)^3\,(b_2/b_1) + 1}\right]^{1/2} \tag{4}$$

$$r_{1i} = P_1/P_i = (P_1/P_2)\,(P_2/P_i) = (P_1/P_2)\,(1/r_{i2}) \tag{5}$$

The mass flow rate per unit length, m'; the center of pressure, $\bar{X}_c$, and the seal stiffness per unit area, K_s^1; are given by the following expressions 2

$$m' = \frac{h_2^3}{24\mu b_2} \rho_2 P_2 \quad 1 - r_{i2}^2 \tag{6}$$

$$\bar{X}_c = \frac{Xc}{b} = \int^1 \frac{(p-p_1)\,\bar{x}\,d\,\bar{x}}{\bar{W}\,(p_2-p_1)} \tag{7}$$

$$\bar{K}_s = \frac{K_s^1\ h_1}{\bar{H}\ (P_2-P_1)} = \frac{d\bar{W}}{dr_{i2}} \quad \frac{dr_{i2}}{d\bar{H}} \tag{8}$$

The quantities $\bar{W}$, $\bar{X}_c$, $\bar{K}_s$, and $\bar{M}$ are given in Figures 2, 3, 4 and 5. for $b_1/b = 0.25$.

For optimum performance the seal should be designed to operate near the maximum stiffness [2].

2. Controlled Leakage

In order to control the leakage rate, it can be seen from equation (6) that the film thickness or clearance must be controlled to a very small value. To do this Figure 1c shows that a force curve that changes rapidly with clearance about the operating point is needed. This means a high film stiffness K_s^1. Equations (6) and (7) determine the film thickness required for a given maximum leakage rate.

3. Film Shape [2]

The shape of the film gap has a very critical influence on film stability, stiffness and leakage rate. A convergent film where the clearance decreases in the direction of flow given a positive stiffness. An excessive convergence is undesirable because it opens up the gap, reduces the stiffness, and increases the leakage rate. A divergent film is unstable, leading to surface rubbing.

4. Dynamic Tracking [2]

To permit tracking runner motion without diminishing the film thickness excessively, the primary seal must have a high stiffness film and a low mass. The seal face may be subjected to various irregular motions of the rotor due to runout, initial warping, and elastic or thermal distortion of the runner.

If the bending and twisting deformation of the seal ring is small in comparison to the film thickness, a rigid ring may be assumed. For large diameter seals, the axial and angular vibrations can be represented by a one-degree-of-freedom model that gives the following response of ring amplitude to runner amplitude

$$\frac{\circ}{f} = \left[\left(1 - \frac{m\,\omega^2 - k^*}{k_f}\right)^2 + \left(\frac{\omega c}{k_f}\right)^2\right]^{-1/2} - 1 \tag{9}$$

5. Seal Ring Distortions [2]

Since the static stability of a fluid film depends critically upon the shape of the gap, distortions of the seal ring can play an extremely important role in designing face seals.

The three types of distortions are:

(a) Initial distortions which are due mostly to a lack of perfection in manufacturing.

(b) Pressure distortions caused by unbalanced external pressures acting on the seal ring.

(c) Thermal distortions which are the result of friction heating in the fluid film.

The combined distortions must be controlled so that the film clearance does not become divergent, since this condition leads to seal instability.

6. Non-homogeneous Flow [3]

A gas seal contaminated with a liquid, such as oil, will hydrostatically seek a smaller operating gap, due to the change in the pressure profile existing across the seal face. A plain hydrostatic gas seal will often fail when contaminated with oil. The addition of spiral grooves, however, allows successful operation of a liquid-contaminated seal, due to the much greater pumping term produced from the larger viscosity of the liquid.

7. Balance [3]

It is important to have the seal hydrostatically balanced to operate at the desired gap at high pressure with zero contact. At high pressures, the hydrostatic force balance not only creates an initial gap for starting, but serves to keep the gap from becoming excessively large due to hydrodynamic action.

Principles of seal balance discussed under Mechanical Face Seals, also apply here.

8. Flow Phenomena Problems 4

The analysis used for calculating pressures in narrow passages (seals) is valid only for slow flow. These calculations are affected by the following:

(a) Vena Contracta at the entrance to the seal, similar to an orifice or labyrinth seal. A significant reduction in flow area should be expected followed by a small expansion angle. This will cause a low pressure region at the seal entrance and reduced separating force.

(b) Turbulence which will affect the leakage rate. Normally this turbulence is a consequence of the high rotational speed. In order to take this into account, the methods used under Viscoseals can be applied directly.

(c) Choked flow at the slot exit which occurs at high pressure ratios. Choking will cause an increase in the average gap pressure, thus tending to open the seal gap.

9. Basic Equations

Most of the design data has been calculated using the non-inertial, laminar flow equation, neglecting entrance and exit losses. As discussed under Viscoseals, turbulent flow analyses are available, however.

For incompressible flow the volumetric flow rate is

$$q\ (x) = -\frac{h^3}{12\mu}\ \frac{\partial P}{\partial x} = \text{const.} \qquad (10)$$

Since the flow through the seal clearance must be constant, this equation can be integrated for a given film clearance distribution, h (x), to give the pressure and separating force.

For a step seal operating under incompressible conditions the flow is

$$m' = \rho q = \frac{h^3}{12\mu\ b_2}\ \rho_2\ P_2\ \left(1-\frac{Pi}{P2}\right). \qquad (11)$$

This expression can be compared to equation (6) which was solved for compressible isothermal flow conditions, i.e.

$$m' = -\rho\ \frac{h^3}{12\mu}\ \frac{\partial P}{\partial x} = \text{const.} \qquad (12)$$

From these two basic equations (10, 12) all of the hydrostatic face seal data has been computed.

10. Design Data

In addition to the data already given for the hydrostatic step seal, some additional design data is included here.

D. Fundamentals of Hydrodynamic Face Seals

1. General

Many of the design considerations which apply to hydrostatic face seals apply also to hydrodynamic face seals. For example:

(a) A <u>positive film spring constant</u> is necessary.

(b) Controll of <u>leakage</u> depends upon the spring constant.

(c) The <u>film shape</u> must have a hydrodynamic characteristic. It is desirable to have a convergent film, however, the hydrodynamic design has a higher degree of tolerance for a divergent film shape.

(d) To insure <u>dynamic tracking</u> the film stiffness should be maximized.

(e) <u>Seal ring distortion</u> may have an adverse effect on the seal performance but it is usually not as severe as for the hydrostatic design.

(f) <u>Liquid contamination</u> of a gas seal usually increases the hydrodynamic effect and, therefore, will open the clearance rather than close it. This may result in a short term increase in leakage but it avoids rubbing contact which might be catastrophic.

(g) <u>Balance</u> of hydraulic forces is just as important here as in any other face seal.

(h) Laminar <u>flow</u> is usually assumed here. However, turbulent analyses, as used for Viscoseals, are available. Convective inertia effects are only beginning to be looked at.

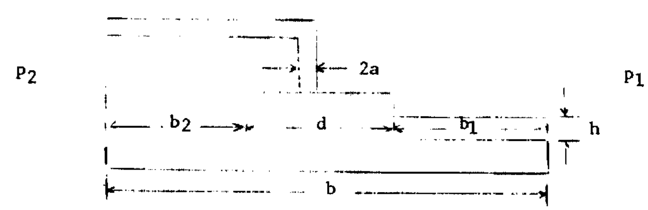
(i) The <u>basic equations</u> used are the same as for hydrodynamic thrust bearings. Usually a seal dam is used along with the hydrodynamic lifting region. The leakage over the dam can be determined from the equations already presented (10, 11).

2. Design Data

Specific design data is still somewhat limited for hydrodynamic face seals. However, all of the hydrodynamic thrust bearing data and theory are available to apply to this technology.

Design information for hydrodynamic face seals may be found in [1, 2, 4, 6, 7, 8, 9].

Hydrostatic - Orifice Compensated Seal [2]



For maximum film stiffness and $b_1/b = 0.25$, $p_1/p_2 = 0.2$:

$$\bar{K}_s = K_s' \, h / \left[\bar{H} \, (p_2 - p_1)\right] = 0.48$$

$$\bar{H} = h \left[\frac{\ell \, \rho_2 \, (RT_2)^{1/2}}{24 \mu \, b \, \pi \, a^2} \right]^{1/3} = 0.55$$

$$\bar{W} = W \Big/ \left[(p_2 - p_1) \, b\right] = 0.82$$

$$\bar{M} = m' \left[\frac{24 \mu b R T_2}{h^3 \, p_2^2} \right] = 2.75$$

where

ℓ = seal length between orifices, in

ρ_2 = gas density at supply

R = universal gas constant $in^2/sec^2 {}^oR$

T_2 = absolute temperature at supply, ^oR

μ = gas viscosity, psi-sec

a = orifice radius, in

K_s' = seal film stiffness per unit area, $(lb/in)/in^2$

W = seal load per unit length, lb/in

m' = mass flow rate per unit seal length

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TABLE 1

Hydrostatic Controlled Clearance Face Seal Design Data [5]

Radial Clearance Equations Laminar incompressible flow	Mean pressure ratio: $\bar{P} = \frac{\bar{p} - p_2}{p_1 - p_2}$; $(p_1 > p_2)$ Mass flow rate $\dot{m} = \rho \frac{\pi s^3 (p_1 - p_2)}{6 \eta \ln R} \cdot C_m$ Power loss $N_v = \frac{\dot{m}}{\rho}(p_1 - p_2) + \frac{\pi \eta \omega^2 r_i^4 (R^4 - 1)}{2s} \cdot C_N$	$S = \frac{s}{e}$; $U = \frac{S}{S+1}$ $R = \frac{r_a}{r_i}$; $T = \frac{R}{(R-1)S} + 1$ $R_1 = \frac{r_s}{r_i}$; $W = 1 - \frac{1}{(R-1)S}$ $R_2 = \frac{r_a}{r_s}$ $\Omega = \frac{(3/20) \rho \omega^2 r_i^2 (R^2 - 1)}{p_1 - p_2}$

Clearance geometry	$\bar{P}$	C_m [1)]	C_N
1	$\frac{-\ln\frac{R}{U}+(R^2-1)T(1+\frac{T}{2})+(R-1)^2(1-T^2U)TS}{(R^2-1)\left[\ln\frac{R}{U}+T(1+\frac{T}{2})-TU(1+\frac{TU}{2})\right]}(1+\Omega)-\frac{\Omega}{2}$	$\frac{T^3 \ln R(1+\Omega)}{\ln\frac{R}{U}+T(1+\frac{T}{2})-TU(1+\frac{TU}{2})}$	$\frac{4(R-1)^4S^4}{R^4-1}\left[-\frac{U^{-3}-1}{3}+\frac{3}{2}T(U^{-2}-1)-3T^2(U^{-1}-1)-T^3\ln U\right]$
2	$\frac{R^2\ln RU-(R^2-1)W(1+\frac{W}{2})+(R-1)^2(1-W^2U)WS}{(R^2-1)\left[\ln RU-W(1+\frac{W}{2})+WU(1+\frac{WU}{2})\right]}(1-\Omega)+\frac{\Omega}{2}$	$\frac{W^3 \ln R(1-\Omega)}{\ln RU-W(1+\frac{W}{2})+WU(1+\frac{WU}{2})}$	$\frac{4(R-1)^4S^4}{R^4-1}\left[\frac{U^{-3}-1}{3}-\frac{3}{2}W(U^{-2}-1)+3W^2(U^{-1}-1)+W^3\ln U\right]$
3	$\frac{-\ln R_2-U^3(\ln R_1-\frac{R_1^2-1}{2})+\frac{R^2-R_1^2}{2}}{(R^2-1)(U^3\ln R_1+\ln R_2)}(1+\Omega)-\frac{\Omega}{2}$	$\frac{\ln R(1+\Omega)}{U^3\ln R_1+\ln R_2}$	$\frac{U(R_1^4-1)+R_1^4(R_2^4-1)}{R^4-1}$
4	$\frac{R^2\ln R_1+U^3(R^2\ln R_2-\frac{R^2-R_1^2}{2})-\frac{R_1^2-1}{2}}{(R^2-1)(\ln R_1+U^3\ln R_2)}(1-\Omega)+\frac{\Omega}{2}$	$\frac{\ln R(1-\Omega)}{\ln R_1+U^3\ln R_2}$	$\frac{R_1^4-1+UR_1^4(R_2^4-1)}{R^4-1}$
1) For determination of $\dot{m}$ consider change of clearance s for various Ω (Equilibrium of forces)			
5	$\left(\frac{1}{2\ln R}-\frac{1}{R^2-1}\right)(1+\Omega)-\frac{\Omega}{2}$	1	1
6	$\left(\frac{R^2}{R^2-1}-\frac{1}{2\ln R}\right)(1-\Omega)+\frac{\Omega}{2}$	1	1

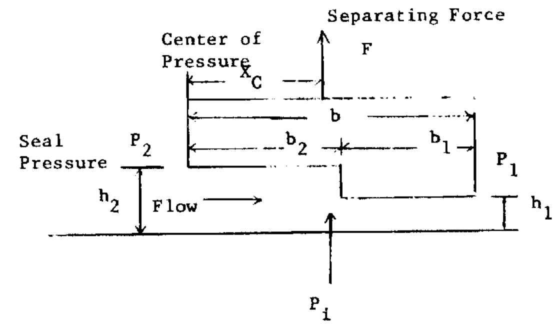


Figure 1a

Cross Section of Hydrostatic Step Seal

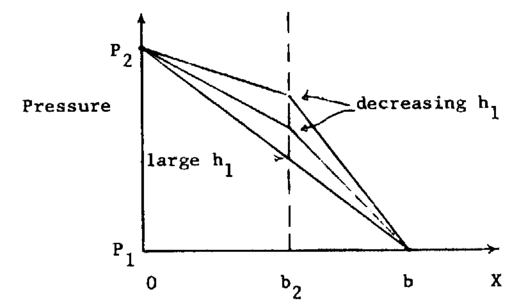
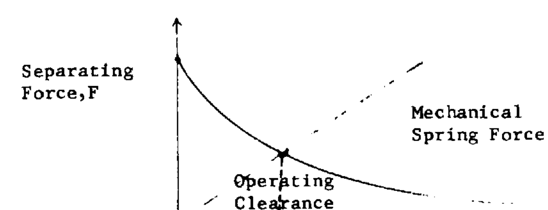


Figure 1b

Laminar Flow Pressure Profiles



Minimum Clearance, h_1

Figure 1c

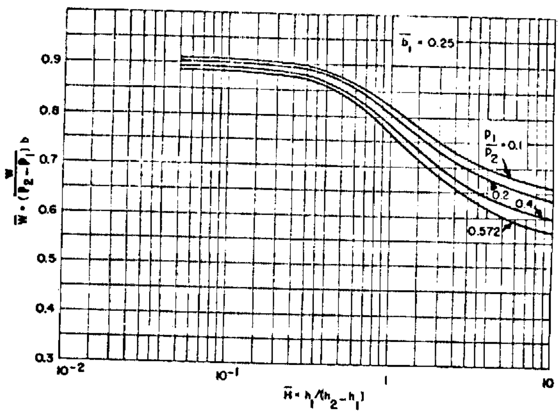


Figure 2. Dimensionless load $\overline{W}$ versus step ratio $\overline{H}$ of a typical hydrostatic step seal ($\bar{b}_1 = 0.25$) under different pressure ratios.

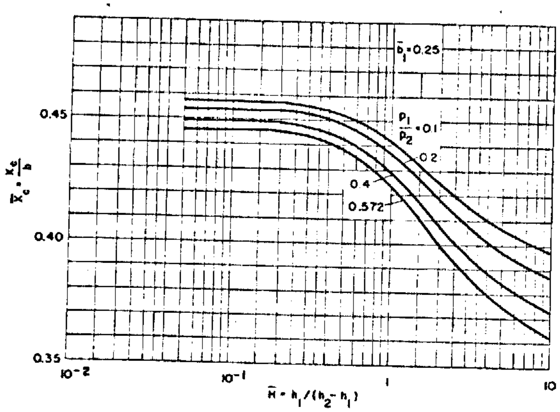


Figure 3. Dimensionless center of pressure $\overline{X}_c$ versus $\overline{H}$ of a typical hydrostatic step seal ($\bar{b}_1 = 0.25$) under different pressure ratios.

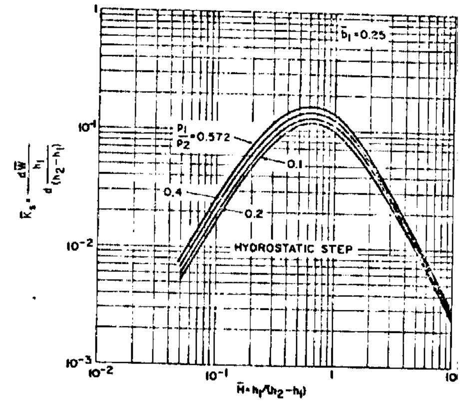


Figure 4. Dimensionless stiffness $\overline{K}_s$ versus $\overline{H}$ of a typical hydrostatic step seal ($\bar{b}_1 = 0.25$) under different pressure ratios.

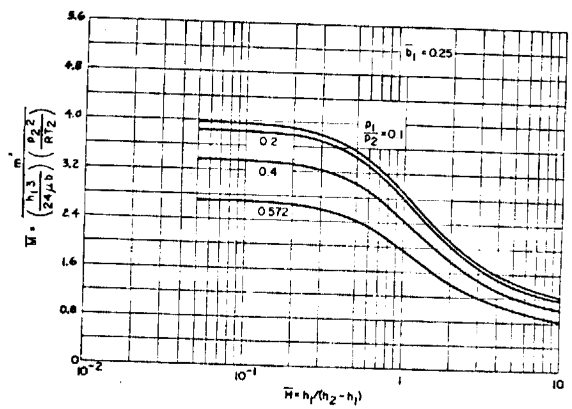


Figure 5. Dimensionless flow $\overline{M}$ versus $\overline{H}$ of a typical hydrostatic step seal ($\bar{b}_1 = 0.25$).

C. BUSHING SEALS

BUSHING SEALS

I. Introduction

Description

The bushing seal is a clearance type seal which relies on a close clearance bushing, or sleeve, to limit leakage along a shaft (Figure 1). It is desirable not to have the shaft touch the bushing, and ideally they are concentric.

There are two basic types of bushing seals. They are:

Fixed Bushing [1]

This is a simple design with the sleeve or bushing fixed to the housing.

Advantages

Simple construction and assembly.

Disadvantages

(a) Acts as a redundant journal bearing.

(b) Large clearances to allow for eccentricity shaft motion and misalignment.

(c) High leakage.

(d) Long bushings must be used to cut down on leakage. This leads to more difficult alignment problems.

Floating Bushing [1]

In order to overcome some of the disadvantages of the fixed bushing seal, this design allows the bushing to move radially with the shaft (Figure 2).

Advantages

(a) Bushing tends to align itself with shaft, run concentric, and follow shaft motion.

(b) Thus rubs are less likely to occur.

(c) Smaller clearance can be used.

(d) Lower leakage results.

(e) Shorter bushings can be used.

Disadvantages

(a) More complex and costly.

(b) A moving secondary seal must be provided.

(c) Inertia, squeeze film, friction and springs may prevent the sleeve from following the shaft as closely as desired.

(d) Journal bearing action may cause "whirl" at higher speeds.

Disadvantages (continued)

(e) Seal balancing, i.e. balancing of axial hydraulic forces, may be required for high pressure applications.

Single Floating Bushing (Figure 2)

(a) Not recommended for high pressure applications.

(b) Large mass (inertia).

(c) Tends to unseat secondary seal during shaft misalignment.

Floating Ring Seal (Figure 3)

(a) Improves on above problems by using multiple bushings of short length.

(b) Each ring moves independently.

(c) Permits greater shaft misalignment.

(d) Comparable sealing effectiveness.

(e) Assembly length is increased.

Segmented Ring Seal (Figure 4)

(a) Segmented ring.

(b) Assembled with garter spring.

(c) Chamfered interface between seal ring and retainer. Keeps primary seal clearance small and provides for secondary seal at the same time.

II. Seal Design

A. Leakage

1. Single Phase, Incompressible Flow [2, 3, 4, 5, 6]

The experimental corrolations of Yamada [2], and Tao and Donovan [3]; and the analyses of Vohr and Chow [4], Ng and Pan [5], and Elroad and Ng [6], are applicable to this problem.

Assumptions:

(a) Homogeneous, single-phase flow.

(b) Isothermal, constant viscosity μ.

(c) Incompressible, constant density ρ.

(d) Smooth surfaces.

(e) Inner cylinder rotating.

(f) Fine clearances.

(g) Clearance is constant axially, i.e., no misalignment or changes in geometry.

(h) Entrance and exit losses are neglected analytically but are included in experimental measurements.

Flow Regimes [2] (Figure 5)

Laminar
Laminar & Taylor Vorticies
Turbulent
Turbulent & Taylor Vorticies

Flow Resistance [2, 3]

Pressure Drop

$$\Delta P = (fL/2c)\ (\rho \bar{w}^2/2g) \tag{1}$$

Friction Factor

$$f = f\ (Re_z,\ Re_\omega,\ {}^c/R,\ \varepsilon) \tag{2}$$

Average axial velocity

$$\bar{w} = Q/(\pi DC) \tag{3}$$

Reynolds number (through flow)

$$Re_z = c\,\bar{w}/\nu \tag{4}$$

Reynolds number (rotational)

$$Re_\omega = cU/\nu = cR\,\omega/\nu \tag{5}$$

Laminar Flow

$$[2]\quad f = {}^{48}/Re_z\ ;\ Re_z < 1000\ ,\ 0 \leq Re_\omega \leq 1000 \tag{6}$$

$$[3]\quad f = 41.5/Re_z \tag{7}$$

Turbulent Through flow; $Re_\omega = 0$

$$[2]\quad f = 0.26\ Re_z^{-0.24} \tag{8}$$

$$[3]\quad f = 0.237\ Re_z^{-0.21} \tag{9}$$

Turbulent Rotational Flow; $Re_\omega \geq 10^4$, Re_z any value

$$[2]\quad f = 0.26\ Re_z^{-0.26}\left\{1 + \left(\frac{7}{8}\right)^2\left(\frac{Re_\omega}{2Re_z}\right)^2\right\}^{0.38} \tag{10}$$

$$= 0.26\ Re_z^{-0.24}\left\{1 + 0.765\ (R\omega/2\bar{w})^2\right\}^{0.38} \tag{11}$$

$$[3]\quad f = 0.237\ Re_z^{-0.21}\left\{1 + (R\,\omega/2\,\bar{w})^2\right\}^{0.395} \tag{12}$$

Laminar + Taylor Vortices / Turbulent + Taylor Vortices [2]

(see Figure 7)

Eccentric Cylinders [3]

Laminar Flow: $$\frac{Q_\varepsilon}{Q} = (1 + 1.5\,\varepsilon^2);\ \varepsilon = \frac{e}{c} \qquad (13)$$

Turbulent Flow: $$\frac{Q_\varepsilon}{Q} \approx (1 + a\,\varepsilon^2);\ .2 \le a \le .4$$

use $a = .3$ (14)

It is not clear, for the rotating cylinder case, just where the laminar or turbulent eccentricity corrections should be applied. Obviously there must be a transition where the correction factor goes from $(1 + 1.5\,\varepsilon^2)$ to $(1 + .3\,\varepsilon^2)$ and not two distinct factors.

Direct Calculation of Leakage

Using the friction factor approach requires a trial- and-error solution for the leakage. However it may be noted that over much of the range of interest the friction factor is given in the following form:

$$f = \frac{K}{V^n} \quad ;\ V = \bar{\omega} \qquad (15)$$

where

$$Q = VA = V\pi\ D.\ C. \qquad (16)$$

Thus the leakage rate, for the general case, is given by

$$Q = \pi\ D\ C\left[\frac{4\ c\ \Delta\ P}{K_p\ L}\right]^{\frac{1}{2-n}} \qquad (17)$$

Laminar Flow $Re < 10^3$, $R_\omega < 10^3$

$$f = \frac{48}{Re_z} = \left(\frac{48\mu}{\rho c}\right)\Big/ V \qquad n = 1 \qquad (18)$$

$$Q = \pi\ D\ C\left[\frac{4\ c\ \Delta\ P}{(48\mu/\rho c)\ \rho L}\right]^{\frac{1}{2-1} = 1} = \frac{\pi\ D\ c^3\ \Delta\ P}{12\ \mu\ L} \qquad (19)$$

Turbulent Through Flow $Re_\omega = 0$

$$f = \frac{0.26}{Re^{0.24}} = 0.26\left(\frac{\mu}{\rho c}\right)^{0.24} \frac{1}{V^{0.24}} \quad ;\ \varkappa = 0.24 \qquad (20)$$

$$Q = \pi\ D\ C\left[\frac{4\ c\ \Delta P}{0.26\ (\mu/\rho\delta)^{0.24}\ \rho L}\right]^{\frac{1}{2-0.24} = \frac{1}{1.76} = .568}$$

$$= \frac{4.72\ \pi\ D\ C^{1.7}}{\mu^{0.136}\rho^{0.43}}\left(\frac{\Delta P}{L}\right)^{.568} \qquad (21)$$

Turbulent Rotational Flow: $Re_\omega > 10^4$, any Re_z

$$f = 0.26\ Re^{-0.24}\left\{1 + \left(\frac{7}{8}\right)^2\left(\frac{R_\omega}{2Re}\right)^2\right\}^{0.38}$$

$$\simeq 0.26\ Re^{-0.24}\left(\frac{7}{16}\right)^{0.76}\left(\frac{R\,\omega}{Re}\right)^{0.76}$$

$$= \frac{0.139\ R_\omega^{\ 0.76}}{Re}$$

$$= 0.139\left(\frac{\mu}{\rho c}\right)^{0.24} \frac{(r\omega)^{0.76}}{V} \quad ;\ \varkappa = 1 \qquad (22)$$

$$Q = \pi\ D\ c\left[\frac{4\ c\ \Delta P}{0.139\left(\frac{\mu}{\rho c}\right)^{0.24}(r\omega)^{0.76}\ \rho L}\right]^{\frac{1}{2-1} = 1}$$

$$= \frac{28.8\ \pi\ D\ c^{2.24}}{\mu^{0.24}(\rho\ r\ \omega)^{0.76}} \cdot \frac{\Delta P}{L}$$

$$= \frac{346}{R_\omega^{\ 0.76}}\left(\frac{\pi\ D\ c^3\ \Delta\ P}{12\ \mu\ L}\right) = \frac{346}{R_\omega^{\ 0.76}}\ Q_{lam}; \qquad R_\omega > 10^4,\ \left(\frac{R\,\omega}{2Re}\right)^2 \gg 1 \qquad (23)$$

(linear part of curves)

For a comparison of these results with those of [4,5, 6], see the discussion in the chapter on viscoseals.

2. Including Heat Generation Effects, Laminar Flow [8]

A simplified method for calculating leakage and power loss in a laminar flow bushing seal is given in [8]. The assumptions made in this paper are:

(a) The sleeve and shaft are concentric.
(b) Flow is laminar. Negligible inertia effects.
(c) Small clearance.
(d) Incompressible fluid, constant specific heat.
(e) Fluid viscosity can be represented by, $\mu = \mu_i e^{-\beta T}$.
(f) All rotational, viscous heat generation goes to increase the temperature of the fluid, i.e., no heat transfer.
(g) Neglect the viscous heat generation due to axial leakage.

The calculational procedure is as follows:

Given

L = seal length, in
c = seal clearance (radial), in
ΔP = seal pressure differential, psi
β = thermal coefficient of viscosity, $^{o}F^{-1}$
ρ = fluid density, lb_m/in^3
N = shaft speed, rpm
D = seal diameter, in
μi = viscosity at inlet temperature, psi-sec
Cp = specific heat, $Btu/lb_m{}^oF$

Calculate the following:

(a) Isothermal Flow Rate

$$Qi = (\pi\ D\ c^3\ \Delta P)\ /\ (12\ \mu i L) \qquad (24)$$

(b) Adiabatic Flow Number

$$\Phi = (J\ \rho\ Cp\ c^4\ \Delta P)\ /\ \left[12\beta\ (\mu_i\ L\ \upsilon)^2\right] \qquad (25)$$

J = mechanical equivalent of heat
$U = R\omega$

(c) Obtain R (Figure 8)

(d) Adiabatic Flow Rate

$$Q_a = R\ Qi \qquad (26)$$

(e) Obtain θ (Figure 9)

(f) Adiabatic Temperature Rise

$$\Delta T = \theta/\beta \qquad (27)$$

(g) Isothermal Power Loss

$$Hp_i = (\pi\ D\ U^2 \mu_i\ L)\ /\ (12x550c) \qquad (28)$$

(h) Adiabatic Power Loss

$$Hp_a = Hp_{i/R} \qquad (29)$$

3. Including Pressure Deformations [9]

To calculate leakage and friction losses, in laminar flow, the isothermal values are first computed. Corrections to these values may be obtained from Figures 10 and 11 after estimating the elastic deformations at the ends of the seal.

The effect of the high pressure on the viscosity and the clearance, is a large change in frictional torque but a relatively small change in leakage.

Assumptions:
(a) Laminar flow
(b) Isothermal flow
(c) Incompressible flow
(d) Neglect tangential pressure gradients
(e) Pressure effect on viscosity given by $\mu = \mu_o\ (1+B\ \rho)$ (30)
(f) Deflection of sleeve and shaft assumed to (1) be proportional to pressure or (2) vary linearly in the axial direction.
(g) Sleeve and shaft assumed to have the same modulus of elasticity.

Given

L = length of seal
R = seal radius
ϵ = seal eccentricity ratio (0-1)
C_o= minimum seal clearance (estimated from ambient pressure)
C_1= maximum seal clearance (estimated from sealed pressure)
P_s= sealed pressure
ΔP= seal pressure differential
μ_o= viscosity at base conditions
B = pressure coefficient of viscosity, psi^{-1}
U = shaft speed ($R\omega$)

Calculate the following:

(a) Isothermal Flow Rate

$$Qo = \left[(\pi\ D c_o^3 \Delta\ P)/(12\ \mu_o L)\right]\ (1 + 1.5\ \epsilon^2) \tag{31}$$

(b) Isothermal Torque

$$To = (2\pi\ R^2 \mu_o\ U\ L)/\ (C_o \sqrt{1-\epsilon^2}) \tag{32}$$

(c) $$M = \beta\ \rho s \tag{33}$$

$$\gamma = (C_1 - C_o)/C_o \tag{34}$$

$$A = (2p\,sR)/(\epsilon\ C_o) \tag{35}$$

(d) Obtain Q(A) or Q(γ) from Figure 10.

(e) Obtain T from Figure 11.

(f) Corrected Flow Rate

$$Q_z = Q \cdot Q_o \tag{36}$$

(g) Corrected Torque

$$T_\theta = T \cdot T_o \tag{37}$$

4. Compressible Flow

An estimate of the compressible flow leakage rate through a bushing seal may be obtained by approximating the axial flow acceleration effect, i.e.

$$\frac{dP}{d_z} = \left(\frac{dP}{d_z}\right)_{friction} + \left(\frac{dP}{dz}\right)_{acceleration} \tag{38}$$

The friction drop is most easily estimated from [4],

$$\left(\frac{dP}{dz}\right)_{friction} = -\frac{\bar{\omega}\ \mu}{C^2 G_z} \tag{39}$$

$$= -\frac{Q\ \mu}{\pi\ D c^3\ G_z} \tag{40}$$

$$= -\frac{\mu\ W}{\pi\ D c^3\ G_z} \cdot \upsilon\ ;\ v = \text{specific volume} \tag{41}$$

where G_z is obtained from Figure 5 in the viscoseal chapter.

The acceleration pressure drop can be estimated as follows:

$$\frac{dP}{dz}\ \text{acceleration} = -\frac{\rho\ \bar{\omega}}{g}\ \frac{d\ \bar{\omega}}{d\ z} \tag{42}$$

$$= -\frac{W^2}{g(\pi\ Dc)^2}\ \frac{dv}{dz} \tag{43}$$

Substituting equations (39) and (41) into (36) gives

$$\frac{dP}{dz} = -\frac{\mu\ W}{\pi\ D\ C^3 G_z} \cdot v + \frac{W^2}{g(\pi\ Dc)^2}\ \frac{dv}{dz} \tag{44}$$

where the mass flow rate, W, must be a constant.

Since the specific volume and pressure are related the flow depends upon this relationship, i.e.

$$P\ v = RT = \text{const}\ ;\ \text{isothermal flow} \tag{45}$$

$$P\ v^\gamma = P_1\ v_1^\gamma = \text{const; adiabatic flow.} \tag{46}$$

Solution of (42) can be carried out stepwise using the following form of equation (42)

$$\Delta Pj = \frac{\mu\ W}{\pi\ D\ c^3} \cdot \left(\frac{v}{G_z}\right)_j \Delta Lj + \frac{W^2}{g\ (\pi\ D\ c)^2}\ \Delta vj\ . \tag{47}$$

In order to reduce the trial-and-error calculations of equation (45), several estimates can be made. The first is, of course, to neglect the inertia terms and estimate the flow by equations (17), (19) or (21). The second estimate, or in this case, limiting flow, is based on choked or critical flow at the exit of the seal. The critical flow rate is given by

$$W^2_{crit} = -\ g\ (\pi Dc)^2 \cdot \frac{dP}{dv}_2$$

$$= g\ (\pi Dc)^2\ \frac{P_1\ \upsilon_1^{\ \alpha}}{v_2^{\ 1+\gamma}} \tag{48}$$

The third estimate can be obtained by averaging the friction term. To do this, equation (42) is rearranged as follows,

$$\frac{dP}{\upsilon} = -\frac{\mu\ W}{\pi\,D\ c^3 G_z}\ dz + \frac{W^2}{g\ (\pi\ Dc)^2}\ \frac{dv}{v} \tag{49}$$

For isothermal flow this equation becomes

$$\int_{P_1}^{P_2} \frac{PdP}{(RT_1)} = -\frac{\mu W}{\pi Dc^3} \int_0^L \frac{dz}{G_z} + \frac{W^2}{g(\pi Dc)^2} \int_{P_1}^{P_2} \frac{dP}{P} \qquad (50)$$

or

$$\frac{\ln P_1/P_2}{g(\pi Dc)^2} W^2 + \frac{\mu L}{\pi D c^3 \bar{G}_z} W - \frac{(P_1^2 - P_2^2)}{2 R T_1} = 0. \qquad (51)$$

Solving for the flow rate gives

$$W = -\left[\frac{\mu L g (\pi Dc)^2}{2\pi Dc^3 \bar{G}_z \ln P_1/P_2}\right] + \left\{ [\quad]^2 + \frac{(P_1^2 - P_2^2) g (\pi Dc)^2}{2 RT_1 \ln P_1/P_2} \right\}^{1/2} \qquad (52)$$

If the friction loss dominates, which it most likely will, equation (50) can be approximated as follows,

$$W \cong \frac{\pi D c^3 (P_1 - P_2)\bar{G}_z}{\mu L} \cdot \frac{(P_1 + P_2)}{2 P_1 v_1}. \qquad (53)$$

For adiabatic flow equation (47) becomes

$$\int_{P_1}^{P_2} \frac{P^{1/\gamma} dP}{P_1^{1/\gamma} v_1} = -\frac{\mu W}{\pi Dc^3} \int_0^L \frac{dz}{G_z} + \frac{W^2}{g(\pi Dc)^2} \int_{P_1}^{P_2} \frac{dP}{P} \qquad (54)$$

or

$$\frac{\ln P_1/P_2}{g(\pi Dc)^2} W^2 + \frac{\mu L}{\pi D c^3 G_z} W - \frac{\left(P_1^{\frac{1+\gamma-\alpha}{\gamma}} - P_2^{\frac{1+\gamma-\alpha}{\gamma}}\right)}{\left(\frac{1+\gamma}{\gamma}\right) v_1 P_1^{1/\gamma}} = 0 \qquad (55)$$

and

$$W = -\left[\frac{\mu L g (\pi D c)^2}{2\pi Dc^3 \bar{G}_z \ln P_1/P_2}\right] + \left[[\quad]^2 + \frac{\left(P_1^{\frac{1+\gamma}{\gamma}} - P_2^{\frac{1+\gamma}{\gamma}}\right) g (\pi Dc)^2}{\frac{1+\gamma}{\gamma} v_1 P_1^{1/\gamma} \ln P_1/P_2} \right]^{1/2} \qquad (56)$$

$$= \frac{\pi D c^3 \bar{G}_z}{\mu L} \cdot \frac{P_1^{\frac{1+\gamma}{\gamma}} - P_2^{\frac{1+\gamma}{\gamma}}}{\frac{1+\gamma}{\gamma} v_1 P_1^{1/2}} \quad \text{for small inertia effects} \qquad (57)$$

If G_z varies over a wide range and it is desired to take this into account, equation (45) can be solved stepwise as follows:

(a) Assume a value for W

(b) Choose an increment size for $\Delta Pj = (P_1 - P_2)/N$

(c) Calculate P at each station from P_1 and ΔPj.

(d) Obtain v and G_z at each of these stations.

(e) Calculate the ΔLj's for each station.

(f) If when P_2 is reached $\sum_j \Delta L j \neq L$, then a new value for W must be assumed and the procedure repeated.

(g) If a zero or negative value is obtained for $\Delta L j$, a critical flow condition is indicated. If this occurs at some point $z < L$, then the assumed value for W must be reduced. If it occurs at $z = L$ then this is the actual flow rate and a critical flow condition exists at the end of the seal (it cannot occur within the seal unless a converging-diverging flow area exists).

The calculated procedure is indicated in Table 1 and possible calculated pressure curves are shown in Figure 12.

Table 1

Compressible Flow Calculation

$$\Delta L_j = \left[\Delta P_j - \frac{W^2 \Delta v_j}{g(\pi D c)}\right] \cdot \left[\frac{\mu W}{\pi D c^3} \frac{v}{G_z}\right]_j$$

assume W and ΔP_j

Station, n	$P_j=P_1+n\Delta P_j$	v_j	Re	G_{zj}	ΔL_j	$z = \sum_j \Delta L_j$
0	P_1	v_1	-	-	-	0
1	-	-	-	-	ΔL_1	ΔL_1
2	-	-	-	-	ΔL_2	$\Delta L_1 + \Delta L_2$
-	-	-	-	-	-	-
-	-	-	-	-	-	-
N	P_2	v_2	-	-	ΔL_N	L

5. Flashing, Two-Phase Flow

The easiest way to estimate the leakage in this case is to use equation (45) and the calculational procedure outlined in Table 1, assuming adiabatic, homogeneous flow. This is not particularly accurate but most likely is sufficient for most cases rather than using a more complicated approach. For this procedure the friction is estimated using the viscosity for water and the specific volume based on the homogeneous assumption, i.e.

$$v = v_f + \alpha\, v_{fg} \quad , \ ft^3/lbm \tag{58}$$

$$h_1 = h_f + \alpha h_{fg}, \ Btu/lbm^{o}F \tag{59}$$

or

$$v = v_f + \left(\frac{h_i - h_f}{h_{fg}}\right) v_{fg}. \tag{60}$$

B. Friction Power Loss

The friction torque and power loss can be given in terms of coefficient of friction

$$C_E = \frac{\tau}{1/2\, \rho \, \frac{v}{Z}^2} \tag{61}$$

where the torque is

$$T = \tau AR = \tau\ (\pi DL)\ D/2$$

$$= \tau \frac{\pi}{2} D^2 L \tag{62}$$

and the power loss is

$$q = T\omega\ . \tag{63}$$

Using the approach of Ng and Pan [5] the friction coefficient is given in terms of the Couette friction coefficient shown in Figure 13 of the viscoseal chapter, i.e.

$C_E = C_c$; (Figure 13, Chapter)

$$= 8/Re_\omega \ ; \ Re_\omega < 200\text{-}600 \ , \text{ laminar flow} \tag{64}$$

$$= 0.064\ Re_\omega^{-0.25} \ ; \ Re_\omega > 600\text{-}2000, \text{ turbulent flow.} \tag{65}$$

where

$$Re_\omega = Uc/\nu \quad ; \quad U = R\omega \tag{66}$$

Yamada [7] gives similar results, i.e.

$$C_c = 8/Re_\omega \quad ; \text{ laminar flow} \tag{67}$$

$$= 0.0606\ Re_\omega^{-0.24} \ ; \text{ turbulent flow.} \tag{68}$$

Yamada shows experimental results for no through flow, $Re_z = 0$, (Figure 13) that indicates the equation underestimates the actual friction in the turbulent regime. The effects of Taylor vorticies are also shown in Figure 8.

To account for the effects of through flow Yamada 7 gives the following expression,

$$C_c = 0.0606\ Re_\omega^{-0.24} \left[1 + \left(\frac{8}{7}\right)^2 \left(\frac{2\ Re_z}{Re_\omega}\right)^2\right]^{0.38} \tag{69}$$

where

$$Re_z = \bar{w} c/\nu. \tag{70}$$

This expression also tends to underestimate the friction torque. Yamada's experimental curves show a dependence on the clearance ratio, c/R, as well. These curves are somewhat complex and his measurements are for $0.0136 \leq c/R \leq 0.226$, whereas for bushing seals a ratio of $c/R = 10^{-3}$ is more common. Thus it is felt that the above expressions can be used for estimating the torque and power loss, realizing that the estimates are somewhat low. Note, however, from Figure 13 that as c/R becomes smaller the predictions become better.

C. Materials [1]

Material selection for bushing seals is much the same as for journal bearings since in many respects that is what it is. The close clearance, possibility of rubbing, dirt and thermal expansion problems all play a role in the selection process.

Conditions	Material
General	Carbons soft bearing materials
Moderate Temperatures	Carbons soft bearing materials (babbits)
Higher Temperatures	Carbons bronzes aluminum alloys
Floating Bushings (bearing contact loads minimized)	Hard, score resistant materials carbon soft bearing materials
Air Seals Water Seals	carbon hard materials
Thermal Expansion Problem	materials having similar coefficients of thermal expansion carbon shrunk into hcusing

D. Special Design Problems

Structural Problems

High pressure seals require heavier cross-sections and higher strength construction.

Shrink fits and/or some keying device may be needed to prevent rotation of the stationary member.

Flashing Flow

This condition may leave solid material in the seal clearance. It may be compensated for by introducing a flush or injection of cooler liquid

at the high pressure end of the seal. This is particularly effective when used with a multiple floating ring seal and the cooler liquid is injected between two of the rings.

Viscous Heat Generation

When considering the differential thermal expansion problem, it may be necessary to include the effects of heat generation within the seal due to the viscous action of the sealed fluid.

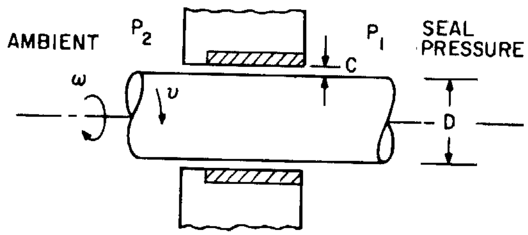


Figure 1 Fixed Bushing

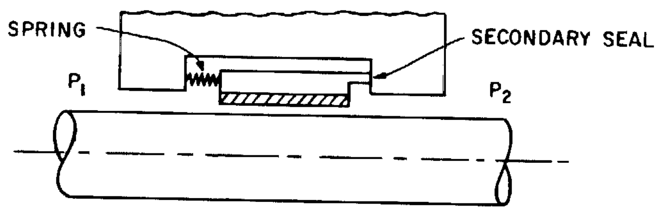


Figure 2 Single Floating Bushing

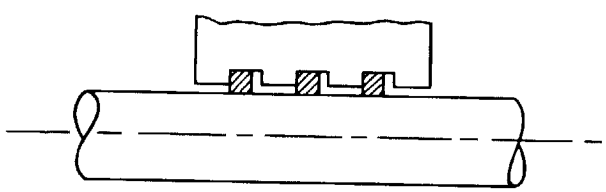


Figure 3 Floating Ring Seal

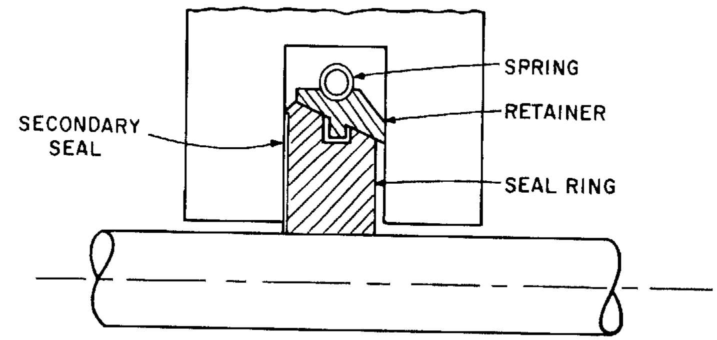


Figure 4 Segmented Ring Seal [1]

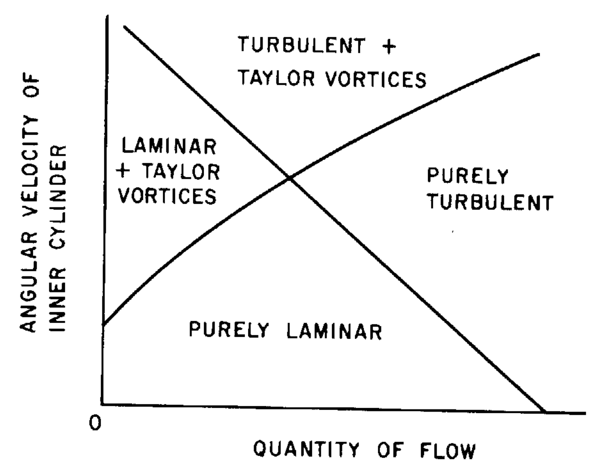


Figure 5 Flow Regimes [2]

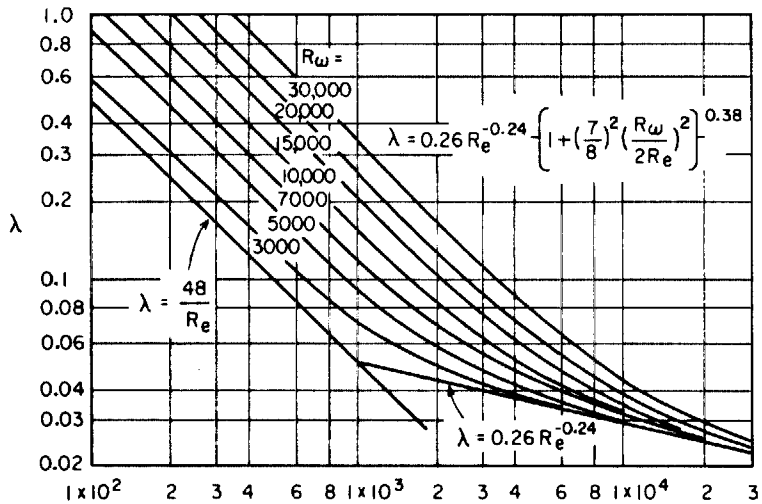


Figure 6
Theoretical Curves of $\tau - R_e$ when R_ω is constant [2]

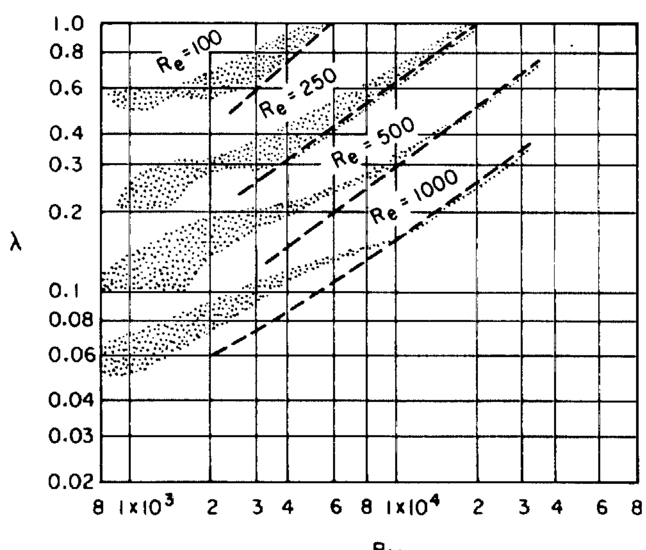


Figure 7
Relation between τ and R_ω for various Re [2]

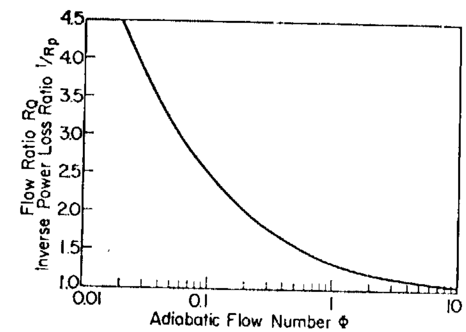


Figure 8 Flow ratio versus adiabatic flow number [8]

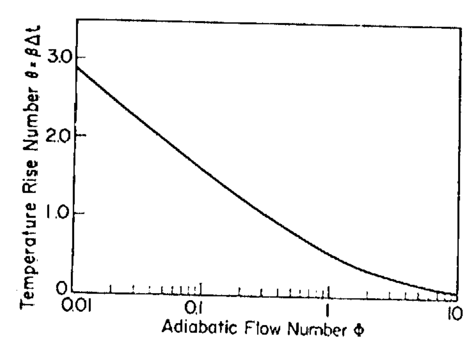


Figure 9 Dimensionless seal temperature rise versus adiabatic flow number [8]

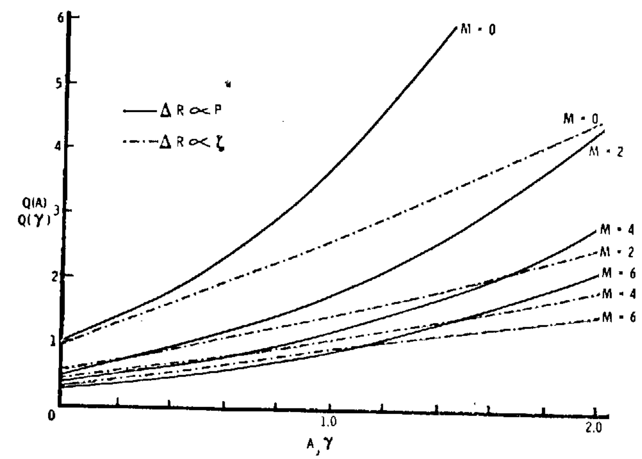


Figure 10 Flow Curves [9]

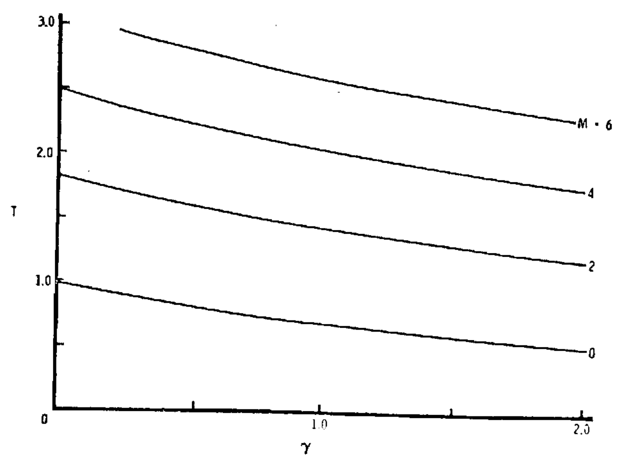


Figure 11 Torque Curves [9]

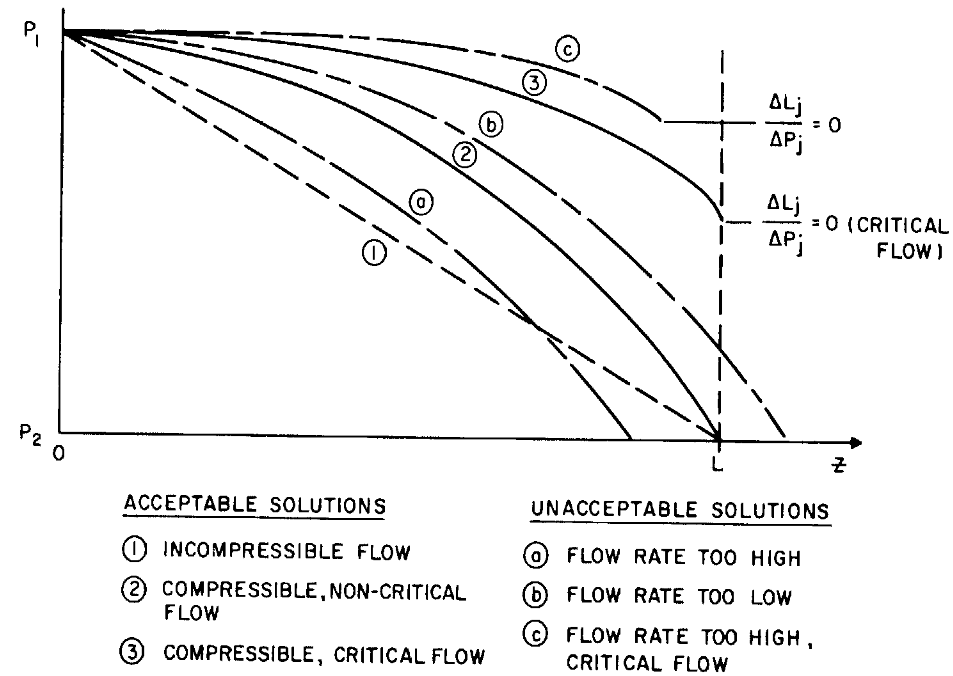


Figure 12 Pressure Profiles in a Bushing Seal

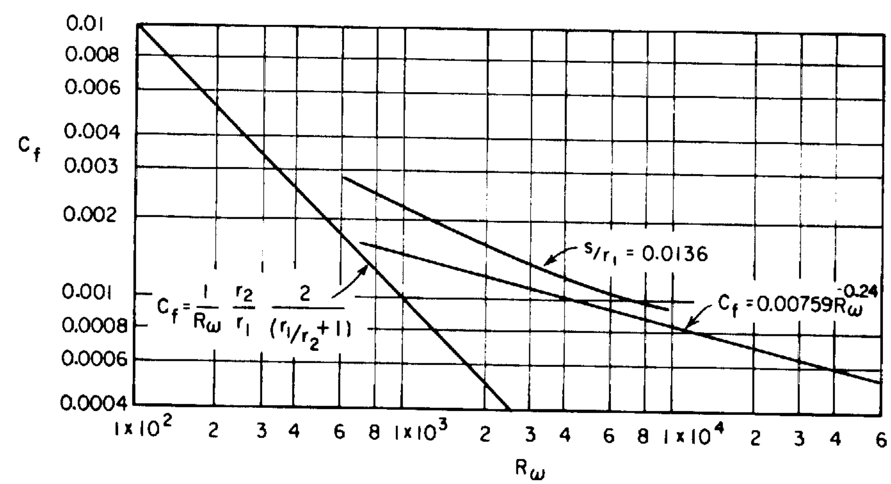


Figure 13 Relation between C_f and R_c with no axial flow and no grooves [7]

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Nomenclature

A	=	leakage flow area
B	=	pressure coefficient of viscosity
C_c	=	Couette friction factor, for torque
C_p	=	specific heat
c	=	radial clearance
D	=	seal diameter
e	=	eccentricity
f	=	friction factor, for leakage
G_z	=	turbulent flow correction factor
H_p	=	horsepower
h	=	enthalpy
L	=	seal length
N	=	shaft rpm
$P_1=P_s$	=	sealed pressure
P_2	=	ambient pressure
ΔP	=	$P_1 - P_2$
Q	=	leakage rate, volumetric
R	=	seal radius
Re_ω	=	rotational Reynolds number
Re_z	=	through flow Reynolds number
τ	=	torque and temperature
U	=	shaft surface speed
ν	=	specific volume
W	=	leakage rate, mass
$\bar{\omega}$	=	average axial flow velocity
z	=	axial coordinate
α	=	lb of steam/lb of mixture

Nomenclature (continued)

β	=	thermal coefficient of viscosity
γ	=	C_p/C_υ = specific heat ratio
ϵ	=	eccentricity ratio
μ	=	viscosity
ν	=	kinematic viscosity (μ/ρ)
ρ	=	density
τ	=	shear stress
ω	=	shaft velocity

D. VISCOSEAL

Viscoseal

(Spiral Grooved Screw Seal)

I. Introduction

Operating Principle

The noncontacting viscoseal, Figure 1, operates on the principle of hydrodynamic pumping by means of helical grooves cut in either the shaft or the housing. When operating at zero leakage it is essentially a screw pump functioning at shut-off conditions.

Relative motion between the shaft and housing induces viscous, Couette drag flow in the helical grooves. Thus the names, viscoseal or screw seal. This flow is opposed by pressure induced Poiseuille flow in the grooves and over the lands.

Advantages

The viscoseal offers the advantages of zero leakage and noncontacting surfaces for long life and reliable service. It may also act as a pump or a controlled leakage seal. Its operation is simple and the design theory fairly well developed, with some notable exceptions.

Disadvantages

The viscoseal requires close clearances for effectiveness with accompanying close tolerances, minimum radial motion, and power loss (heat generation) that is high enough to be an important design consideration. The pumping action of the seal depends upon the fluid viscosity, in the laminar regime, and on the fluid viscosity and density in the turbulent regime. Therefore, its application is usually limited to fluids of high viscosity and/or density. Although it may find some use as purely a gas seal, it usually is used with a buffer fluid in this application, as shown in Figure 3.

Design Analysis

The viscoseal analyses available in the literature are, in general, constrained by the following assumptions:

1. The fluid is homogeneous.

   (Actually over much of the operating range of a liquid viscoseal, gas will be ingested into the liquid at the gas-liquid interface. The consequences are: (a) deviation of performance from calculations,

   (b) possible gas ingestion to the sealed cavity, and (c) influence on the stability of the gas-liquid interface and eventual "seal breakdown." Cavitation is also neglected.)

2. Gas-liquid interface location does not vary circumferentially.

   (When the shaft is eccentric with the housing the interface will change location circumferentially causing a radial force on the shaft. The flooded length will be longer on the wide clearance side of the seal.)

3. End effects are neglected.

   (At the end of the viscoseal and at the gas-liquid interface, the pressure profile across the land-groove combinations must smooth out to a constant pressure. This "end effect" results in a reduced seal effectiveness, which is usually a small correction (<5%) for most viscoseals.)

4. Convective inertia effects are neglected.

   (Because of the abrupt changes in flow areas in the land-groove combinations, convective inertia effects tend to be of importance even in the laminar flow regime. These effects have started to be investigated just recently. The result is that theoretical viscoseal predictions tend to underestimate the seal effectiveness.)

5. Groove sidewall effects are neglected.

   (It is assumed that the ratio $\omega_{g/h}$ is so large that the effect of the groove side walls is not significant in the analysis of the groove flow.)

6. Flow regimes are either:

   (a) Laminar with no inertia terms of any kind considered, i.e. "creeping flow."

   (b) Turbulent with no convective inertia terms considered. Turbulence is taken into account by including eddy viscosity effects by means of turbulent flow coefficients given as functions of Reynolds number. They are continuous and may be applied through the transition region to laminar flow.

7. Constant fluid properties, including incompressible flow.

   (Heat generation in the fluid could change the fluid viscosity. However, cooling will be primarily to the seal walls rather than by circulation of flow. Therefore, this may not be a critical assumption. A compressible analysis of the problem has not been made but is within the capability of the available techniques.)

Applications

Liquid-Gas Seal

This is the most commonly discussed type of application. In this case the liquid to be sealed is under pressure and is sealed from gas, vapor or vacuum ambient. There is a tendency for gas ingestion which is one of the major drawbacks for the viscoseal.

Gas-Liquid-Gas Seal

This is an accepted way to seal a gas from a gas ambient. It has the advantage of high liquid viscoseal effectiveness and preventing contact and mixing of the two gases. The liquid recirculation also provides some cooling.

Liquid-Liquid or Gas-Gas Seal

This application is seldom seen in practice. One of the potential problems is mixing of the two fluids. Little is known about this operating mode. Operating with gases alone is practical but not nearly as effective as with liquids because of the lower viscosity.

A special case of this application is when the fluids (or gases) are the same as with pump interstage seals. No mixing problem is present but it is not likely that the seal will operate at zero leakage. This is due to the fact that the sealing length is constant and not a variable. Therefore, depending upon the shaft speed and the pressure differential the seal will either leak or pump.

Operating Modes

For the general application the three possible operating modes of the viscoseal should be considered, i.e. zero leakage, leaking, and pumping.

Zero Leakage

This mode of operation is the one most often considered in the literature. The application described is usually that of a liquid-gas seal with a variable sealing length (or wetted length). Under some conditions (not precisely established as yet), "seal breakdown" occurs. This is not a net seal leakage as such, but rather a liquid-gas interface instability that allows droplets to break away from the surface and leak out. From a theoretical point of view this is still the zero leakage mode. This is true for the "gas ingestion" phenomenon also. It can occur during the so called zero leakage mode. In the extreme conditions it actually results in gas being pumped into the seal cavity.

Leaking Seal

The viscoseal will have a net leakage when the pumping effect is not sufficient to offset the pressure induced flow. In this operating mode it is reasonable to compare the viscoseal leakage with the leakage from a bushing seal of comparable dimensions. This is done because it is possible, under certain conditions of pressure, speed, dimensions and fluid properties, for the grooves

in a viscoseal to increase the leakage over that of a bushing seal. This type of comparison has been made for the viscoseal operating in the laminar flow regime. No comparable study has been made for turbulent flow. In fact, most analyses consider only the zero leakage seal coefficient and, although, relationships for the leaking or pumping seal are implicit, they are not specifically given.

Pumping Seal

Although the viscoseal was originally viewed as the "screw pump", more recently studies have looked at it only from the zero leakage point of view. In applications where a liquid exists on both sides of the viscoseal, it is quite possible for a net pumping to occur since the full length of the seal is active. It may be argued that this mode is not particularly important. There certainly is not very much in the literature concerning it, at least in the turbulent regime. Here again, the basic analyses available can be applied to this case. The published results are just not slanted in this direction.

II. Viscoseal Design

A. General

Since most of the viscoseal analysis has been concerned with the zero leakage, homogeneous flow mode, it seems reasonable that these results should be used to arrive at a roughly optimum seal geometry. The discussion in this section will, therefore, follow the following format:

1. Formulas for zero leakage seal coefficients and friction coefficients.
2. Consideration of the leaking and pumping modes of operation.
3. Comparison with experimental results.
4. Discussion of observed phenomena not predicted by the theory.
5. Design procedure based on the above.

B. Zero Leakage Sealing Coefficient

Laminar Flow (Creeping Flow, i.e. Convective inertia effects neglected)

Fisher and Stair [3] state that the laminar sealing coefficient given by Boon and Tal [6] shows the best agreement with experiment. This expression was used by Hodgson and Milligan [4] and derived by Vohr and Chow [5] for laminar flow in herringbone-grooved gas bearings.

$$\psi = \frac{\Delta P h_r^2}{\mu U L} = \frac{\left[6 \tan\Phi \cdot \alpha (1-\alpha)\right] (\Gamma-1) \quad (\Gamma^3-1)}{\left[1 + \tan^2\Phi\right] \Gamma^3 + \left[\tan^2\Phi \cdot \alpha (1-\alpha)\right] (\Gamma^3-1)^2} \tag{1}$$

Similar expressions, including the effects of shaft eccentricity, are given by McGrew and McHugh [1]

$$\psi = \frac{\left[6 \tan\Phi \cdot \alpha (1-\alpha)\right] \beta^2}{\left[1 + \tan^2\Phi\right] \beta^3 (1+\beta) E + \left[\tan^2\Phi \cdot \alpha (1-\alpha)\right] (1+\beta)}$$

$$= \frac{\left[6 \tan\Phi \cdot \alpha (1-\alpha)\right] (\Gamma-1^2)}{\left[1 + \tan^2\Phi\right] \Gamma E + \left[\tan^2\Phi \cdot \alpha (1-\alpha)\right] \Gamma (\Gamma-1)^3} \tag{2}$$

where

$$E = 1 + 1.5 E^2 \tag{3}$$

is the eccentricity correction factor, and by McHugh [2]

$$\psi o = \frac{\left[6 \tan\Phi \cdot \alpha (1-\alpha)\right] \beta^2}{\left[1 + \tan^2\Phi\right] \beta^3 (1+\beta) E + \left[\tan^2\Phi \cdot \alpha (1-\alpha)\right] (1+\beta) (1+3\beta)}$$

$$= \frac{\left[6 \tan\Phi \cdot \alpha (1-\alpha)\right] (\Gamma-1)^2}{\left[1 + \tan^2\Phi\right] \Gamma E + \left[\tan^2\Phi \cdot \alpha (1-\alpha)\right] \Gamma (\Gamma+2) (\Gamma-1)^2} \tag{4}$$

These authors optimized the sealing coefficient with respect to the parameters, Φ, α and Γ (or β), and obtained the results given in Table 1.

Table 1

Laminar Flow Optimum Sealing Parameters

Optimized Sealing Parameter	Concentric Shaft (E=1.0) Ref [1]	Ref [2]	Ref [4]	Eccentric Shaft (E=2.5) [1]	[2]	[4]
Clearance Ratio $\beta = c/h$ ($\Gamma = \frac{hg}{hr}$)	0.36 (3.78)	0.277 (4.60)	0.37 (3.70)	0.31 (4.225)	0.24 (5.16)	no data
Groove-Land Ratio $\alpha = Wg/(Wg+Wr)$	0.50	0.50	0.50	0.50	0.50	
Helix Angle Φ	21.6°	11.9°	15°	26.2°	15.2°	
Sealing Coefficient ψ	0.61	0.449	0.55	0.36	0.275	

For purposes of comparison with the turbulent flow sealing coefficient it is desirable to make the following approximations.

$$1 + \tan^2 \approx 1 \quad , \Phi \leq 15^\circ$$

$$(\Gamma^3-1) \approx \Gamma^3 \quad , \Gamma > 3.6$$

The laminar flow sealing coefficient as given by (1) now reduces to

$$\psi = \frac{\left[6 \tan\Phi \cdot \alpha (1-\alpha)\right] (\Gamma-1)}{1 + \left[\tan^2\Phi \cdot \alpha (1-\alpha)\right] \Gamma^3} \tag{5}$$

<u>Turbulent Flow</u> (convective inertia effects neglected)

One of the most outstanding turbulent flow analyses for viscoseals is that given by Vohr and Chow [5]. For the <u>concentric</u> viscoseal the turbulent flow sealing coefficient given by them is

$$\psi = \left[0.5 \tan\Phi \cdot \alpha (1-\alpha)\right] (\Gamma-1) \left(\Gamma^3 G_{zg} - G_{zr}\right) \div$$
$$\Big[\Gamma^3 G_{zr} G_{zg} + \tan^2\Phi \cdot \alpha (1-\alpha) G_{zr} G_{xr} + \tan^2\Phi \cdot (1-\alpha)^2 \Gamma^3 G_{zr} G_{xg}$$
$$+ \tan^2\Phi \cdot \alpha^2 \Gamma^3 G_{zg} G_{xr} + \tan^2\Phi \cdot \alpha (1-\alpha) \Gamma^6 G_{zg} G_{xg}\Big], \tag{6}$$

which is the same expression obtained by Pape and Vrakking [7]. For laminar flow this equation reduces to (1).

If

$$\Phi \leq 15^\circ \quad , \quad \Gamma > 3.6 \ , \ 0.2 < \alpha < 0.8$$

the following approximate version of (6) is given in [7]

$$\psi = \frac{\left[0.5 \tan\Phi \cdot \alpha (1-\alpha)\right] (\Gamma-1)}{G_{zr} + \left[\tan^2\Phi \cdot \alpha (1-\alpha)\right] \Gamma^3 G_{xg}} \tag{7}$$

The authors' claim an accuracy within 5% and that these conditions cover the total range of interest for a viscoseal.

For the viscoseal, the turbulent correction factors, G_x and G_z, are given as a function of the Reynolds number, (Vh/υ), alone. Vohr and Chow [5], and Ng and Pan [10], give the values of G_x and G_z, in Figure 5, as predicted by a linearized theory. These factors hold only for turbulent flows where the Poiseuille flow component is small compared to the Couette flow component.

By comparing the laminar flow sealing coefficient (5) and the turbulent flow sealing coefficient (7) it can be seen that as

$Re \to 0$ (laminar, creeping flow).

$$G_x, G_z \to \frac{0.5}{6} = \frac{1}{12} = \underline{\underline{.0833}},$$

and (6) reduces to (1), [7].

Vohr and Chow [5] optimized the seal parameters as a function of Reynolds number, Figure 6. They also calculated the sealing coefficient as a function of eccentricity, Figure 7, and misalignment, Figure 8, for the optimum geometry given in Figure 6. Unfortunately these calculations had to be performed by numerical solution of the authors' [5] general differential equations and no closed form solution is given.

Eccentric Seal

It is interesting to note that the laminar flow eccentricity correction factor used by McGrew and McHugh [1,2] in equations (2) and (4), was applied only to the term representing the leakage over the lands (ridges). Vohr and Chow's [5] analysis takes into account the distributed effect of eccentricity on land and groove flow. Their results in Figure 7 for laminar flow (Re=1) do not indicate the same type of eccentricity correction as given in (2) and (4). This can be demonstrated as follows:

For Re = 1

Optimum geometry: $\Phi = 15.80$, $\alpha = .50$, $\Gamma = 3.7$

$$\psi = \frac{6 \tan 15.80 \quad (.50)\ (.50)\ (3.7-1)}{1 + \tan^2 15.80\ .\ (.50)\ (.50)\ (3.7)^3}$$

$$= \frac{1.148}{1 + 1.01} = \underline{\underline{.57}} \qquad \text{(Concentric, } E = 1) \qquad (8)$$

For E = .80

$$E = 1 + 1.5\ E^2 = 1.96$$

and according to equations (2) and (4) should be applied as follows:

$$\psi\ E = .8 = \frac{1.148}{E + 1.01} = \frac{1.148}{2.97} = \underline{\underline{.386}} \qquad (9)$$

From Figure 7 we see that the results are approximately .55 and .41. If it were assumed that the eccentricity correction could be included in this fashion, i.e.

$$\psi = \frac{\left[.5 \tan \Phi\ .\ \alpha\ (1-\alpha)\right]\ (\Gamma-1)}{Gzr\ E + \left[\tan^2 \Phi\ .\ \alpha\ (1-\alpha)\right]\ \Gamma^3\ Gxg} \qquad (10)$$

where

$$E = 1 + a\ E^2,$$

then the following values of "a" can be calculated from Figure 7.

Table 2

Optimum Geometry

Re_r	Φ	α	Γ (β)	Re_g	Gzr	Gxg	$\psi\varepsilon$=0(Calc.)	$\psi\varepsilon$=.8	a
1	15.8^o	.50	3.7(.37)	3.7 (.37)	.0833	.0833	.55(.57)	.41	1.24
10^3	15.4^o	.50	6.0(.20)	6.0×10^3	.07	.021	1.15(1.10)	.86	.97
$5x10^3$	14.0^o	.50	8.3(.137)	41.5×10^3	.036	.005	2.7 (2.84)	2.3	.83
10^4	12.0^o	.50	9.3(.12)	9.3×10^4	.023	.0027	4.6 (4.64)	3.7	.81

Vohr and Chow [5] calculated the results in Figure 7 by assuming that the liquid gas interface was not inclined with respect to the axis due to the eccentricity. They note that the flooded length would be longer on the wide clearance side of the seal and shorter on the narrow clearance side, but their analysis is not capable of treating this complex situation. McGrew and McHugh [1,2] make this same assumption.

Optimum Geometry

The optimum geometrics given in Table 1 are, of course, based on theory. From a practical point of view it will be seen later that due to convective inertia effects it may be desirable to increase α above 0.5. Therefore, three questions arise regarding the use of these "optimum" geometrics in setting the viscoseal design. They are: (1.) What is the variation in ψ, about the optimum, with respect to Reynolds number, (2.) What is the variation in ψ, about the optimum, with respect to changes in geometry, and (3.) How do these theoretical values of ψ compare with experiments. The first two subjects will be considered next and the third discussed in another section.

(1.) Variation in ψ with Re.

In Table 3 and Figure 9 the variation in ψ with Reynolds number is shown for the three cases where the geometry has been optimized at Re=1, 10^3 and 10^4 respectively. If the range of Reynolds number is known for a particular application, Figure 9 will help in choosing the desired seal geometry.

Table 3

1. $Re_r = 1$, $\Phi = 15.8^o$, $\alpha = .50$, $\Gamma = 3.7$

Re_r	Re_g	G_{zr}	G_{xg}	$\psi = 0.0955/(G_{zr}+1.01G_{xg})$
1	3.7	.0833	.0833	.570
10^3	3.7×10^3	.0700	.0300	.952
5×10^3	1.85×10^4	.0360	.0095	2.095
10^4	3.7×10^4	.0230	.0055	3.350
10^5	3.7×10^5	.0045	.0009	17.650

2. $Re_r = 10^3$, $\Phi = 15.4^o$, $\alpha = .50$, $\Gamma = 6.0$

Re_r	Re_g	G_{zr}	G_{xg}	$\psi = 0.1725/(G_{zr}+4.125G_{xg})$
1	6	.0833	.0833	.403
10^3	6×10^3	.0700	.0210	1.100
10^4	6×10^4	.0230	.0038	4.450

3. $Re_r = 10^4$, $\Phi = 12^o$, $\alpha = .50$, $\Gamma = 9.3$

Re_r	Re_g	G_{zr}	G_{xg}	$\psi = 0.22/(G_{zr}+9.07G_{xg})$
1	9.3	.0833	.0833	.262
10^3	9.3×10^3	.0700	.0160	1.020
10^4	9.3×10^4	.0230	.0027	4.640

(2.) Variation in ψ with geometric parameters, Φ, α and Γ (β).

In general it can be seen from Figures 10 to 12 that the sealing coefficient, ψ, does not vary rapidly with the geometrical parameters and the optimums are relatively flat. Therefore, the choice of design parameters seems to be more dependent upon the Reynolds numbers to be encountered in the application.

<u>Power Loss</u>

The friction coefficient, as given by Pape and Vrakking 7 is

$$C_E = \frac{\tau}{1/2\ P(\frac{U}{Z})^2} = \frac{64}{\pi} \cdot \frac{T}{P\omega^2 D^4 L}$$

$$= \underbrace{\alpha\ C_{cg} + (1-\alpha)\ C_{cr}}_{\text{Couette component of friction}} + \underbrace{\frac{4\ \alpha\ \tan\Phi\ .\ (\Gamma-1)}{Re_r} \cdot \psi \cdot \frac{\Gamma^3\ G_{zg} - G_{zr}}{\Gamma^3 \tan^2 G_{xg} + \frac{\alpha}{1-\alpha} G_{zr} + \Gamma^3 G_{zg}}}_{\text{Poiseuille component of friction (usually small)}} \qquad (11)$$

$$\cong \alpha\ C_{cg} + (1-\alpha)\ C_{cr}. \qquad (12)$$

The Couette friction coefficients are given in Figure 13. For laminar flow the wall shear stress is

$$\tau = \frac{\mu U}{h} \qquad (13)$$

Therefore

$$C_c = \frac{8}{(PUh/\mu)} = \frac{8}{Re} \qquad (14)$$

The viscoseal friction coefficient for laminar flow is, then

$$C_\epsilon \simeq \frac{8\,\alpha}{Re_g} + \frac{8(1-\alpha)}{Re_r}$$

$$= \frac{8}{Re_r}\left[\frac{\alpha}{\Gamma} + 1-\alpha\right] \qquad (15)$$

as given in [1]. The Poiseuille component of friction has been neglected.

C. Leaking and Pumping Modes

McHugh [2] compared the leaking viscoseal to the bushing seal in the laminar flow regime. No comparable comparison has been made in the turbulent regime. Few analyses even donsider the leaking or pumping modes so most of the results are given in terms of the zero leakage sealing coefficient.

McHugh defines the "limit of effectiveness" as that condition when the viscoseal is leaking and its leakage rate is equal to that of a bushing seal with the same clearance and length. This is an interesting concept since for some applications it may be better to use a bushing seal than a viscoseal.

The turbulent lubrication theory discussed in [10,11,5,7] forms a general approach to the study of viscoseals and bushing seals, and the comparison between them. There are, however, several difficulties in doing this. First, the above mentioned theory does not take into account the effects of Taylor vortices as does Yamada [18]. Secondly, the turbulent flow coefficients given in Figure 5 are good only for the case where the axial pressure flow effect is small compared to the rotational Couette flow effects.

Yamada's [18] results can be summarized as follows:

Laminar Flow (Creeping Flow)

$$Q_{1am} = \frac{\pi\ D\ C^3\ \Delta\ P}{12\ \mu\ L} \qquad (16)$$

Turbulent Flow - No Rotation

$$Q = \frac{4.72\,\pi\ D\ C^{1.7}}{\mu^{0.136}\rho^{0.43}}\left(\frac{\Delta P}{L}\right) \qquad (17)$$

Turbulent Flow - (Rotation effects dominate i.e. $Re_c > 10^4$

$$Q = \frac{346}{Re_c^{0.76}}\ Q_{1am} \qquad (18)$$

From their turbulent analysis, Elrod and Ng [11] obtained the turbulent flow coefficients G_x and G_z such that the average circumferential and axial velocities are given by [5,7]

$$\overline{u} = -\frac{h^2}{\mu}\ G_x\frac{\partial P}{\partial x} + \frac{U}{2} \qquad (19)$$

$$\overline{w} = -\frac{h^2}{\mu}\ G_z\ \frac{\partial P}{\partial z} \qquad (20)$$

For the general case (not including Taylor vortices) the values of these coefficients may be obtained from [11]. For the concentric bushing seal $\partial P/\partial \theta = 0$ and Figure 14, taken from [11] is of interest for this case. This figure can be compared to Figure 15 taken from Yamada [18] for the bushing seal.

Comparing equations [19] and [20] we see that for either laminar Poiseuille flow or Couette flow,

$$G_z = 1/12 = 0.0833 \qquad (21)$$

as before. For turbulent flow with no rotation, from Yamada 18

$$Q = \frac{4.72\,\pi\ D\ h^{1.7}}{\mu^{0.136}\rho^{0.43}}\cdot\left(\frac{\Delta P}{L}\right) \qquad (22)$$

$$= \frac{\pi\ D\ h^3}{\mu}\cdot\left(\frac{\Delta P}{L}\right) G_z \qquad (23)$$

or $$G_z = 4.72\left(\frac{h^3}{\mu}\ \cdot\ \frac{\partial P}{\partial z}\right)^{-0.432} \quad ; \quad \frac{\partial P}{\partial z} = \frac{\Delta P}{L} \qquad (24)$$

$$= 4.72\ (\mathrm{GRAD\ P})^{-0.432}$$

This equation is compared with G_z, taken from Figure 14 [11], for the horizontal parts of the curves, i.e. where G_z is a function of (GRAD P) only. This is shown in Figure 16.

The "linearized" theory, shown in Figure 5, forms an envelope for the G_z curves. Actually, it represents the condition where the rotational Couette flow dominates and G_z is a function of Re_h only, i.e. not a function of (GRAD P). This can be compared with equation [18] derived from the work of Yamada for $Re_h > 10^4$,

$$Q = \frac{346}{Re_h^{0.76}} \cdot \frac{\pi D h^3}{12 \mu} \cdot \left(\frac{\Delta P}{L}\right) \tag{25}$$

$$= \frac{\pi D h^3}{\mu} \left(\frac{\Delta P}{L}\right) G_z \tag{26}$$

or

$$G_z = 28.8 \, Re_h^{-0.76} \tag{27}$$

This is shown in Figure 17.

For conditions where G_z is a function of Re_h and (GRAD P) both, the non-horizontal portions of the constant (GRAD P) lines in Figure 14 apply when using the Elrod, Ng 11 approach, or Figure 18 from Yamada [18]. The data in Figure 18 [18] indicates the effect of Taylor vorticies which Elrod and Ng do not consider and which most likely are not important in viscoseals, only the bushing seals.

The axial mass flux per groove-land pair, as given by Vohr and Chow 5, is

$$W^7 = -P\left[\frac{(1-\alpha)\, h_r^3}{\mu} \frac{\partial P_r}{\partial z} G_{zr} + \frac{\alpha h_g^3}{\mu} \frac{\partial P_g}{\partial z} G_{zg}\right] \tag{28}$$

For a concentric viscoseal, $\partial P/\partial \xi = \partial P/\partial \theta = 0$, and the leakage rate is

$$Q = \frac{\pi D W^7}{P} = \pi D \left\{ \frac{1}{\mu} \frac{\partial P}{\partial z} \left(-G_{zg}\, \alpha\, h_g^3 - G_{zr} (1-\alpha)\, h_r^3\right) \right.$$

$$+ \left[\frac{1}{\mu} \frac{\partial \bar{P}}{\partial z} \cot^2 \Phi\, \alpha (1-\alpha) (h_g^3 G_{zg} - G_{zr})^2 + \right.$$

$$\left. - 0.5\, \upsilon\, (h_g - h_r)\, \alpha (1-\alpha) (h_g^3 G_{zg} - h_r^3 G_{zr}) \cot^2 \Phi \right] \div$$

$$\div \left[(1-\alpha)\, h_g^3 (G_{xg} + G_{zg} \cot^2 \Phi) + \alpha\, h_r^3 (G_{xr} + G_{zr} \cot^2 \Phi)\right.$$

$$= \pi D \left\{ \frac{h_r^3}{\mu} \frac{\partial \bar{P}}{\partial z} (-\Gamma^3 \alpha G_{zg} - (1-\alpha) G_{zr}) \right.$$

$$+ \left[\frac{h_r^3}{\mu} \frac{\partial \bar{P}}{\partial z} \alpha (1-\alpha) (\Gamma^3 G_{zg} - G_{zr})^2 - 0.5\, \upsilon h_r (\Gamma - 1)\, \alpha (1-\alpha) (\Gamma^3 G_{zg} - G_{zr}) \right.$$

$$\left.\left. \times \tan \Phi \right] \div \left[(1-\alpha)\, \Gamma^3 (\tan^2 \Phi\, G_{xg} + G_{zr})\right] \right\}. \tag{29}$$

Rearranging this expression we have

$$Q = \pi D \left\{ -\frac{h_r^3}{\mu} \frac{\partial \bar{P}}{\partial z} \left[\Gamma^3 G_{zr} G_{xg} + \tan^2 \Phi\, \alpha (1-\alpha) G_{zr} G_{xr} \right.\right.$$

$$+ \Gamma^3 \tan^2 \Phi\, (1-\alpha)^2 G_{zr} G_{xg} + \alpha^2 \Gamma^3 \tan^2 \Phi\, G_{zg} G_{xr}$$

$$\left.\left. + \Gamma^6 \tan^2 \Phi\, \alpha (1-\alpha) G_{zg} G_{xg}\right] - 0.5\, \upsilon h_r\, \alpha (1-\alpha) \tan \Phi\, (\Gamma^3 G_{zg} - G_{zr}) \right\}$$

$$\div \left\{ (1-\alpha)\, \Gamma^3 (\tan^2 \Phi\, G_{xg} + G_{zg}) + \alpha (\tan^2 \Phi\, G_{xr} + G_{zr}) \right\}. \tag{30}$$

Note, that setting the leak rate equal to zero will give the sealing coefficient (6).

If the same approximation as made by Pape and Vrakking 7, in obtaining (7), is made here, then the leak rate is

$$\rightarrow Q \simeq \frac{\pi D}{(1-\alpha)} \left\{ -\frac{h_r^3}{\mu} \frac{\partial \bar{P}}{\partial z} \left[G_{zr} + \Gamma^3 \tan^2 \Phi\, \alpha (1-\alpha) G_{zg}\right] \right.$$

$$\left. - 0.5\, \upsilon h_r\, \alpha (1-\alpha) \tan \Phi\, (\Gamma - 1) \right\} \tag{31}$$

where

$$\Phi \leq 15^\circ, \quad \Gamma > 3.6, \quad 0.2\, \alpha < 0.8.$$

For laminar flow equation (31) becomes

$$Q = \frac{\pi D}{(1-\alpha)}\left(\frac{-h_r^3}{12\mu}\frac{\partial \bar{P}}{\partial \bar{z}}\left[1 + \Gamma^3 \tan^2 \Phi\, \alpha\, (1-\alpha)\right] - 0.5\, \upsilon h_r\, \alpha\, (1-\alpha)\tan\Phi\, (\Gamma-1)\right) \tag{32}$$

This expression can now be compared with that of McHugh [2],

$$Q = Q_1 + Q_p - Q_d. \tag{33}$$

The leakage across the lands is [2]

$$Q_1 = \frac{\pi\, D\, h_r^3 \Delta\, P}{12\,\mu\, L}\quad \frac{t}{ne}\quad E = \frac{\pi\, D}{(1-\alpha)} \cdot \frac{h_r^3}{12\mu}\left(\frac{\Delta P}{L}\right) E \tag{34}$$

where

$$E = 1 + 1.5\, E^2$$

is the eccentricity correction factor. It can be seen that this term compares exactly with the first term in equation (31) with exception of the E factor.

The back flow into within the groove due to the pressure gradient is [2]

$$Q_p = (1+3\beta)\frac{h_r^3}{12\,\mu}\left(\frac{\Delta P}{L}\right)\left(\frac{h}{h_r}\right)^3 \alpha\, \pi\, D \tan^2 \Phi$$

$$= \pi\, D\, \frac{h_r^3}{12\mu}\left(\frac{\Delta P}{L}\right)\alpha \tan^2 \Phi\,(\Gamma - 1)^3\,(1+3\beta). \tag{35}$$

Comparing this expression with the second term in equation (31) we see that they differ by a factor of $(\Gamma-1)^3\ (1+3\beta)$ to Γ^3. In McHugh's analysis he assumes a Poiseuille velocity profile from the bottom of the groove to the surface of the housing. However, he integrates this profile from the bottom of the groove only to the top of the groove, a distance of h. Thus the factor $(1+3\beta)h^3$ appears in this leakage term. If he had integrated from the bottom of the housing to the surface of the housing this term would be replaced by h_g^3 and the $(\Gamma-1)^3\ (1+3\beta)$ term would become Γ^3. With this change the two expressions would be identical.

The flow rate due to viscous drag alone is [2]

$$Q_d = -0.5\, \upsilon\, h\, (\kappa\, \omega_g \cos \Phi)\ /\ (1+\beta)$$

$$= 0.5\, \upsilon\, h_r\, (\Gamma-1)\, \pi\, D\, \alpha \tan \Phi\ /\ (1+\beta). \tag{36}$$

Here again McHugh integrated only over the groove depth, h, and the $(1+\beta)$ term resulted from this. Had he integrated over h_g the $(1+\beta)$ term would not appear and the expression would be identical to the third term in equation (31).

How it is possible to compare the viscoseal leakage to the bushing seal leakage, i.e.

$$Q_{VS} = \frac{\pi D h_r^3}{\mu}\,\frac{\Delta P}{L}\left[\frac{G_{gr}}{(1-\alpha)} + \Gamma^3 \tan^2 \Phi\, .\, \alpha\, G_{xg}\right] - 0.5\, \pi D \upsilon h_r \alpha \tan\Phi\, .\, (\Gamma-1) \tag{37}$$

$$Q_{Bush} = \frac{\pi D h_r^3}{\mu}\,\frac{\Delta P}{L}\quad G_{gr} \tag{38}$$

For the condition when the viscoseal is no more effective than the bushing seal a "limit of effectiveness" seal coefficient can be determined by setting the two leakage rates equal, i.e., for

$$Q_{\upsilon s} = Q_{Bush} \tag{39}$$

$$\Psi_{\nu/B} = \frac{\Delta P}{L}\,\frac{h_r^2}{\mu\upsilon} = \frac{0.5\alpha\ (1-\alpha)\ \tan\Phi\ .\ (\Gamma-1)}{\alpha\, G_{gr} + \Gamma^3\, \alpha(1-\alpha)\ \tan^2 \Phi\ .\ G_{xg}} \tag{40}$$

where $G_{gr} = G_{xg} = .0833$ for laminar flow. It should once again be mentioned that convective inertia effects have been neglected so that the leakage expression (37) would overestimate the leakage and should not be used in the limit where $\upsilon = 0$ for turbulent flow. To hold for this case the leakage expression should be reducible to a labyrinth seal equation (see section on labyrinth seal equations).

D. Experimental Verification of Theory

Figure 19, taken from [3] and [5] shows a comparison of several turbulent theories with experiment. There are other figures in [5] that show better agreement between the Vohr and Chow theory, equations (6) and (7), and experiment. Figure 19, however, serves to also compare the various theories. The theory of Vohr and Chow [5] has been used throughout this section because it represents a unified approach that can be used for either the viscoseal or the bushing seal and for either turbulent or laminar flow. It can be argued that some of the other empirical theories, shown in Figure 19, give closer agreement to some of the data. However, they do not represent as unified an approach and since there are simplifications made in all of the analyses and ultimate experimentation is required, it seems quite reasonable to use equations (1), (5), (6), (7), and (12) for design purposes.

A summary of some of the other turbulent theories is given below.

Ketola and McGrew [9]

$$\frac{Q}{\nu \omega_g h_g} = \frac{1}{2} + A_1 \psi + A_2 \tag{41}$$

where

Q = flow rate im pumping direction

$\nu = \upsilon \cos\Phi$ = component of shaft surface velocity along groove

ω_g = groove width

h_g = groove depth plus radial clearance

$\psi = \Delta P\, h_r^2/\mu\upsilon L$ = sealing coefficient

$\psi(Q{=}0) = -(0.5+A_2)/A_1$, solved by iteration since $A_2 = f(\psi)$ (42)

$A_1 = \tan\Phi/\beta^2\left[K_1 + K_2\left(\frac{Re_c \cos\Phi}{\beta}\right)^{\kappa}\right]$

$A_2 = \left[\psi C^1/(1-\alpha)\cos\Phi\right]\left[\beta/\alpha Re_c \sin\Phi \cos\Phi\right]^{\frac{1-\lambda}{2-\lambda}} \cdot \left[1 + \left(\frac{1-\alpha}{2C_1\psi}\right)^{\frac{2}{2-\lambda}} Re_c^{\frac{2-2\lambda}{2\lambda}}\right]^{\frac{1-\lambda}{4-2\lambda}}$

$C_1 = 4\,(2)^{\lambda}C$

$C = 0.316$ empirical turbulent flow constant

$\lambda = 0.19$ " " " " (= 1 for laminar flow)

$\kappa = 1.003$ " " " "

$K_1 = 12$ " " " "

$K_2 = .00725$ " " " "

$Re_c = {}^{P c\upsilon}/_{\mu}$ (= 0 for laminar flow)

Stair and Hole [8]

$$Q = \frac{\pi}{0} \ldots - \frac{h_r^3}{\mu}\frac{d\bar{P}}{dz}\left[\Gamma^3 K_\eta + \left(\alpha(1-\alpha) + \Gamma^3(1-\alpha)^2 + \Gamma^3\alpha^2 + \Gamma^6\alpha(1-\alpha)\right)\tan^2\Phi \cdot K_\xi\right]$$

(continued)

$$-.5\,\upsilon h_r\,\alpha(1-\alpha)\tan\Phi \cdot (\Gamma-1)\,(\Gamma^3-1) \ldots (1-\alpha)\,\Gamma^3 + \alpha \tag{43}$$

where

$K = \frac{1}{12\,(2.8F+1)/3.8}$; K_η implies F_η etc.

$\quad = \frac{1}{12}$ (laminar flow)

$F_\eta = F_\xi = 1$, laminar flow (Re_η, $Re_\xi < 2000$)

$F_\eta = \frac{0.646}{64}\, Re_\eta^{\,1-0.246}$, transition flow ($Re_\eta \geq 2000$)

$F_\xi = \frac{0.326}{64}\, Re_\xi^{\,1-0.253}$, transition flow ($Re_\xi \leq 2000$)

$Re_\eta = 2000 \sin\Phi\; Re/Re_{crit}$

$Re_\xi = 2000 \cos\Phi\; Re/Re_{crit}$

$Re = c\upsilon/\nu$

$Re_{crit} = 41.1\left[\frac{(D/2c)}{(1-\alpha)+\alpha\Gamma}\right]^{1/2}$

For zero leakage (Q=0) the sealing coefficient is

$$\psi = \frac{h_r^2}{\mu\upsilon}\cdot\frac{\Delta P}{L} = \frac{.5\tan\Phi \cdot \alpha\,(1-\alpha)\,(\Gamma-1)\,(\Gamma^3-1)}{(K_\eta + K_\xi \tan^2\Phi)\,\Gamma^3 + \left[\tan^2\Phi \cdot \alpha\,(1-\alpha)\right](\Gamma^3-1)^2 K_\xi} \tag{44}$$

This expression reduces to equation (1) for laminar flow and differs from equation (7) primarily in the determination of the turbulent flow correction factors, K_η and K_ξ. These factors are compared with G_x and G_z in Figure 5.

$$K = \frac{3.8}{12(2.8F)} \tag{45}$$

or

$$K_\xi = 22.2\, Re^{-0.747} \tag{46}$$

$$K_\eta = 11.2\, Re^{-0.754} \tag{47}$$

The laminar flow power loss, q, is given as follows

$$\phi_f = \frac{q\,c}{\pi D\,\mu\,L\upsilon^2} = (1-\alpha) + \frac{\alpha}{\Gamma} + \frac{3\tan^2\phi\,.\,\alpha\,(1-\alpha)\,(\Gamma-1)^2\,(1-\alpha+\alpha\,\Gamma^3)}{\Gamma^3\,(1+\tan^2\phi) + \tan^2\phi\,.\,\alpha(1-\alpha)\,(\Gamma^3-1)^2} \qquad (48)$$

The friction parameter is

$$\text{F. P.} = 4\pi\,\phi_f/Re_c \quad ;\ \text{laminar flow.}$$

$$\text{F. P.}_o = 0.164\,\pi/Re_c^{0.43};\ \text{turbulent flow.}$$

Both relationships give good agreement with experimental results,

McGrew and McHugh [1]

$$\psi = K_1 + K_2\,Re_h^{\varkappa} \qquad (49)$$

where K_1 K_2, $\varkappa$ are functions of geometry. No general theory is given.

The theoretical friction factor relationships,

$$f = \frac{\tau}{P\upsilon^2/2g} = \frac{2}{Re_c}\left[\frac{\alpha}{\Gamma} + (1-\alpha)\right] \quad ;\ \text{laminar flow} \qquad (50)$$

$$= \frac{0.078}{Re_c^{0.43}}\left[\frac{\alpha}{\Gamma} + (1-\alpha)\right] \quad ;\ \text{turbulent flow} \qquad (51)$$

were found to give predictions about three times lower than measurements.

E. Discussion of Viscoseal Phenomenon

Optimized Parameters

The optimum seal geometry for laminar operation is not optimum for turbulent conditions.

[8] From the viewpoint of pressure stability the lower values of helix angle, ϕ, are preferred since in both laminar and turbulent flow a given change in Γ produces a smaller change in ψ. Small angles give the best range of sealing.

[8] For laminar flow α=0.5 is optimum while for turbulent sealing α=0.7 gives a slight improvement in ψ.

[7] It appears that the optimum performance in the turbulent region, $Re_c > 3000$, is not very sensitive to changes in the optimum values of the geometrical parameters. Values of Γ = 0.6 to 0.7 are recommended.

[9] The optimum geometry proposed is : ϕ = 14.5°, Γ = 6.5, α = 0.62.

[15] Since the convective inertia effects characterize the pressure generation, a complete reevaluation of the optimum geometry, including the groove-ridge shape, must be investigated before conclusions on optimal viscoseal performance, in the noncreeping flow regime, can be made.

End Effects

[7] For seals with small L/D ratio the end-effects should be opposed by increasing the number of threads.

[17] At Reynolds numbers above 10,000, the wetted length of a test seal (D=2 in., C = 3.2 x 10^{-3} in., ϕ = 14.5°, α = 0.62) became less than 0.2 in. and there was an apparent decrease in sealing coefficient. The higher the pressure the higher is the Reynolds number at which this drop off occurs. It is not certain whether this phenomenon is a result of peculiarities in pressure measurement or an actual decrease in sealing coefficient because of the low wetted length.

[15] The viscoseal has a certain ineffective length at the high pressure end of the seal. This is called the end effect, and it has been suggested that this ineffective rotor length be subtracted from the calculated wetted-seal length in order to compare theory with experiment.

[9] Another effect, that has not been considered by any author in the turbulent viscoseal, is the influence of the radial groove walls on the flow in the groove direction. This effect will become increasingly important with increased number of threads or upon increasing the ratio of groove depth to groove width. For laminar operation this effect has been taken into account by Booy [14].

[3] It is assumed that the ratio w_g/h is large enough that the groove side walls do not affect the turbulent flow in the groove.

Convective Inertia Effects

[9] Viscoseal theory has neglected the "entrance" effects of the flow across the lands and grooves. It has been assumed that this flow is fully developed throughout. These effects may, however, become important when the number of threads is increased so that the ratios of the land and the groove width to the clearances h_r and h_g increase, respectively.

[15] Use of a modified Reynolds number is recommended where

$$Re^* = Re_c\,\frac{c}{L^*}\ ,\ \text{if}\ \frac{c}{L^*} < 1 \qquad (52)$$

$$L^* = (\omega_g + \omega_r)/\sin\Phi = \pi D/\kappa \qquad (53)$$

At $Re^* = 1$ significant improvement in the sealing coefficient is to be expected due to the increasing effect of convective inertia and not to the onset of turbulence.

Convective inertia effects cause other significant parameters to arise: (a) fluid density ρ, (b) groove aspect ratio ω_g/h, and (c) the number of thread starts, κ.

[3] The characteristic length, $L^* = \pi D/\kappa$, has a negligible effect on the performance of viscoseals having the same Φ, Γ and α.

Flow Transition

[8] As α increases the onset of turbulence occurs at lower Reynolds numbers.

[7] At Reynolds numbers $Re_c \approx$ 300-1000, the flow apparently becomes turbulent, manifested by a relative rise in sealing coefficient and friction factor.

[3] At low Reynolds numbers, $Re_c \leq$ 200-600, the seal operates in the laminar flow regime. A smooth transition in the flow regime occurs at $Re_c \geq$ 600-1000 as indicated by an increase in ψ. The transition starting point is apparently a function of seal geometry.

[16] The Reynolds number below which the experimental sealing coefficient retained a laminar characteristic could be effectively determined by using a critical seal Reynolds number defined as:

$$Re_{crit} \leq 41.1 \left[\frac{D/2}{(1-\alpha)C + \alpha\Gamma c}\right]^{1/2} \qquad (54)$$

Effect of Seal Eccentricity

[8] For laminar operation, in the absence of gas ingestion, the sealing coefficient was essentially independent of eccentricity for $0 \leq \Sigma \leq 0.6$. In the turbulent range ψ increased slightly with Σ, which is contradictory with the theory.

Increasing the seal eccentricity ratio increases the intensity of gas ingestion.

Gas Ingestion

[8] No indication of gas ingestion was detected in the laminar flow regime. The Reynolds number at which gas ingestion is first detected is

not the same for each seal but appears to be a function of seal geometry. The same is true for the maximum Reynolds number for stable operation with gas ingestion.

At low turbulent Reynolds numbers the seal performance changes during a long term test and may reach a stable operating point at a slightly lower ψ than is obtained in a normal short term run.

As the Reynolds number increases the seal pressure gradient becomes unstable with one of the following possible characteristics:

(1) Small fluctuations leading to large drop in ψ followed by stable operation to higher Reynolds numbers.

(2) Small, low frequency fluctuations and increased seal length. Increased pressure causes the effective length of the seal to be exceeded and a steady leak results. If both speed and pressure are increased the leak becomes pulsating.

(3) High frequency fluctuations in pressure and a pulsating leak. Most common with a small trapped sealed cavity.

In order to minimize the effects of gas ingestion on ψ, large Γ and α, and small helix angles, Φ, should be employed.

[12] Experiments have shown gas ingestion to be a severe problem and it can adversely affect sealing capacity. Gas ingestion characteristics are fundamentally different for the grooved rotor as compared to the grooved housing. In the grooved housing, centrifugal action works to hold the gas bubbles out of the grooves, thus the bubbles tend to escape over the lands rather than be pumped to the seal high-pressure end, and a condition of no gas ingestion can be achieved. In the grooved rotor, centrifugal action forces the gas bubbles into the rotor grooves, thus the gas bubbles are trapped and pumped to the seal high-pressure end; increasing speeds cause increasing gas ingestion.

For the same dimensions, a grooved housing gave a higher ψ and no detectable gas ingestion. With a grooved shaft, gas ingestion was always present.

Recent analytical work indicates that convective inertia effects, groove aspect ratio, h/ω_g, fluid density and number of helix starts are important gas ingestion parameters.

[13] Gas ingestion requires further investigation since this phenomenon poses a serious deterrent to a wide application of the visco seal. The response of the turbulent visco seal to gas ingestion is a function of the sealant system as well as seal geometry.

[9] Indications are that a larger groove depth may help to avoid seal breakdown.

The drop off in sealing coefficient at high Reynolds numbers can be attributed to short wetted lengths, where gas ingestion can have a major and serious affect on performance.

Tests with different fluids indicate that the shaft surface velocity at which seal breakdown occurs roughly correlates with the surface tension of the sealed fluid. The location of the threads, shaft or sleeve, did not affect the surface velocity at which breakdown occurred.

[1] The on set of "seal breakdown,"* which appears as a small loss of sealed liquid and a decrease in the seal pressure, was studied in relation to Reynolds number = $\upsilon c/\nu$, Froude number = υ^2/gc , and Weber number = $\upsilon^2 c\rho/\sigma$. Weber number appeared to give the best correlation with seal breakdown, but was not conclusive. The higher the surface tension, the higher the shaft speed before seal breakdown. Seal breakdown occurred in the laminar, turbulent, and transition regimes, depending on the fluid tested.

[3] Normal accelerations or velocities toward the liquid of an interface extending between closely spaced solid surfaces having a relative motion of a periodic nature, as in a viscoseal, can produce an interface instability which may result in gas ingestion. Surface tension tends to prevent the development of such instabilities.

In viscoseals having the same Φ, Γ, α and speed, wider grooves and lands tend to minimize gas ingestion.

Gas ingestion in viscoseals appears to be a naturally occurring phenomenon which cannot be eliminated by optimization of the geometry, but which can be avoided only by imposing limitations on shaft speed.

[15] Gas ingestion can be eliminated by using a grooved housing. Secondary grooves on the viscoseal land areas improved the sealing capacity.

* [3] While the mechanism of "seal breakdown" or secondary leakage may be significantly different from that of gas ingestion, both appear to be related to some form of instability of the gas-liquid interface.

It has been reported that some viscoseals work on an air-liquid emulsion or with a mixture of oil and gas. These modes of operation could be due to gas ingestion. Leakage observations reported on "secondary leakage" and "seal breakdown" could result from gas ingestion into a closed cavity. Gas ingestion characteristics may present no problem with a vacuum on the low pressure end.

Viscoseal Operating Regions [15]

Figure 20 illustrates five typical regions that exist in high speed viscoseal operation where a pressurized liquid is being sealed axially from a lower pressure gaseous environment.

These regions are:

(1) Pressurized Cavity Region

(2) End Effect Region

Where the helical grooves connect directly with the pressurized cavity, the pressure developed along the land leading edge decays sharply. The pressure decay within the groove is due to the equalization of the land leading and trailing edge pressures, since a common pressure must exist in the cavity.

The end-effect axial length is approximately equal to one-half the axial groove width.

Visual studies indicate high fluid convective exchange in the end effect region. The end effect region can be reduced by utilizing a circumferential dam; however, performance is penalized with less convective cooling exchange.

(3) Homogeneous Fluid Region

This region is of prime importance in determining the pressure generation capability of the viscoseal because the axial pressure gradient is the steepest here, and this region is amenable to theoretical analysis.

(4) Liquid-Gas Mixing Region

Under conditions of an unstable liquid-gas interface, there will be a liquid-gas mixture region between the nonwetted region (5) and the homogeneous fluid region (3). Under certain conditions the entire axial length may be a liquid-gas mixture, and as a result, gas may be pumped to the pressurized cavity (gas ingestion).

The axial pressure gradient is nonlinear in the liquid-gas region. As the bubble size progressively decreases when moving from the low pressure end to the high pressure end, the axial pressure gradient becomes steeper.

(5) Scavenging or Nonwetted Region

This region is not wetted and changes as the wetted length changes. If this region is large enough, there will be no seal fluid leakage during seal "blowout" since this leakage is scavenged back into the wetted length of the seal.

Observations for grooved housing design

Re*	Observation
1.75	Clear, stable interface
2.58	Large gas pockets, gas ingestion, leaking into vent line, lower ψ.
3.50	Various degrees of gas ingestion. Bubbles get smaller traveling from
6.10	low pressure to high pressure end.
7.00	No detectable gas ingestion.
10.84	Two regions: (a) gas-liquid, (b) homogeneous.

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Nomenclature

a	=	constant in eccentricity relationship
c	=	radial land clearance = h_r
C_ε	=	friction power loss coefficient
C_c	=	Couette friction coefficient
D	=	real diameter
Σ	=	eccentricity factor = $1 + a\varepsilon^2$; $a = 1.5$ for laminar flow
e	=	land width along axis
F	=	friction function [8]
F. P.	=	friction parameter [8]
G_x, G_z	=	turbulent flow correction factors
h	=	groove depth
h_g	=	radial groove clearance
h_r	=	radial land (ridge) clearance = c
K	=	friction function [8]
L	=	active seal length
L^*	=	characteristic length = $\pi D/\kappa$
κ	=	number of thread starts
P	=	pressure
ΔP	=	pressure differential across seal
Q	=	seal leakage, volumetric
q	=	friction power loss
r	=	seal radius
Re	=	Reynolds number, rotational = $\omega h/\nu$
Re_g	=	$\omega h_g/\nu$
Re_r	=	$\omega h_r/\nu = \omega c/\nu = Re_c$

Nomenclature
(continued)

Re_z	=	Reynolds number due to axial flow $= \bar{\omega}\, h_r/\nu$
Re_{crit}	=	critical Reynolds number for flow transition
Re^*	=	modified Reynolds number $= Re_c\ (c/L^*)$
t	=	thread pitch $= \pi\ D \tan \Phi$
υ	=	tangential surface velocity $= R\omega$
$\bar{\upsilon}$	=	average tangential fluid velocity
ν	=	surface velocity component along groove $= \upsilon \cos \Phi$
ω^7	=	seal leakage rate, mass
$\bar{\omega}$	=	average axial fluid velocity
ω_g	=	width of groove
ω_r	=	width of land (ridge)
x, z	=	coordinates, tangential and axial
$\partial P/\partial z$	=	local axial pressure gradient
$\partial \bar{P}/\partial z$	=	average axial pressure gradient $= \Delta P/L$
GRAD P	=	$(h_r^3 / \mu\nu)\ \partial P/\partial z\ ;\ \partial P/\partial \theta$
α	=	$\omega_g / (\omega_g + \omega_r)$
β	=	h_r/h
Γ	=	h_g/h_r
Σ	=	eccentricity ratio $= \Delta r/c$
η, ξ	=	skewed coordinates, along groove and across groove [5], across groove and along groove [1, 2, 3, 8, 9]
θ	=	tangential, angular coordinate
μ	=	fluid dynamic viscosity
ν	=	fluid kinematic viscosity $= \mu/\rho$
ρ	=	fluid density

σ	=	fluid surface tension
τ	=	shear stress
Φ	=	helix angle
Φ_f	=	friction power loss coefficient [8]
ψ	=	sealing coefficient $= h_r^2/\mu\,\upsilon\quad \Delta P/L$
ω	=	shaft rotational speed

Subscripts

c	=	Couette flow
l	=	land or laminar
p	=	pressure
d	=	drag
g	=	groove
r	=	ridge (land)
vs	=	viscoseal
Bush	=	bushing

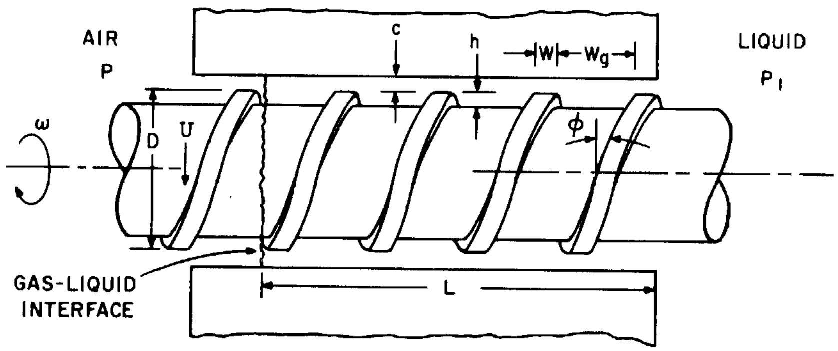

Figure 1
Viscoseal Geometry

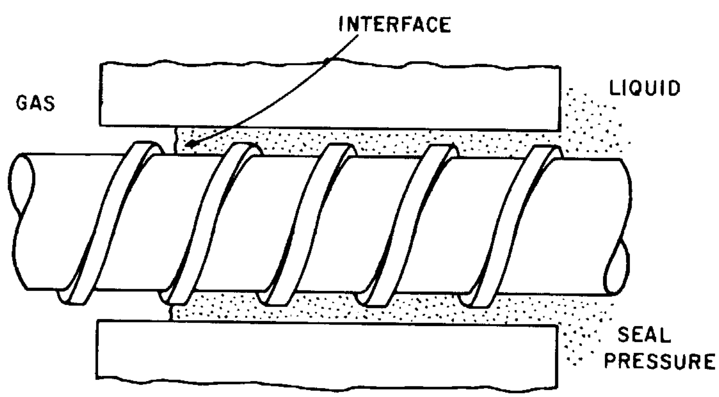

Figure 2
Liquid - Gas Seal

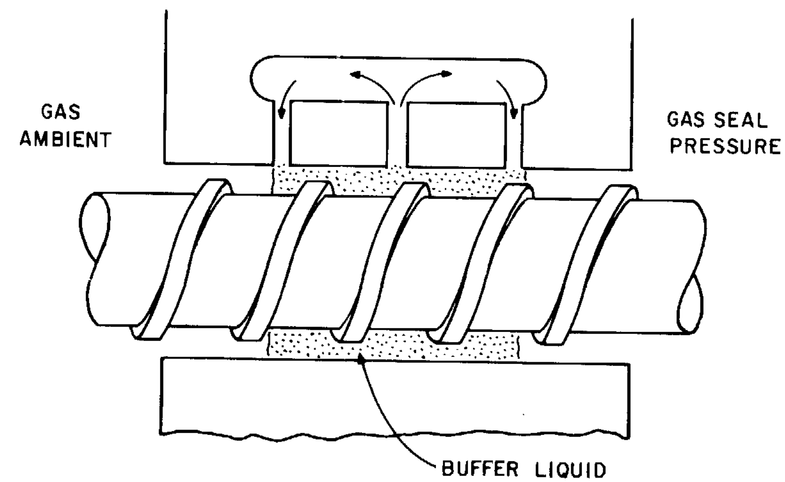

Figure 3
Gas-Liquid-Gas Seal

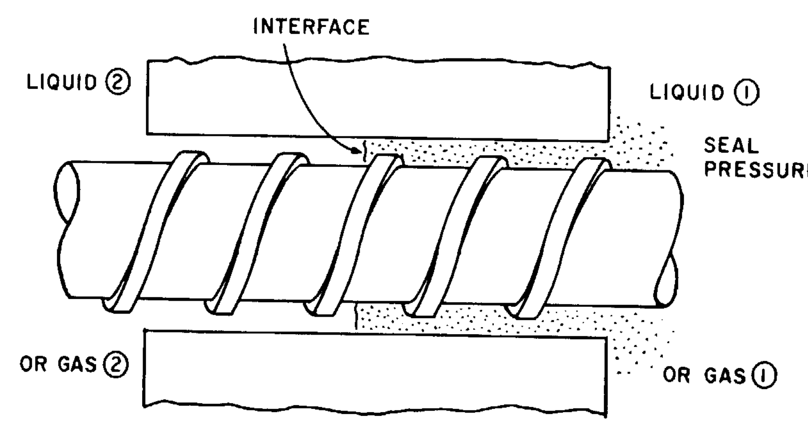

Figure 4
Liquid-Liquid or Gas-Gas Seal

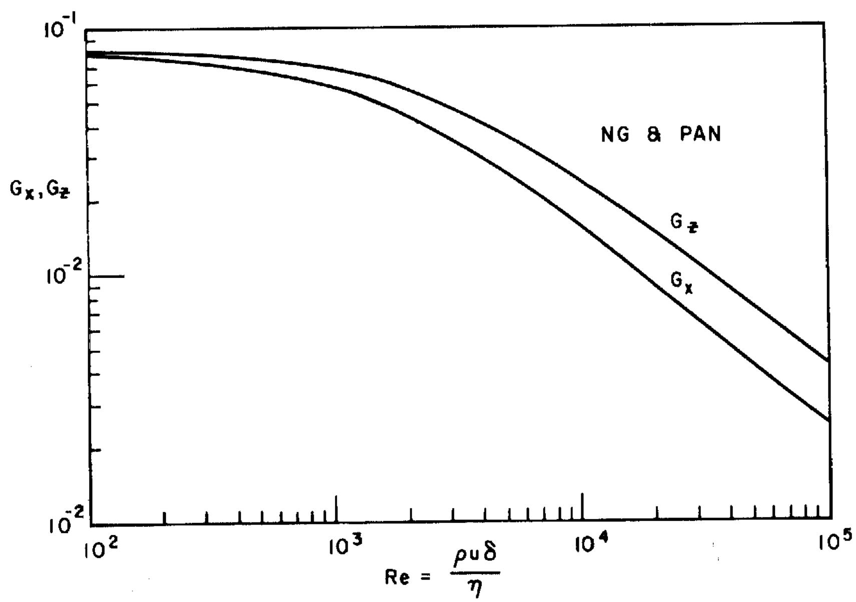

Figure 5
Turbulent Flow Correction Factors

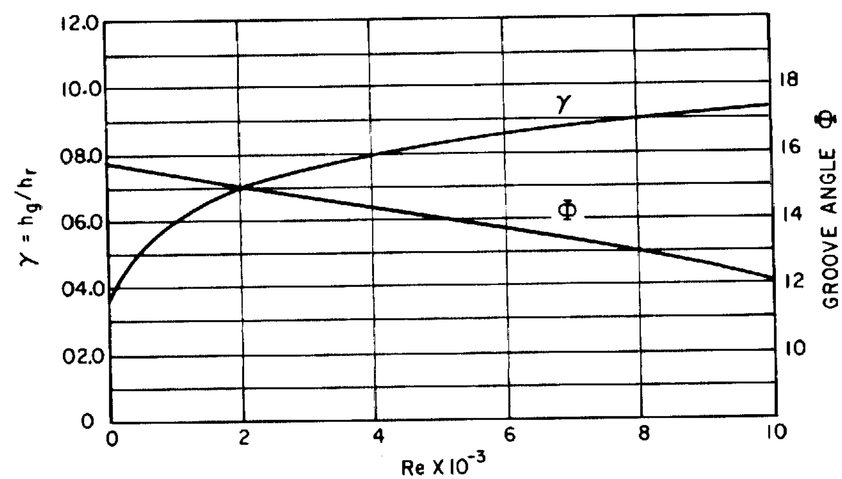

Figure 6
Optimum Geometrical Parameters for Sealing Coefficients [5]

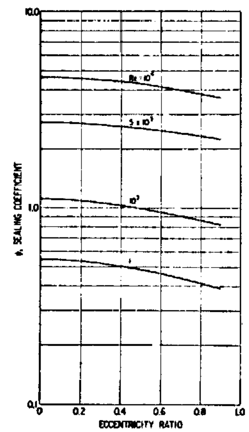

Figure 7 - Effect of Eccentricity On Sealing Coefficient

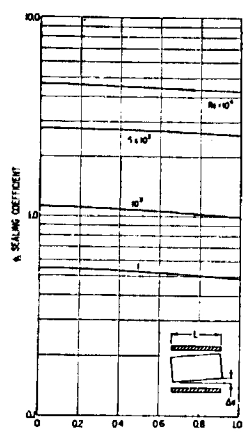

Figure 8 - Effect of Misalignment On Sealing Coefficient

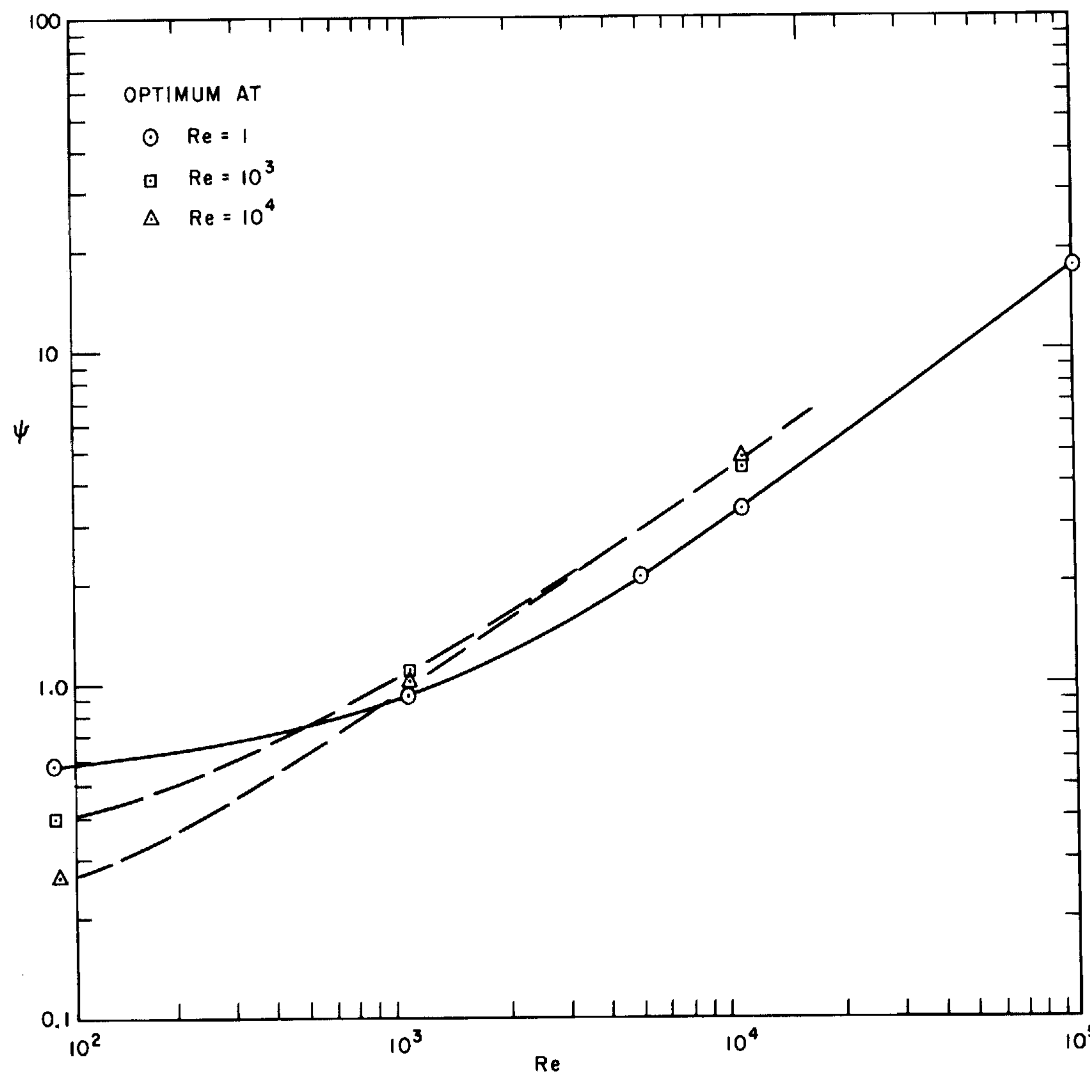

Figure 9

Sealing Coefficient as a Function of R_e for Three Different Geometries

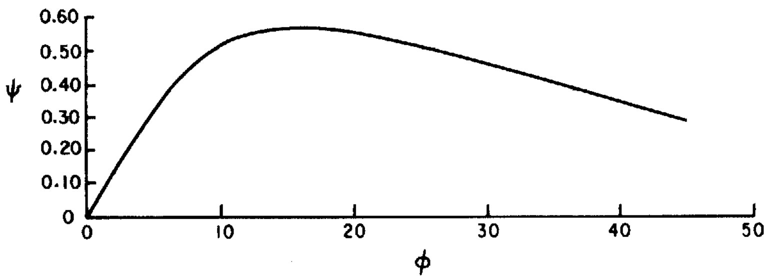

Figure 10
ψ vs. ϕ for Optimum Geometry at R_e=1: γ =.50, β=0.37(=3.7)

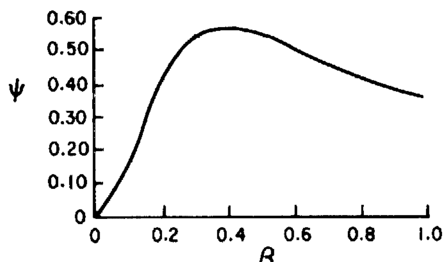

Figure 11
ψ vs. β for Optimum Geometry at R_e=1: ϕ =15.8°, γ=.50

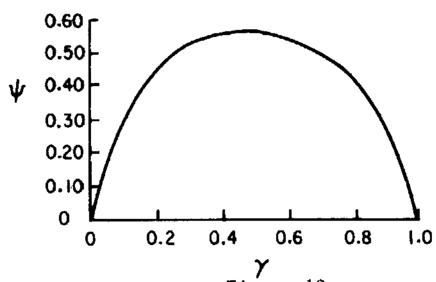

Figure 12
ψ vs. γ for Optimum Geometry at R_e=1: ϕ=15.8°, β=0.37

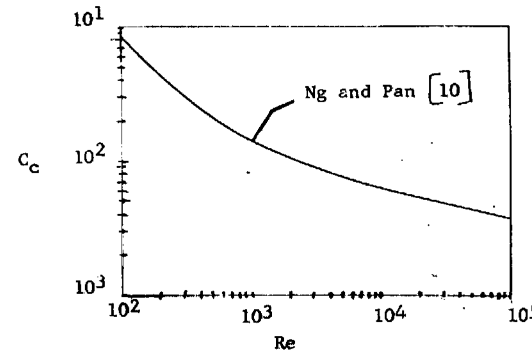

Figure 13
Couette Friction Factor vs Reynolds' Number
[7]

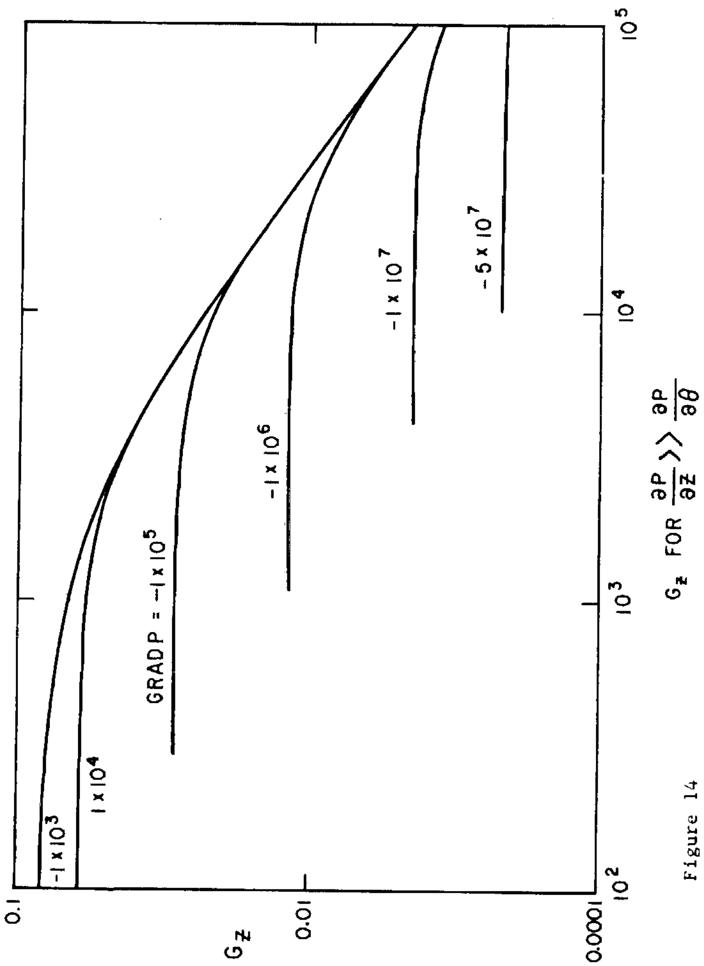

Figure 14
Turbulent Flow Correction Factor

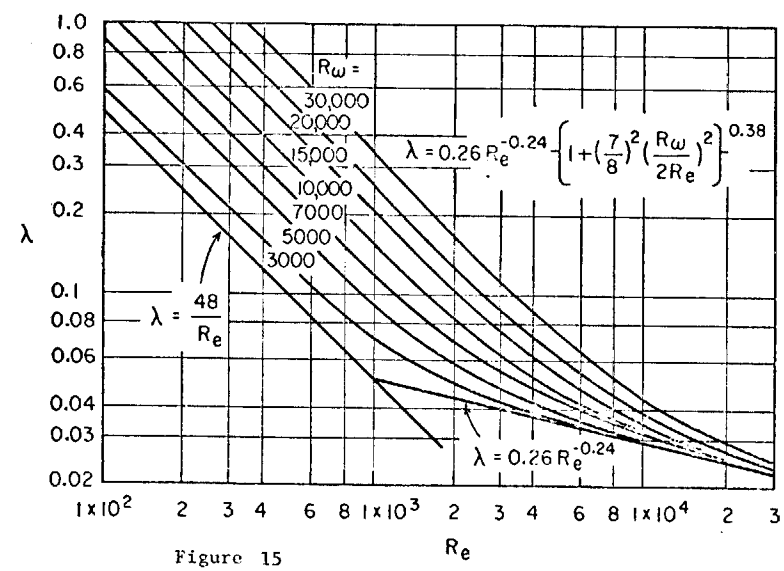

Figure 15
Theoretical Curves of τ - R_e when R_ω is constant [18]

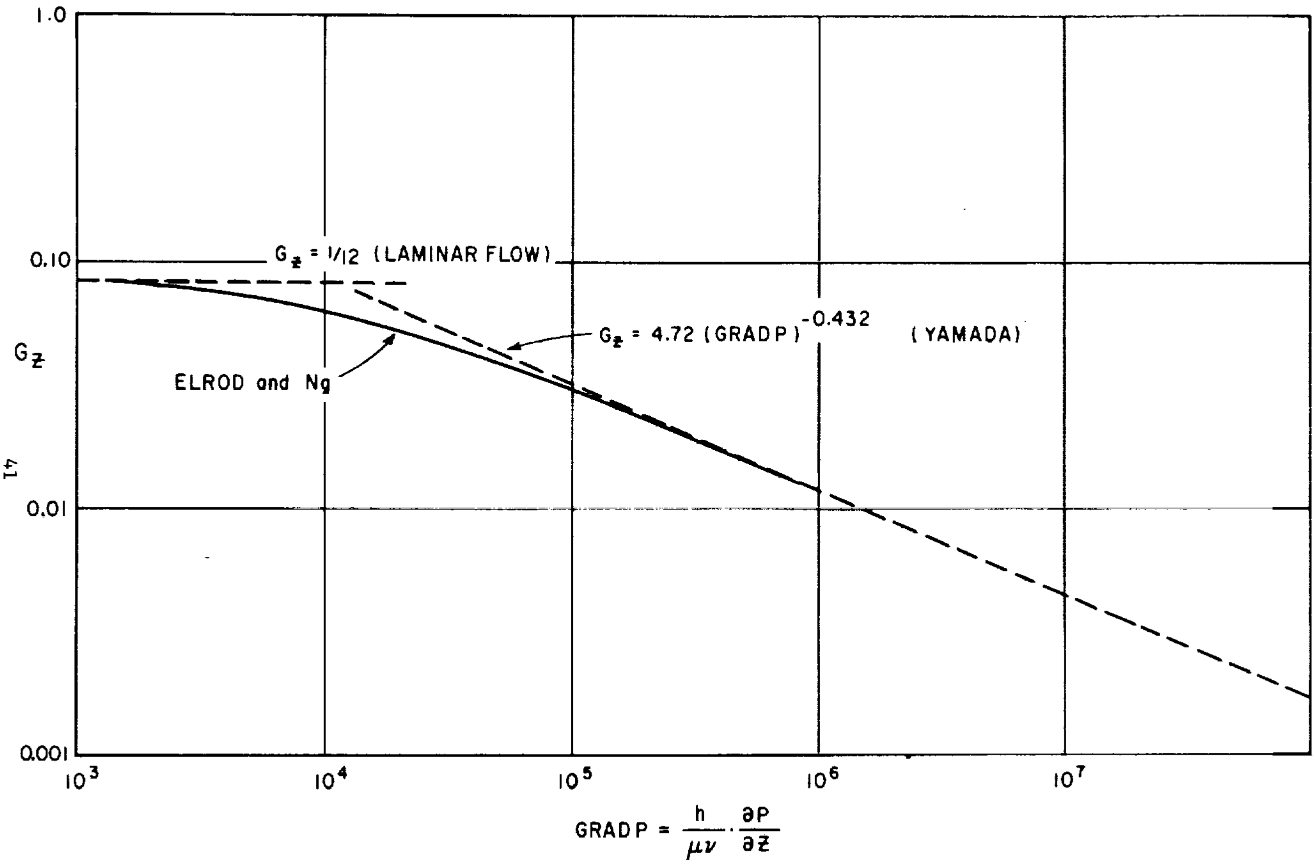

Figure 16

Turbulent Flow Correction Factor For ω=0

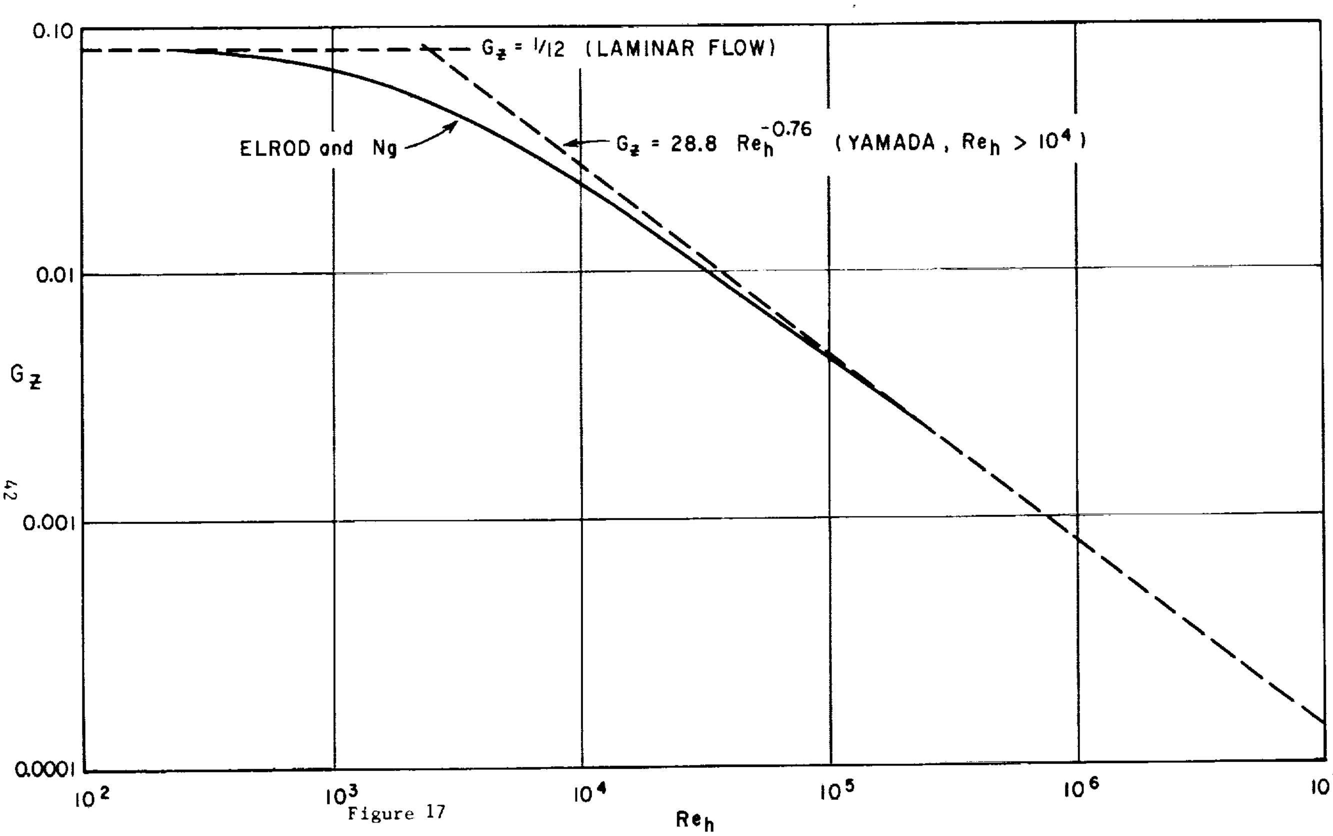

Figure 17

Turbulent Flow Correction Factor For No Through Flow

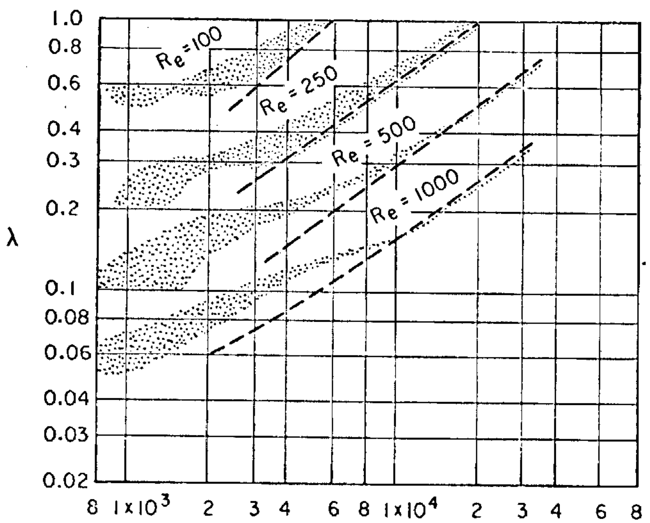

Figure 18

Relation between τ and R_ω for various Re [18]

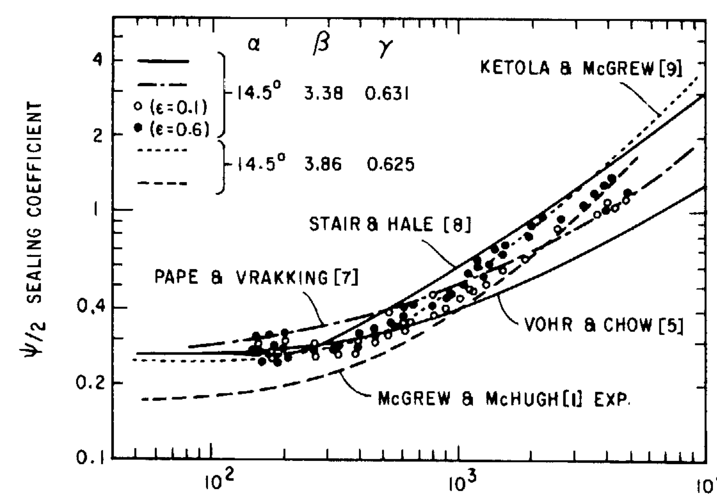

Figure 19

Comparison of Theoretical and Experimental Performance of Typical Viscoseals [3, 5]

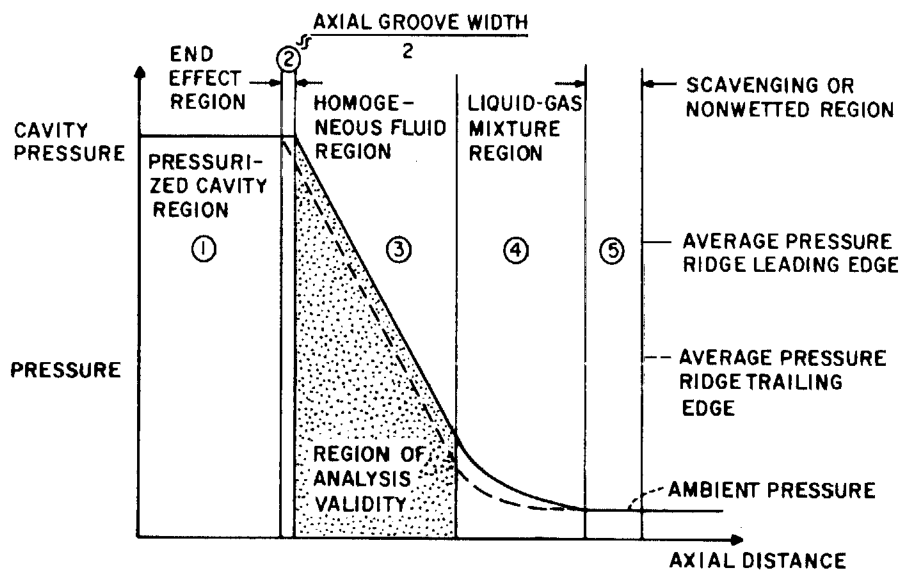

Figure 20

Five Typical Regions of High Speed Viscoseal Operation [15]

E. LIP SEALS

LIP SEALS

Including Positive Action Spiral Groove Seals

or

Helixseal

I. Introduction

Principles of Operation

Since the Helixseal or Positive Action Spiral Groove seals (S. G. seal) are special lip seal designs, they will be discussed under the overall heading of "lip seals."

Figure 1 shows a schematic of a radial lip seal including the appropriate nomenclature. It is primarily used for retaining lubricating oils, at moderate pressures, from leaking along rotating or reciprocating shafts.

[1] The sealing action of a normal lip-type seal has been demonstrated to be due to surface tension effects in the fluid at the outer edge of the contact area. As long as the film of oil between the sealing lip and the shaft remains thin enough ($<10^{-4}$ in.) surface tension is able to form a meniscus which bridges the gap and prevents leakage. If the clearance between shaft and seal becomes too great, the oil can no longer bridge the gap and the seal will leak. The clearance may be locally increased due to shaft damage, seal damage, seal damage on the contact band or eccentric running of the shaft and inability of the sealing lip to follow the radial displacement of the shaft surface.

[1] In the case of the spiral groove lip seal (Figure 2) the threads collect oil that has leaked past the contact area and directs it back to the main film of oil in this zone. Leakage is never permitted to build up to the point where a large drop can form and run out. Thus the positive action of the grooves acts as a back up to the normal lip seal action and is useful only under extreme conditions where a normal seal would leak.

Advantages of Lip Seals

(1.) Low cost.

(2.) Easy to install.

(3.) Effective if properly designed and installed.

(4.) Small space.

(5.) Will tolerate a certain amount of shaft run-out, misalignment and vibration.

Advantages of Lip Seals (continued)

(6.) With proper selection of materials, will handle many oils and hydraulic fluids.

Disadvantages of Lip Seals

(1.) In general not for extreme temperature or pressure use (3-7psi).

(2.) Film thickness and contact pressure between seal lip and shaft must be controlled precisely.

(3.) Stick-slip phenomenon causes leakage.

(4.) Does not operate well dry unless contact pressure is very light.

Advantages of Spiral Grooved Lip Seals

(1.) Because of the grooves it can tolerate larger film thicknesses.

(2.) Normally operates as an ordinary lip seal, using the spiral grooves only as back up.

(3.) Possible to set contact pressure lighter so as to tolerate periods of dry operation.

Disadvantages of Spiral Grooved Lip Seals

(1.) In general, they are effective for only one direction of rotation. This does not appear to be a major problem, however.

(2.) Much more expensive than ordinary lip seals.

II. Design

A. Seal Types

A. Seal Types	Description	Application
Single Lip	Figure 3a	Low shaft speed High viscosity lubricants
Single Lip, Spring Loaded	3b	Higher shaft speeds Lower viscosity lubricants
Double Lip, One Spring Loaded	3c	Containing lubricant Excluding dry contaminants
Dual Lip	3d	Containing lubricant Excluding liquid contaminant
Split Seal	3e	Difficult assemblies Splash lub. systems, will leak if flooded
External Seals	3f	Fixed shaft, rotating bore Centrifugal force maintains contact
Hydrodynamic, Spiral Groove (Helixseal)	Figure 2	For applications where a normal lip seal will not perform sufficiently well. [1] Automotive crankshafts main application. Large oil flows and shaft vibration

B. Operating Parameters

Parameter	How Operation is Affected	Recommended Limits
Contact Pressure	Most important performance factor. High contact pressure causes heating and wear. Low pressure results in leakage.	Low as possible to still insure sealing.
Shaft Speed	Major effect on seal life. Rubbing speed produces friction heating. Other parameters are more restricted at high speeds, i.e., temperature, shaft tolerances, etc. Seals rated in ft/min surface speed.	<4000 ft/min, synthetic seals.
Shaft Alignment, Shaft Dynamic Runout, Eccentricity	Causes flexing of lip. Interrupts seal contact. Causes leakage. Spring must be used at higher speeds.	
Shaft Surface Condition	Too rough gives high friction and wear. Too smooth, will loose oil film.	Hardness $\geq$ 30 Rockwell C Finish 10 to 20 μ in. No better. Polished or ground surfaces without directional finish marks are preferred. Any finish works should pump inward.
	Excessive out-of-round and lobing require excessive lip force.	Shaft out-of-round < 200 μ in.
	Oversize shaft, excessive lip pressure. Undersized shaft, leakage occurs.	Tolerance Diameter ±.001
Fluid Pressure	Increases contact pressure in properly designed seal.	7psi/1000 fpm 5psi/2000 fpm 3psi/>2000 fpm 10 psi upper limit
Operating Temperature	Changes material properties, i.e., thermoplastic change in fluorocarbons and loss of spring tension due to annealing. Hardening (cause cracking) and loss of flexibility. Reduces oil viscosity. Increases oil volatility, flemability and chemical activity. Usually 25 to 50°F above surroundings.	-75 to +350°F moulded synthetic >350°F fluorocarbon plastics and elastomers. Should be held as low as possible, below 225°F.

Parameter	How Operation is Affected	Recommended Limits
Sealed Fluid	Materials must be selected that will not be attacked by sealed fluid.	
Lubrication	Dry running will cause heating, rapid wear and early failure.	
Ambient Conditions	Dirt and abrasives should be kept out of the contact zone.	
Installation	Careful seal handling to avoid damage and assure proper alignment and contact pressure.	
Start Up	Seal must not be dry.	

Contact Pressure of Radial Load

[3] Radial load between a sealing lip and its shaft is one of the most important criteria in lip seal design. The measurement of the radial load is a difficult problem which has not yet been satisfactorily solved. Very little is known about contact loads and pressures, especially under running conditions. Contact load directly affects the ability of the lip seal to retain oil, which depends upon keeping the oil film thickness below a certain value.

The contact band of a new seal is about .005" and increases .020" to .060" with running. This results in a reduction of the contact pressure for a given radial load.

Permanent set of the lip may also reduce the contact pressure. Running contact pressures will be lighter than static over because of the friction heating and temperature rise at the lip.

Current methods for measuring radial loads are useful for purposes of quality control, but are not adequate for research studies.

[4] Considerable changes in radial load due to working conditions can cause the failure of a seal. More attention should be paid to a thorough consideration of the working conditions before a seal is designed.

It is felt to be impossible to manufacture seals in quantities having tolerances much closer than ± 1/3 of the expected radial load.

Examples of the variation of radial load under different field conditions show the specific radial load, interference, and Young's modulus are subject to quite considerable changes.

Effect of radial load on seal behavior is:

(a) If the radial load is too light, the seal will tend to leak.

(b) High loads give increased wear and shortened life.

(c) Friction and lip temperature are directly affected.

(d) Shaft wear is directly related.

(e) Decisive factor for allowable static and dynamic eccentricity of the shaft.

(f) Limits sealing pressure.

The radial load is the most important factor for design and control, assuming there are no seal or shaft defects and the shape of the lip is correctly designed. The sealed fluid is also very important because of the lubrication and cooling it provides.

A comparison of various seal shapes, made by different manufacturers, often shows radial loads different by a factor of four for the same diameter. This is a result of empirical design. The seal designer, therefore, should be in a position to evaluate the radial load beforehand.

The following factors will affect the radial load during operation:

(a) Material Changes

Reversible changes in the elastic modulus are shown in Figures 4 and 5. Irreversible changes in the modulus are due to swelling (decrease in E) and hardening. These are a function of time, temperature and fluid sealed.

(b) Dimension Changes

Reversible changes are due to the thermal expansion or shrinkage of the lip (Figure 5).

Irreversible changes are due to the relaxation effect.

(c) Garter Spring Changes

Stress relaxation over a period of time will reduce the garter spring force. This is influenced most strongly by temperature.

Figure 6 shows the effects of garter spring, initial thermal expansion and time changes in swelling, hardening, relaxation and spring fatigue.

Figure 7 shows a lip seal model to which the following expression for radial load applies

$$P_r = \pi \ D \ C \ \delta + P_{rF}$$

where

P_{rF} = radial load due to garter spring
D = shaft diameter
δ = diametral interference ($D-D_{seal}$)
C = seal spring constant ($E \cdot f$)
E = modulus of elasticity
f = seal shape factor $3/4 \left(\frac{SA}{h_L}\right) \cdot \left(\frac{h_L}{D}\right)^2 + 1/8 \left(\frac{SA}{hL}\right)^3$.

Seal Friction [2]

Seal coefficient of friction can be characterized as follows

$$f = \Phi \cdot G^{1/3}$$

where

$$G = \mu \ \upsilon \ B/P.$$

For a given seal the characteristic number Φ is given in Figure 8 and the seal leakage is related to it. The absolute value of this parameter will depend upon the seal design and material.

Seal Lip Temperature [8]

The life of a radial oil seal is very much influenced by the temperature in the contact zone of lip and shaft. Early failure will be caused by exceeding the temperature limit of the seal material or sealing fluid.

The temperature difference of the lip above the surrounding sealed fluid was found to be proportional to $D^{1/3} \cdot W^{2/3}$ and to increase with increased viscosity. Consequently, small oil seals are exposed to considerably higher temperatures than larger seals at the same shaft speed.

The correct oil level is a parameter which considerably influences excess temperature and resultant seal life.

The range of application of radial seals is limited by the maximum lip temperature because high temperatures may cause: (a) adverse changes in mechanical properties of the sealing lip compound, and (b) increased tendency for the lubricants and their additives to react chemically.

The excess temperature, with respect to the ambient, occurs near the sealing edge only. Therefore, design measures to improve heat transfer will be of advantage only if they are applied in the immediate vicinity of the sealing edge.

The maximum temperature of the sealing lip is determined mainly by the frictional heating in the contact zone and the heat transfer along the shaft. The frictional torque is dependent on the medium to be sealed. The best transfer from shaft to surroundings depends upon the heat transfer coefficient at the shaft surface. As soon as the shaft ceases to be surrounded by oil, or other fluid medium, very high temperatures at the sealing edge should be expected (Figures 9 and 10).

Eccentricity [2]

It is noted that the followable limit, shown in Figures 11 and 12, is independent of shaft speed and can only be estimated if the visco-elastic property of the seal material is taken into consideration. Also the seal lip shape, material and interference with the shaft all have an important influence on the dynamic sealing ability of the seal.

Eccentricity promotes the tendency to leak. The followable ability of silicon-rubber is excellent because of its visco-elastic character.

C. Seal Parameters

Parameter	Importance	Recommendations
Hardness of Lip Material	Change in hardness due to exposure to sealant or heat can change contact pressure and cause seal failure.	Durometer hardness should be 80 to 90 for high speed seals.
Lip Design	Most critical in high-speed seals	Design Considerations [6]; (1) Lip area at the point of contact should be as small as possible. (2) Contact diameter should be uniform and under as light a tension as consistent with contamination resistance and tolerance limitations. (3) Lip juncture with metal seal case should be as thin as possible for minimum resistence to bending. (4) Diameter at contact should be molded to size, rather than trimmed. (5) Lip runout with shaft should be a minimum.

Seal Geometry [5]

Figure 1 defines the pertinent seal lip geometric factors. The relation between the seal design parameters and operating variables is shown in Figure 13. A summary of these interactions is given below

Temperature: If high, the radial load must be minimized to control film oil temperature.

Eccentricity: Higher lip interference.
Longer, more flexible beam.
Seal lip should be parallel with the shaft.

Shaft Speed : Results in high friction heating. Radial load must be controlled.

Shaft Finish: Rough finish requires a higher radial load.

Fluid Pressure: A major factor.
Beam thickness should be increased and length shortened.

Life: High approach angle on dry side to keep lip contact width small and minimize wear.
Properly positioned (R-value) garter spring is used for optimum loading and good compromize between initial sealing reliability and ultimate life.
Greatly affected by operating temperature.

D. Materials

Material	Applications	Advantages/Disadvantages
Lips		
Nitrile Elastomers (Buna N)	Most common lip material Temperature range -65 to 225°F.	Hardens in air/oil>180°F Low cost. Compromise between oil resistance and processibility.
Acrylate Elastomers	Temperature range +0 to 300°F.	Resistant to extreme pressure additives. Selection based on a compromise between processing, oil resistance, and low temp. flexibility. Cost 4.5 x Buna N. Tear and fluid resistant.
Silicones Oil resistant types (a) Methyl (b) Methy Vinyl (c) Fluorosilicone	Temperature range -70 to 350°F (-90 to 400°F+)	Swell in some oils. Poor resistance to extreme pressure additives High permeability to oil, assures lubrication of lip. Low wear. High cost=10xBuna N.
Fluoroelastomers	Temp. range -40° to 400°F.	High resistance to Many additives and fluids. More difficult to mold.
TFE Fluorocarbons Viton Fluorel	Temp. above 300°F. Special lubricants, fuels (up to 450°F [7])	High resistance to active and corrosive fluids. High cost = 30 x Buna N Fillers needed Corrosion resistant
Metal Parts		
Stainless Steel	Finger Springs Cases	
Carbon Steel	Garter springs Cases	
Aluminum	Cases	
Brass	Cases	

Major requirements for seal parts [7] :

Lip : Primary seal element.
Stress-strain characteristics, thermal stability, visco-elastic properties, frictional properties, wear and abrasion resistance, are major factors to consider.
Synthetic elastomers are most widely used material.
Leather impregnated or treated to control porosity.
Should not cause corrosion of the shaft.

Case : The metal must be capable of being fabricated.
Resistant to sealed fluid.
Maintain tight press fit during temperature cycling.
Low carbon cold rolled steel commonly used.
Brass
Stainless steel

Spring: Initial tension, load deflection characteristics, and extension when the seal is installed determine the spring's contribution to the radial load.
Close wound helical spring.
Cold drawn or oil tempered spring wire common.
Music wire or stainless steel occasionally required.

E. Expected Performance

Parameter	Expected Performance	Comments
Seal Life	Long >1000 hrs; normal conditions 500 hrs; severe conditions	
	Medium 400-600 hrs	Progressively poorer performance
	Low 100 hrs	Misapplication. Poor design; fabrication.
	Immediate failure	Defective part. Dry running. Poor installation.

F. "Optimum" Lip Seal Design [9]

Although all elements of a conventional lip seal design can be optimized, the frequency of seal failure is still intolerable due to the large number of production variables which affect leakage.

The major factors which cause leakage in a lip seal application are the shaft, the seal, and the assembly operation, in that order.

Shaft Finish

Optimized shaft finish has been established at 15μ in.with limits of 10 to 20 μ in. These parameters hold true irrespective of shaft size and should

be universally applied for quality control in all radial lip seal applications.

Shaft Geometry

All production shafts are inherently out-of-round and this condition si aggravated by press fits over shafts, keyways, and splines.

Out-of-round geometries in excess of 200μ in will cause leakage. Out-of-round geometries usually occur in an uneven number of lobes. As the frequency of lobing increases, seal tolerance to lobe height decreases. A grinding chatter geometry of 45 lobes and out-of-roundness of 200μ in, is worse than a three-lobe geometry of 500μ in.

Many lip seal designs in use today vary substantially from vendor to vendor. Also variations in lip cross section due to manufacturing variables are quite scattered. Results of tests, directed at establishing a better design technology, lead to the following observations:

(a) Molded or trimmed seal lip interference with the shaft should be 0.035 in. Any variation from this value, due to manufacturing variables, should be controlled toward lesser interference.
(b) The garter spring center line, in relation to the seal lip contact surface, should be toward the air side of the sealing element. A good working tolerance for manufacturing purposes is 0.20 to 0.40 in.
(c) The total radial lip force acting on the shaft should be approximately 0.5 to 0.7 lb per in. of circumference.
(d) Some eccentricity is desirable from the standpoint of aiding the seal to establish a fluid film. However, values of eccentricity in excess of 0.015 in. becomes detrimental as seal lip contact surfaces become wider.
(e) An optimum seal lip contact width should be less than 0.010 in. Any increase in contact width produces an increase in leakage.
(f) The flex section thickness should be approximately 0.035 in. Sections heavier and thinner than this value cause problems with increased leakage.
(g) Seal material modulus should be held to approximately 40 psi at 5 percent elongation and 200°F.

Seal material corrosion is an important design factor because of the corrosive nature of most elastomers or other organic seal compounds in contact with the metal shafts. Corrosion is also a major cause of wear of the sealing element and shaft, since once the shaft is corroded, an abrasive action occurs which generally wears both elements of the system. This occurs in the presence of high humidity and can be accelerated by heat. It is only slightly inhibited by application of lubricants to the sealing areas.

Seal material quality control is another application problem. A review of material hardness values used in the seal industry shows a range from 55 to 85 durometer shore A. The entire range has been applied for all types of elastomers without any apparent detrimental effect to the final function of sealing.

G. Spiral Grooved Lip Seal (Helixseal). [10]

The inherent limitations of the sharp lip or contact sealing system suggests that a different method of sealing be considered. The system wherein immediate benefits can be realized, is one where hydrodynamic and contact methods of sealing are combined. In this manner the deficiency of one system is compensated for by the other.

The spiral grooved lip seal was designed specifically for this purpose. Its static lip contains the oil when the shaft is stationary and also restricts leakage flow during dynamic operation. The helical grooves function as a dynamic seal and prevent oil from migrating much beyond the air-oil interface in the contact zone.

Tests have shown that zero leakage can be achieved for extended periods with little wear. This seal can also tolerate shaft and seal imperfections to a greater degree than the radial lip seal.

Some of the most important spiral groove lip seal design parameters are:

Rib Design

The rib section at the contact area must be deformed to develop a good static seal. The volume of rubber that must be compressed should be kept to a minimum. This is done by increasing the sealing contact approach angle β and the toe angle ψ (Figure 2). The upper limit of the sealing angle β is primarily dictated by the amount of helix rib contact required to produce adequate dynamic sealing; $\beta = 25^o$ to 30^o is recommended. A recommended upper limit for ψ is 60^o. Any greater value may cause installation problems.

The helical rib height, width and cross section should be at a minimum to reduce the volume of rubber which must be compressed. Angular or rounded surfaces aid in this and are easier to mold.

The groove to rib width ratio should always be greater than one with values between 1.5 and 2 preferred.

Spring Position

The garter spring should be placed near the contact point, 0.010 to 0.025 in., and always on the side toward the flex section.

Interference

Poor static sealing is encountered when the interference is too high and poor dynamic sealing results when the interference is too low. Diametral interferences between 0.020 and 0.050 in. are recommended for a 2 in. diameter seal.

Helix Angle

A helix angle between 20^o and 30^o is recommended. If the angle is smaller, the rib contact length on the shaft is increased as does the volume of rubber to be compressed. If the angle is larger, dynamic pumping may not occur.

Static Sealing Lip

The static lip provides a good static seal. However, if the lip is too wide, >0.005 in., and the interference is low, the helical ribs will not contact the shaft and the seal will lose its hydrodynamic pumping capabilities. Poor helix contact also occurs when the crest of the helical ribs is not on the same plane as the seal lip.

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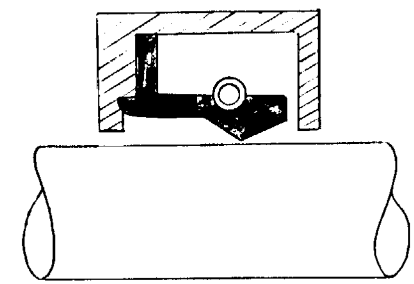

Figure 1a - Radial Lip Seal - cross section.

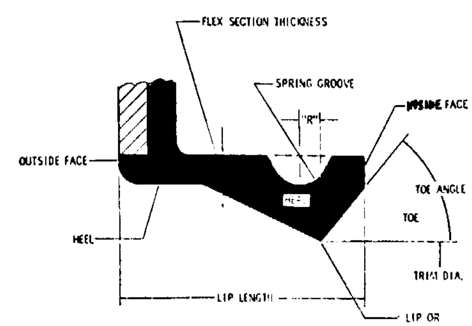

Figure 1b - Radial Lip Seal - nomenclature [11]

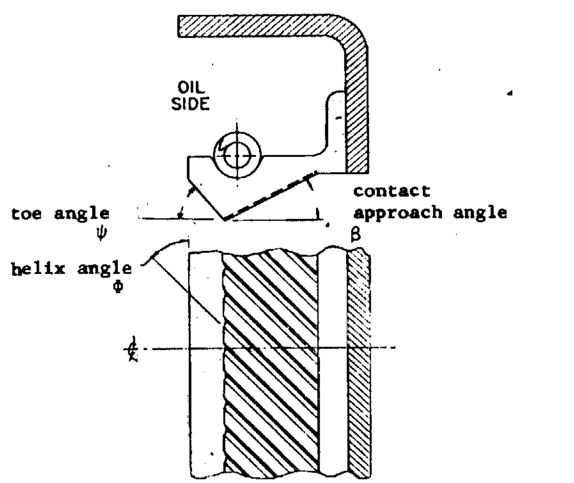

Figure 2 - Large Angle Helixscal with Hydrodynamic Pumping Ribs [9].

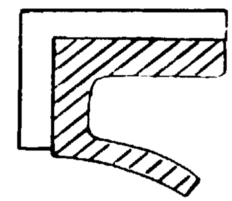

Figure 3a - Single-lip seal.

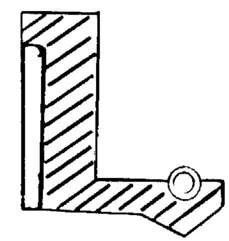

Figure 3b - Single-lip seal, springloaded.

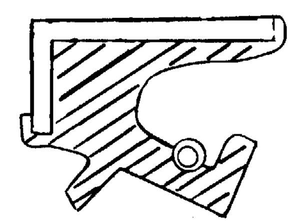

Figure 3c - Double-lip seal, one lip springloaded.

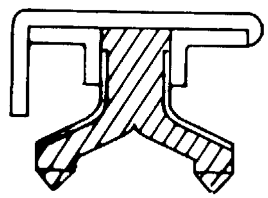

Figure 3d - Dual Lip Seal

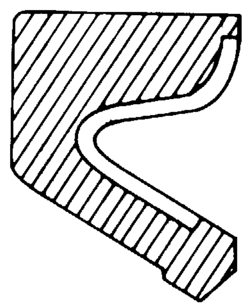

Figure 3e - Split Seal

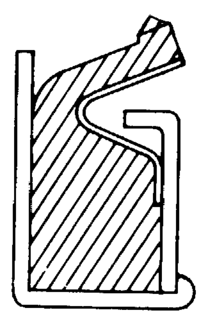

Figure 3f - External Seal

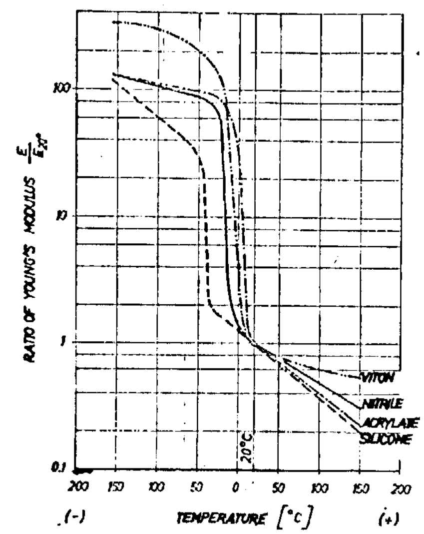

Figure 4 - Young's Modulus Versus Temperature (4).

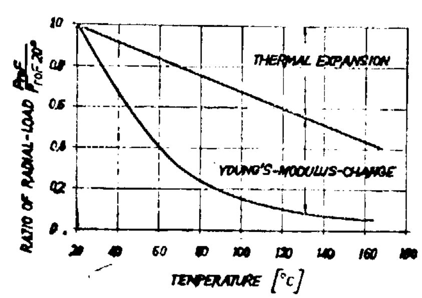

Figure 5 - Radial load ratio Versus Temperature (4).

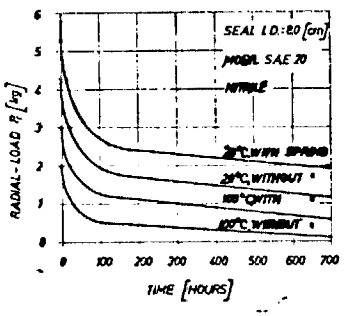

Figure 6 - Radial Load as a function of Time and Oil Temperature With and Without Garter Spring (4).

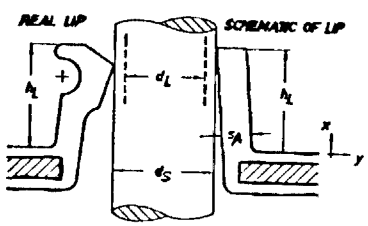

Figure 7 - Transformation of Real Lip Shape Into a Schematic Lip (4).

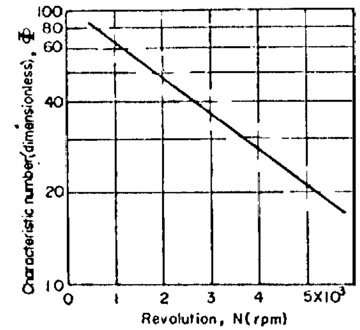

Figure 8a - Dependence of Characteristic Number On RPM. (2).

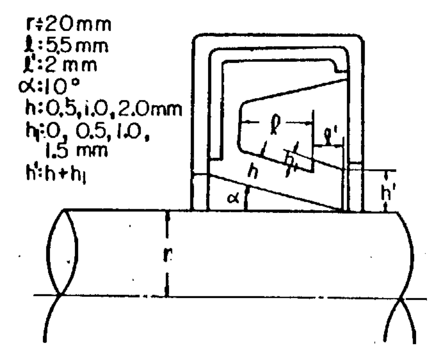

Figure 8b - Cross Section of Sample (2).

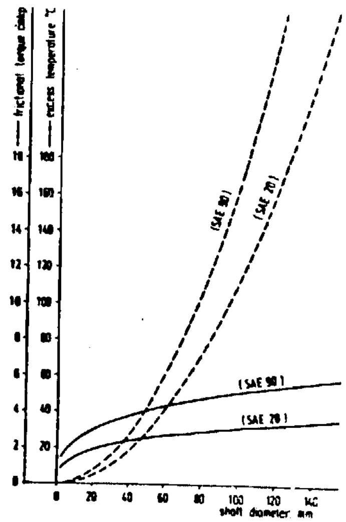

Figure 9 - Excess Temperature at the Sealing Edge.

Data: 100°C, Shaft Wetted by Oil, to 50%, 3000 RPM (8).

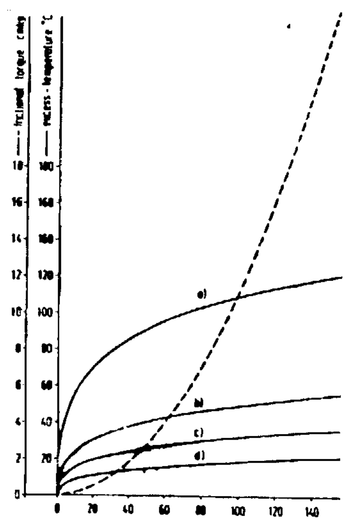

Figure 10 - Excess Temperature at the Sealing Edge.

a) Shaft not wetted by fluid (typical for sealing of grease).
b) Shaft wetted by fluid to 25%.
c) Shaft wetted by fluid to 50%.
d) Shaft completely wetted by fluid.

Data: Motor Oil SAE 20, 110°C, 3000 RPM (8).

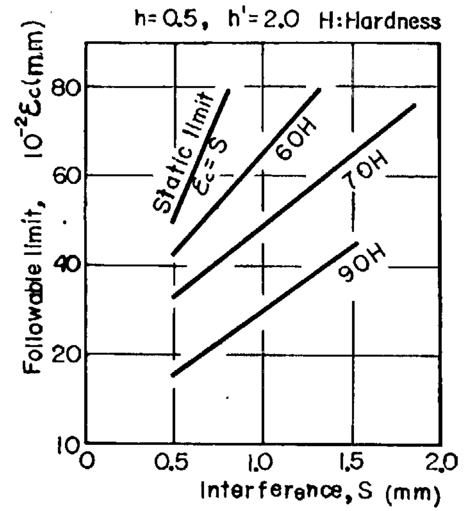

Figure 11 - Followable Limit (2).

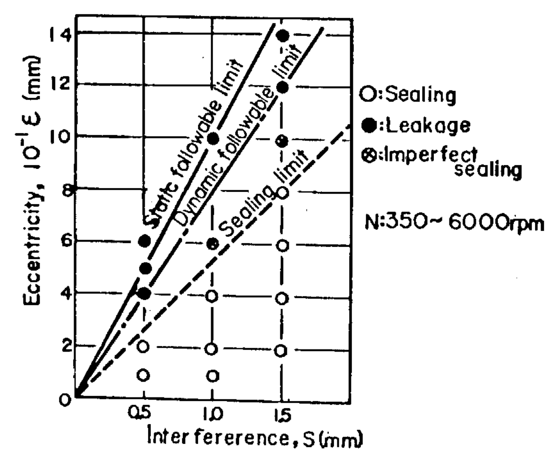

Figure 12 - Relation of Followable Limit and Sealing Characteristics.(2)

Figure 13 (table): Application Variables

		TEMPERATURE	ECCENTRICITY	SHAFT SPEED	SHAFT FINISH	PRESSURE	LIFE
LIP	Interference	1	1	2	3	3	1
	Hoop Area	1	3	2	1	1	1
	Approach Angle	1	2	1	3	1	1
BEAM	Length	3	1	3	3	1	1
	Thickness	2	1	1	3	1	1
	Section Geometry	2	1	1	3	1	1
SPRING	Tension	1	2	1	1	2	1
	Position	3	2	1	3	1	1
OPTIONAL SEALING AIDS	Secondary Lip	3	3	1	3	3	1
	Helices	1	3	1	2	3	1
SURFACE TREATMTS	Pre-lubes & Coatings	2	3	2	2	3	1
COMPOUND	Physical Chemical	1	1	1	1	1	1

Figure 13(5)

1 - Severe Interaction Between Application Variable and Design Variable.

2 - Moderate Interaction Between Application Variable and Design Variable.

3 - Minor Interaction Between Application Variable and Design Variable.

TABLE 1 [2]

OPERATING CONDITIONS ON SEALING TEST

Type of seal	Metal bonded-case seal with garter spring; single lip type
Size	40 x 62 x 12 (mm)
Revolution	3000 - 6000 rpm
Oil	SAE, 30#; 75.5 c.s. (50°C), 14.7 c.s. (100°C)
Oil temperature	Natural cooling
ε	10/100 - 140/100 (mm)

TABLE 2 [7]

POLYMER	NBR	POLY-ACRYLATE	SILICONE	FLUORO-CARBON	URETHANE	BUTYL	TETRAFLUORO-ETHYLENE	NEOPRENE
TEMPERATURE LIMITS								
1. Upper*	225°F.	275°F.	300°F.	350°F.	200°F.	200°F.	425°F.	200°F.
2. Lower	−40°F.	−10°F.	−100°F.	−10°F.	−30°F.	−60°F.	−100°F.	−50°F.
OIL AND FLUID RESISTANCE								
1. Aliphatic Oils (ASTM #1)	OK	OK	OK	OK	OK	NG	OK	Fair
2. 10W - 30	OK	OK	OK	OK	OK	NG	OK	Fair
3. Aromatic Oils (ASTM #3)	OK	OK	NG	OK	NG	NG	OK	NG
4. ATF-A	OK	OK	OK	OK	OK	NG	OK	Fair
5. E. P. Lubes	OK, 200°F.	OK, 275°F.	OK	OK	OK	NG	OK	OK
6. Diesel Fuel	OK	NG	NG	OK	NG	NG	OK	NG
7. Gasoline	OK	NG	NG	OK	NG	NG	OK	NG
8. Water	OK	NG	OK	OK	NG	OK	OK	OK
9. Coolants	OK	NG	OK	OK	NG	OK	OK	OK
10. Synthetic Lubes or Hydraulic Fluids	NG	NG	OK-Some	OK-Some	NG	OK	OK	NG
11. Brake Fluids, Automotive	NG	NG	NG	OK	NG	OK	OK	NG
RESILIENCE INDEX	60	20	40	50	30	20	—	70
PROCESSIBILITY	Excellent	Poor	Good	Poor	Good	Good	—	Excellent

*Temperature of oil in the application; seals normally run 25°-50°F. hotter than oil temperature at the contact point on the shaft.

F. CENTRIFUGAL (SLINGER) SEAL

CENTRIFUGAL (SLINGER) SEAL [1]

I. Introduction

The slinger seal, in its simplest form, is a rotating disk enclosed in a housing. The sealing function is accomplished as a result of the inertial forces created in a rotating fluid ring. Figure 1 is a schematic of the slinger seal showing the velocity distribution in the radial and the tangential directions for the partially wetted case.

Interest in the performance of the partially wetted disk was recently increased due to the application of this device as a component in dynamic seals for space power systems. In this connection it is used both as a seal and as a scavenging pump. Another recent application of the partially wetted disk is that of a dynamic electrical contact on shafts both for power transmission and as an electrical pickup in instrumentation systems. In addition to these specific applications, there has always been a general interest in the partially wetted disk in so far as it is a common component in rotating machinery.

In all of these applications, the pressure, frictional resistance, and flow characteristics of the partially wetted disk are of primary interest.

Four different flow regimes may be identified according to the value of the disk Reynolds number and the spacing ratio between the disk and stationary wall. The analytical expressions for prediction of the pressure gradient developed and the frictional resistance are uniquely determined by the disk Reynolds number, spacing ratio, and the degree of wetting of the disk.

II. Flow Modes and Functional Relationships

It has been shown that four basic modes of flow can exist for the totally wetted disk rotating within an enclosed chamber. These flow regimes are classified as follows:

Table 1

Regime	Description
I	Laminar Flow Merged Boundary Layers
II	Laminar Flow, Separate Boundary Layers
III	Turbulent Flow Merged Boundary Layers
IV	Turbulent Flow, Separate Boundary Layers

The existence of these modes (Figure 2) of flow is dependent upon the Reynolds Number and spacing ratio combinations of the system. Within each flow regime there is a different functional relationship between the torque coefficient, pressure coefficient and Reynolds number-spacing ratio combination. It is important to note that the spacing ratio plays a crucial role in the problem. First, its absolute value determines in part the mode of flow within the chamber and, secondly, within any particular flow regime, its value affects the drag and pumping characteristic of the disk.

For the analysis of the partially wetted disk, it is assumed that all four possible modes of flow will still exist and that their existence will depend on the Reynolds number-spacing ratio combination of the system. Within any particular flow regime, the operational characteristics of the disk, flow, pressure, and drag will be a function not only of the spacing ratio, s/a, but also of another parameter, the engagement ratio, (a-r)/a. The engagement ratio is a measure of the submergence of the disk in the fluid. An engagement ratio of zero means the disk is not wetted at all, while an engagement ratio of 1.0 is the limiting case of the totally wetted disk.

The pressure characteristic of the partially wetted disk is based on the assumption that the radial pressure gradient is equal to that experienced for a fluid mass rotating about its own axis with angular velocity $K\omega$. The pressure rise is shown to be

$$\Delta p = \rho/2\ (K\omega a)^2 \left[1 - (1 - X)^2\right] \tag{1}$$

where K is a function of the spacing ratio and engagement ratio.

The rotating disk can be considered as a small pump. Within the thin boundary layer on the disk the fluid moves radially and tangentially. If the disk is not enclosed, the fluid leaving the disk at the outer radius is dissipated in the surrounding medium. If the disk is enclosed the radial flow leads to a recirculation within the enclosing chamber. It can be shown that the dimensionless flow $Q/\omega a^3$ is a function of the velocity ratio K and the Reynolds number.

In summary the important parameters which define the operation of a partially wetted disk are:

Reynolds Number	$Re = \frac{\omega a^2}{\nu}$
Axial Clearance Ratio	s/a
Radial Clearance Ratio	c/a
Engagement Ratio	$X = (a - r/a)$

Pumping Coefficient	$K = \frac{\Delta p}{\frac{\rho}{2}(a\omega)^2\left[1 - (1 - X)^2\right]}$
Drag Coefficient	$C_m = \varphi\ (s/a,\ Re,\ X)$
Flow Coefficient	$Q/(\omega a^3)$

Table 2

Pumping Coefficient K

Flow Regime	K
I	0.5
II	Figs. 3, 4
III	0.5
IV	Figs. 5, 6

Table 3

Moment Coefficient C_m

Flow Regime		C_m
I		$\pi\left[1 - (1-X)^4\right] / \left[(\frac{s}{a})\ Re\right]$
II	Rotating Disk	$(\frac{s}{a})^{0.10}\left[0.1528+6.74X-8.72X^2+3.75X^3\right]/Re^{0.50}$
	Rotating Housing	$(\frac{s}{a})^{0.20}\left[0.540+7.31X-8.89X^2+3.68X^3\right]/Re^{0.50}$
III		$0.031\left[1-(1-X)^{4.75}\right] / \left[(\frac{s}{a})\ Re\right]^{0.25}$
IV	Rotating Disk	$(\frac{s}{a})^{0.10}\left[0.006+0.168X-0.235X^2+0.107X^3\right]/Re^{0.20}$
	Rotating Housing	$(\frac{s}{a})^{0.20}\left[0.015+0.181X-0.241X^2+0.106X^3\right]/Re^{0.20}$

Regime I, Laminary Flow Merged Boundary Layers

For this case, a Couette laminar tangential velocity profile is assumed, with no radial components. The fluid is contained in an annular ring of arbitrary inner radius R and outer radius equal to a. Note that the radius of the disk and the stationary wall are approximately the same since the radial clearance is considered very small; therefore, $a \neq b$ (refer to Figure 1 for Nomenclature).

Regime II, Laminar Flow, Separate Boundary Layers

For this regime, it is necessary to make certain assumptions regarding the secondary or radial flows. The basic assumption has been to regard all radial flow as taking place only within the boundary layers. The pattern of flow is radially outward on the rotating disk, axially away from this disk along the cylindrical wall, and then radially inward along the stationary end wall.

Regime III, Turbulent Flow - Merged Boundary Layers

For this case, the spacing ratio is assumed to have a negligible effect so that it is possible to directly equate the moments on the rotating disk and stationary wall. It is necessary to assume some turbulent velocity profile.

Regime IV, Turbulent Flow, Separate Boundary Layers

This case also depends upon the model used by Schultz-Grunow [5] as previously developed for Regime II. The same graphical technique described for Regime II - Separate Boundary Layers - was used to equate moments on the stationary and rotating parts. Figure 5 shows the variation of K with the submersion ratio for various values of s/a.

III. Effect of Flow on Power Loss

The rotating disk can be considered as a small pump. Within the thin boundary layer on the disk the fluid moves radially and tangentially. If the disk is not enclosed, the fluid leaving the disk at the outer radius is dissipated in the surrounding medium. If the disk is enclosed, the radial flow leads to a recirculation within the enclosing chamber. In many applications, such as scavenging a bearing cavity, or cooling a slinger seal, it is desired to use the rotating disk as a small pump.

The approximate effect of throughflow on the frictional loss can be considered by analyzing the rate of change of angular momentum imparted to the throughflow. Thus the friction torque can be expressed as a percentage of the zero throughflow case.

For Regimes II and IV, the core is assumed to be rotating as a solid body with angular velocity β when there is no throughflow. β/ω for the zero throughflow case is denoted as K_o. The fluid is considered to enter the fluid casing without any whirl and leave with angular velocity $K_o \omega r$. Obviously, K_o is a function of (s/a) and X from previous analyses in Regimes II and IV and is assumed to be equal to 0.5 in Regimes I and III.

The increase in torque is given by

$$\frac{\Delta M_Q}{M_o} = \frac{2K_o}{C_m} \cdot \frac{Q}{\omega a^3} \qquad (2)$$

This increase in torque can be related to the original C_m expressions. For example, for the case of Regime II, equation (3) becomes

$$\frac{\Delta M}{M_o} = \frac{2K_o(Re)^{1/2}}{C_{x1}(s/a)^{1/10}} \cdot \frac{Q}{\omega a^3} \qquad (3)$$

Flow Characteristics of a Partially Wetted Rotating Disk

When a throughflow is imposed on a rotating disk, a degradation in pressure gradient and an increase in power consumed can be expected. The degree of performance degradation will depend upon the amount of throughflow. A simple approximation of the effect of throughflow on the frictional loss has been made in equation (3).

The results of throughflow experiments are summarized in Figure 7. The value of K for the flow tests show a similar variation to the no flow values of K, except displaced downward. The results of this investigation show a marked reduction in the sealing pressure with small amounts of throughflow.

Conclusions

An experimental program to determine the pressure, frictional resistance, and flow characteristics of a partially wetted rotating disk has been completed. The results and conclusions may be summarized as follows:

1. The pressure gradient developed in the fluid along a rotating disk in an enclosed housing may be adequately described by assuming that a solid core of fluid is rotating at some fraction K of the disk angular velocity. This holds true regardless of the proportion of the disk that is actually wetted by the fluid.

2. The frictional loss due to the rotation of the disk can be predicted by assuming that the moments on the stationary and rotating portions are equal; then solving the boundary-layer equations to find the unique value of core velocity, β, which satisfies the wall shear stress requirements for the stationary and rotating surfaces. This is valid regardless of the degree of wetting of the disk. This frictional loss can be represented in terms of a dimensionless moment coefficient.

3. Four different flow regimes may be identified according to the value of disk Reynolds number and the spacing ratio between the disk and stationary wall. For each of the regimes, analytical expressions have been developed for prediction of the pressure gradient developed, and the frictional resistance. They are uniquely dependent upon the values of disk Reynolds number, spacing ratio, and engagement ratio.

4. Experimental results for two different flow regimes have confirmed the predictions of the partially wetted disk theory of reference [1], in particular, the changes that take place as the engagement ratio (size of the fluid ring) is varied. Some amount of deviation from the analytical predictions has been found, but this does not significantly affect the design practically of the results.

5. The flow characteristics of the disk indicate that it does pump as expected; however, the pressure gradient is affected, even at the smallest test flows. The increase in torque due to pumping may be adequately estimated by a simple analytical expression.

Slinger Seal Design

Some generalizations about the performance of the slinger seal are necessary to assure meaningful application of the design information. For example, the values of K has been noted to remain fairly constant over a range of spacing ratios in the turbulent regime IV. Since it is desired to operate at the highest value of K possible to generate the largest pressure gradient it may become necessary to go to very narrow spacings that would bring it into Regime III. The defined lines on Figure 2 which denote the various flow regimes tend to indicate that there is a sharp transition between flow regimes. This is not the case, in fact it is possible for two different regimes to exist on the same rotating disk. Thus, it is not possible to specify exactly the value of spacing ratio to be used, however, it will undoubtedly be less than .03 and probably less than .01. There is a penalty to pay in terms of increased power for the narrow spacing. From the experimental data, it is possible to predict that the power loss would be increased approximately 8 percent. It may also be necessary to operate at wider spacing ratios simply from the axial clearance requirements of the application.

Other observations from the experimental data are possible. The value of K decreases significantly for a reduction in the engagement ratio whereas the moment coefficient remains fairly constant over a wide range of engagement ratios to assure the higher values of K. For example the difference between operating at an engagement ratio of .4 and .8 for a spacing ratio of 0.1 can be detailed as follows. Operation at the higher engagement ratio means an 8 percent increase in power loss, however, the pressure rise is increased by approximately 67 percent. Therefore, it is obvious that the higher engagement ratios provide much better sealing without any substantial increase in power loss.

The effect of throughflow on the pressure gradient has been found to be significant, which would not be expected in terms of the natural circulatory flow in the housing. Thus, in attempting to cool a seal with throughflow, the decrease in performance must be considered. Preferably any cooling flow should be used on the side of the disk opposite to that where the maximum pressure gradient is desired. Under such conditions it can be expected that a temperature gradient will exist from one side of the disk to the other in proportion to the heat that is being generated and the amount of cooling flow used. Such thermal calculations are necessary for a high speed application.

Assume now that an application calls for sealing against a pressure of 115 psia from an ambient of 15 psia. For simplicity the sealing fluid is assumed to be water, which has many of the fluid characteristics of typical rocket engine fuels. A speed of 9000 rpm with an 8 inch disk is available. A disk spacing ratio of .06 is to be used. The fluid conditions are assumed as follows:

$$\text{Temp} = 100^{o}\text{F} \qquad \rho = .0357\ \text{lb}_m/\text{in}^3 \qquad \nu = .00102\ \text{in}^2/\text{sec}$$

The engagement ratio to meet this pressure requirement can be calculated from

$$\Delta p = \frac{\rho}{2g_o} \quad (K\omega \cdot a)^2 \left[1 - (1 - X)^2\right]$$

or in terms of X

$$X = 1 - \left[1 - \frac{\Delta p \cdot 2g_o}{\rho (K\omega \cdot a)^2}\right]^{1/2}$$

Using the above values and a first assumption for K of .44 we arrive at a numerical value for X of .54. At this engagement ratio, from Figure 7 it can be noted that the assumed value of K is conservative. A recalculation with the new value of K equal to .45 yields a new value for X equal to .495 or a change of approximately .16 inches in the position of the interface.

With regard to the power requirements, from Figure 5 for an engagement ratio of .495 the value of $C_M Re^{1/5}$ is approximately equal to .0370. The Reynolds number is given by

$$Re = \frac{\omega a^2}{\nu} = \frac{943 \cdot 16}{.00102} = 1.48 \times 10^7$$

Consequently $Re^{1/5}$ is calculated as 27.3, therefore C_M is equal to .001355. The frictional moment on the side of the disk is calculated from

$$M = C_M \times \frac{\rho \omega^2 \cdot a^5}{2g_o} = .001355 \times \frac{(.0357) \cdot (943)^2 \cdot (4)^5}{2\ (286)}$$

or $M = 57.2$ in-lbs

This means that the sealing side of the disk will consume approximately 8.2 horsepower.

The values used in the above calculations were based on the correlations from the experimental data. If the theoretical values had been used the predicted engagement ratio would be .46 with a predicted power consumption of 7.3 horsepower. This gives an indication of the magnitude of the discrepancy between predicted and experimental results.

In this application, depending upon the amount of sealing fluid used it is possible that both sides of the disk would be partially wetted. In this case, it would become necessary to consider both sides of the disk. In addition, for the power calculations, the disk tip friction should be calculated. The same procedures could be used for finding the engagement ratios with the exception that for a given amount of fluid, only one set of engagement ratios could satisfy the pressure requirements of the application. Under conditions of throughflow an appropriate correction should be made to the value of K on the side of the disk where throughflow is allowed, otherwise the procedure would be similar.

For calculating the tip torque, the following correlations are given in terms of a moment coefficient similar to the disk torque correlation. The torque is calculated in the same manner as previously indicated.

For laminar flow at the tip

$$C_{M_{TIP}} = \frac{4\pi b}{a^2 c} \cdot \frac{\nu}{\omega}$$

where b is the disk width and c is the radial clearance.

For turbulent flow

$$C_{M_{TIP}} = \frac{f \pi b}{8a}$$

where f is the turbulent friction factor for pipe flow given as $.316 \, (Re_{TIP})^{0.25}$ and Re_{TIP} is defined as $\omega \, ac/\nu$.

<u>Vanes</u> [2, 3]

The pressure head built up by the slinger seal can be increased by placing vanes on the rotating disk. The addition of tangs on the stator and corresponding recesses on the vanes improves interface stability.

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NOMENCLATURE

a	=	radius of disk (L)
b	=	radius of wall or width of disk (L)
C_m	=	disk torque coefficient $= \frac{2M}{\rho\omega^2 a^5}$ one side only
K	=	velocity ratio = β/ω
M	=	frictional disk torque (L-F), one side only
p	=	pressure (F/L^2)
Q	=	volumetric flow rate (L^3/T)
r	=	radius (L)
R	=	radius of interface (L)
Re	=	disk Reynolds number
R	=	inner radius of fluid (L)
s	=	axial clearance between disk and end wall (L)
X	=	engagement ratio $\frac{a-R}{a}$ or $\frac{b-R}{b}$
α	=	$v_o/(\omega-\beta)r$
β	=	angular velocity of fluid core (1-T)
δ	=	disk boundary-layer thickness or radial clearance (L)
μ	=	absolute viscosity ($F\text{-}T/L^2$)
ν	=	kinematic viscosity (L^2/T)
ρ	=	density (M/L^3)
ω	=	angular velocity of disk (1/T)

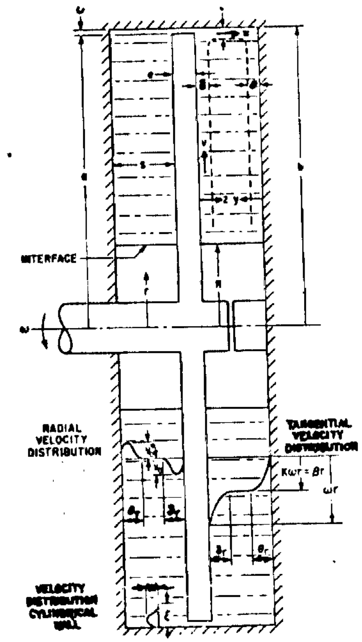


Figure 1. Schematic of Partially Wetted Disk.[1]

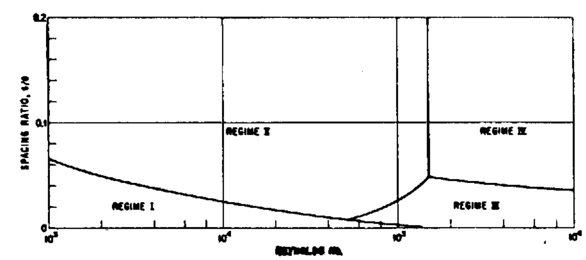


Figure 2. Approximate Operating Flow Regimes for Rotating Disks.[1]

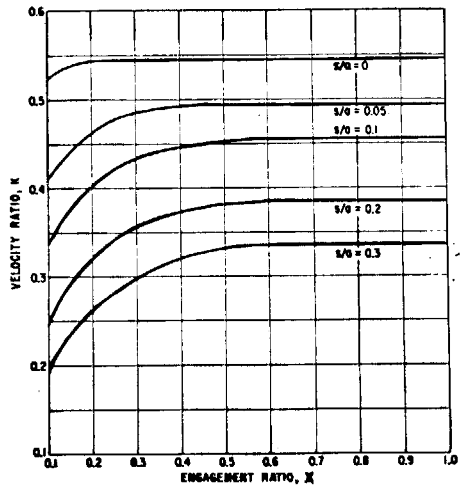


Figure 3. Velocity Ratio Versus Engagement Ratio for Various Spacing Ratios, Rotating Disk, Regime II. (1)

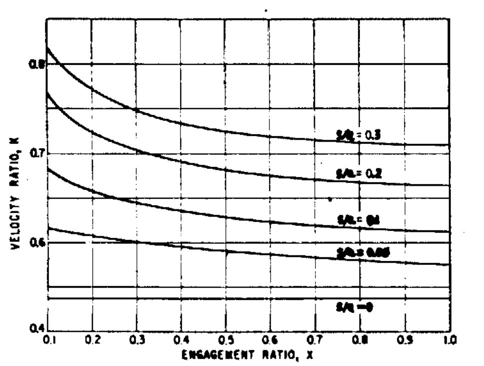


Figure 4. Velocity Ratio Versus Engagement Ratio For Various Spacing Ratios, Rotating Housing, Regime II. (1)

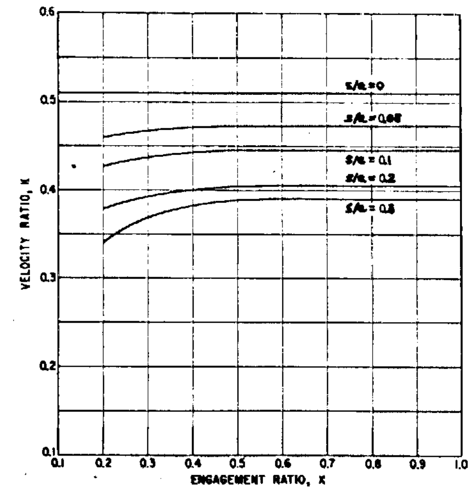


Figure 5. Velocity Ratio Versus Engagement Ratio For Various Spacing Ratios, Rotating Disk, Regime IV. (1)

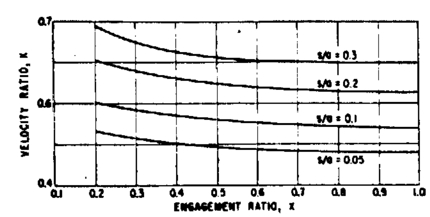


Figure 6. Velocity Ratio Versus Engagement Rotating Housing, Regime IV. (1)

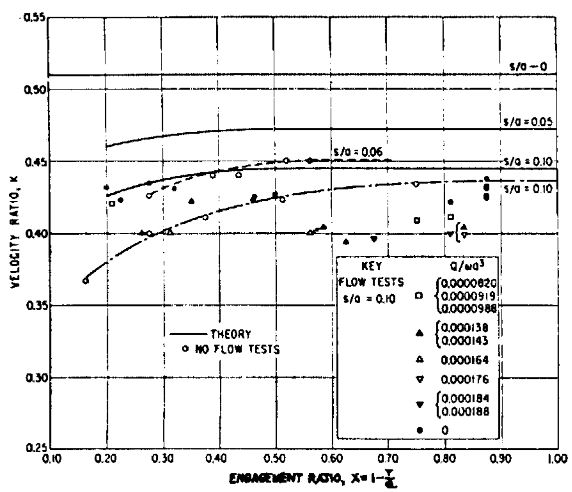


Figure 7. Velocity Ratio Versus Engagement Ratio. (1)

G. LABYRINTH SEAL

The purpose of this article is to assess the state of present knowledge of labyrinth seals. A labyrinth seal is widely used to minimize gas leakage along a rotating shaft as in Figure 1. It basically consists of a series of restrictions placed in the leakage flow path. Labyrinth seals are used where it is not feasible to permit direct contact between sealing members, and as a consequence have a relatively high rate of leakage compared to a "contact" type of seal. "Zero" leakage therefore can only be obtained by establishing auxiliary flow circuits to remove the leakage as in Figure 1d. Labyrinth seals are particularly suited to sealing gases, and in many instances are the only type of seal that can be used.

The available knowledge about labyrinth seals is rated "fair" to "good" in accordance with our definition of the state of knowledge in each field. Three references of major importance, (1), (2), (3), in the English language serve to cover the art of labyrinth sealing. When dealing with fluids and seal configurations for which test data is available, knowledge is rated as "Good" (25% design accuracy). When dealing with different configurations, or using fluids that deviate considerably from ideal properties, the knowledge is rated as fair or poor, and testing becomes necessary in the design.

The labyrinth seal functions by:

a) Restricting the cross-sectional area of the leakage flow path. Thus the clearance between seal and shaft that forms the flow path is made as small as possible. The minimum clearance possible is limited by shaft deflections, the variations in fluid film thickness in the bearings supporting the shaft, unequal thermal expansions, etc.

b) Taking advantage of the large specific volume of a gas. For a given pressure drop across a seal, the less dense the fluid, the lower the resultant mass flow rate. This is one of the reasons that a labyrinth seal that is quite effective in sealing a fluid in its gaseous state may be quite useless to seal the same fluid in its liquid state.

c) Creating a turbulent flow path with as much internal friction as possible in the fluid, thereby making for an inefficient flow process to produce low leakage. This is the direct opposite of conventional flow through nozzles and pipes, where it is desired to make for as little friction as possible.

High turbulence and frictional losses are produced in the seal by expanding the gas as it passes through the orifices formed by successive restrictions, thereby converting the energy of the gas into kinetic energy, then attempting as space and geometry permits to dissipate the kinetic energy into heat by changing the direction of flow between successive restrictions. Several labyrinth seals that do this are illustrated in Figure 1b and 1c. The sources of the energy that must be dissipated in this way are: (1) the pressure drop acting across the seal to cause the fluid to flow, (2) the energy stored in the gas because it is initially under pressure, and (3) the energy the fluid may initially possess because of its temperature. The sum of the energy thus contained in the gas as a consequence of its pressure and temperature state is termed the enthalpy of the gas. Item (2) is large for gases, but negligible for liquids because a liquid is relatively incompressible. This fact is of prime importance in determining the basic actions within a labyrinth seal and makes it a device suited primarily to gas properties rather than liquid properties.

Many have investigated labyrinth seals. The approach is largely experimental because of the difficulty of analysis. The University of Tennessee (1) has recently made a literature survey of labyrinth sealing, noting the basic analytical approach used by each investigator. There are two starting points: i.e., considering the seal as a nozzle, or as turbulent pipe flow. Neither approach is exact; the actual process lies somewhere in between. The analysis must also take into account the varying properties of the gas because of the large change in pressure and density of the gas as it passes through the seal. Hence the simple basic equations of the above processes must be modified; and the useability of the modification (if simple) depends upon the conditions of use. Equations requiring successive trial and error solutions often result. Graphical solutions are also used.

In addition, much of the existent test data has been obtained using steam. Historically, labyrinth seals were developed for steam turbines. Hence, ideally, further corrections should be made for the deviation of steam from the perfect gas laws when applying this data to other gases. Deviations from the perfect gas laws become insignificant at low pressures. Simple, elemental gases, higher molecular weight gases such as O_2, N_2, etc., do not deviate significantly except at very high pressures; higher molecular weight gases such as CO_2, SO_2, H_2O, freons, etc. show significant deviations at moderate pressures, such as may be encountered in rocket engines.

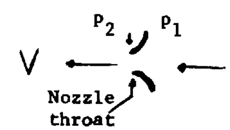
It is therefore not surprising to find that Zabriskie and Sternlicht (2) who have briefly investigated labyrinth sealing literature, note errors of as much as 200% in estimating leakage. Their contribution has been to treat the labyrinth as a rough pipe, replotting data from different sources on a comparable basis employing the Reynolds number, thereby enabling them to select material they believed most useful, to achieve according to their estimation, a probably accuracy in estimating leakage of 20%, using a fairly rapid computational procedure. They lament the lack of adequate flow information for many cases.

Reference (3) although not recent, is recommended because it enables one to rapidly obtain an appreciation of the variables affecting labyrinth seals. Extracts from this reference are given in this article.

It is worthwhile to consider the elementary forms of the equations that can be used to treat flow through a labyrinth. This will give a qualitative appreciation of the significant variables that affect labyrinth sealing to help point out the knowledge needed for seal design.

First, considering an incompressible fluid (a liquid rather than a gas), and using nozzle and turbulent pipe flow relations side by side:

<u>Nozzle</u>

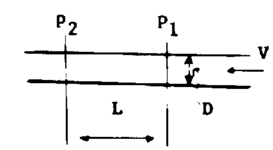


$\Delta p = p_1 - p_2$

H = Head causing flow $= \frac{\Delta p}{\rho}$

f = Friction factor
ρ = Density
V = Velocity
A = Area
Q = Mass flow rate
L = Pipe Length

<u>Pipe Flow</u>



$$V = \sqrt{2gH} \qquad \text{(1)}$$

$$H = f\left(\frac{L}{D}\right)\left(\frac{V^2}{2g}\right) \qquad \text{(11)}$$

$$V = \sqrt{\frac{H}{f}\ \frac{D}{L}\ 2g}$$

$$Q = V \times A \times \rho = A\rho\sqrt{2gH}$$

$$Q = V \times A \times \rho = A\rho\sqrt{\frac{H}{f}\ \frac{D}{L}\ 2g}$$

But the fluid head $H = \frac{\Delta p}{\rho}$. Also $A = \frac{\pi D^2}{4}$. Substituting:

$$Q = A\,(\rho)^{.5}\,(\Delta p)^{.5}\,(2g)^{.5} \qquad \text{(2)}$$

$$Q = A^{1.25}\,(\rho)^{.5}\,(\Delta p)^{.5}\,(2g)^{.5}\left[\frac{1}{fL}\right]\left[\frac{4}{[illegible]}\right]^{.25} \qquad \text{(12)}$$

For a concentric annulus of clearance h, the flow area = A = h π d, where d = diameter of the annulus:

$$Q \propto h \quad \text{———} (3) \qquad Q \propto h^{1.25} \quad \text{———} (13)$$

Flow Area

Putting the equations in the above form shows that the flow cross-sectional area A of the flow path is the most important variable affecting the leakage rate. Hence a major effort is made in constructing a seal to reduce the clearance between shaft and seal.

In fact, clearances may be used that are so small that occasional contact may take place between seal and shaft. The seal may be made of a soft material, and the shaft then run in to wear away the edges of the seal restrictions. On the other hand, the sea edges may be of hard materials, and carbon used as the material capable of wearing. In some constructions (Figure 2), the metal in contact with the edges of the seal is made of a light honeycomb structure so that controlled damage will result. The use of honeycomb introduces the disadvantage that additional leakage takes place through the honeycomb spaces that provide additional flow channels across the edges of the seal.

If sufficient deflection of the shaft occurs to make the annulus forming the leakage path appreciably eccentric, increased leakage will result. For turbulent pipe flow in a circular annulus, the maximum increase in flow (upon contact) is 30%. (4). Flow rate vs. eccentricity is plotted in Fig. 4. Thus it is desirable to decrease both clearance and eccentricity. "Floating" constructions are sometimes used to permit clearances tighter than the total variation in shaft position that occurs as a turbine operates.

The effective flow area at a restriction, depending upon geometry and flow conditions, may also be reduced in a manner analogous to the vena contracts of a constant-cross section nozzle (Fig. 3a) to produce as much as 30% variation in flow. To correct for this contraction in a nozzle, the coefficient of discharge C_d, is introduced into the nozzle formula:

$$Q = C_d \sqrt{2gH} \quad \text{———} (4)$$

Depending upon upstream flow conditions, C_d, for a constant cross-section nozzle or a square edged orifice is in the order of .6 to 1.0. Comparable variations in flow are found for labyrinth seals (Fig. 5a).

Gas Density

Equations (2) and (12) also illustrate the desirability of using a low density gas to reduce the mass flow Q. In each equation:

$$Q \propto \sqrt{\rho} \quad \text{———} (5)$$

Thus a gas at a low pressure (approaching a vacuum) can have a low mass flow rate even though it may actually have a fairly high flow velocity expressed as volume/time. Hence it is desirable to expand a gas to as low a pressure as possible before allowing the gas to emerge from the seal.

Viscosity

The conclusion reached in the previous paragraph remains essentially true even when considering the reduction in viscosity that occurs as the pressure on a gas is reduced to make the gas less dense. Viscosity is also reduced by a decrease in temperature. Both flow processes being considered for illustration are relatively independent of viscosity. For a nozzle, viscosity variations are usually taken into account by plotting the coefficient of discharge versus Reynolds number. This information is available in the literature. The friction factor f also takes into account viscosity variations for turbulent flow, but does not change greatly. To illustrate the order of magnitude of the effect of viscosity, test data on turbulent pipe flow yields the following experimental relations:

$$\Delta p \propto \mu^{.15 \text{ to } .25} \qquad \mu = \text{viscosity}$$

$$\propto D^{1.15 \text{ to } 1.25}$$

$$\propto V^{2.0} \quad \text{(For rough pipes)}$$

Nevertheless, estimating leakage for a gas radically different in viscosity than the gas used to obtain test data may require consideration of the viscosity, particularly for cryogenic gases.

Labyrinth Geometry

With suitable approximations, Egli (3) is able to obtain a simple form of equation that applies to steam through a series of labyrinth restrictions that is similar to the basic nozzle equation $Q = A\sqrt{2g \rho \Delta p}$:

$$Q = \alpha \, \phi \, \gamma \, A \sqrt{g \rho p_1} \quad \text{———} (6)$$

α ϕ γ, are introduced as variables to correct the flow. Using test data, Egli plots these variables as given in Fig. 5.

- α is a contraction factor taking into account vena contracta effects
- ϕ is a function of P_1/P_2, and the number of stages
- γ Carryover correction factor - to take into account the amount of energy not dissipated in successive stages

Thus, by means of Egli's plottings, one can readily see that increasing the number of stages brings diminishing returns in reducing leakage. In this way one is able to select the optimum point when balancing cost and bulk against leakage.

To illustrate the order of magnitude of the leakage in a labyrinth seal, let us use Egli's data on steam (Fig. 5). Assume a 3" diameter (d) concentric annulus with a .003" clearance (h) for the flow restriction; 12 stages of throttling; an initial pressure of 100 psia at saturation conditions, a final pressure of 0.1 psia.

$$Q = \alpha \phi \gamma \quad A \quad \sqrt{g \rho P_1}$$

Q = Leakage flow in pounds per second

A = Leakage area in $ft^2 = \frac{\pi d h}{144} = \frac{\pi x 3 x 0.003}{144} = 0.000196$

P_1/P_2 = Pressure ratio = $\frac{0.1}{100} = 0.001$

P_1 = Initial Pressure = 100 x 144 = 14400 lb/ft^2

ρ = Initial density = $\frac{1}{\text{Specific Volume}} = \frac{1}{4.4} \frac{lb.}{ft^3}$

α = Contraction factor = 0.68

ϕ = 0.275

γ = 2.2 for $h/s = 0.1$

g = 32.2 ft/sec^2

$$Q = 0.68 \times 0.275 \times 2.2 \times 0.000\,196 \quad 32.2 \frac{1}{4.4} \times 14400 = 0.00826 \text{ lbs/sec.}$$

$$\times 3600 = \underline{\underline{29.7 \text{ lbs./hour}}}$$

However, suppose, as for a space application, that is is necessary to decrease both leakage and bulk. Then it would be desirable to apply data such

as Egli's smaller seals, using a Reynolds number plotting. Sternlicht and Zabriskie have made such plottings [2].

In general, the following information should be known to design a labyrinth seal. A deficiency in any one aspect may vitiate the accuracy of leakage estimation.

1. The net flow area at successive restrictions
2. Flow geometry vs. flow effects vs. Reynolds number; and viscosity and density to compute the Reynolds number
3. The thermodynamic properties of the gas; i.e. specific volume, enthalpy, entropy, vs. temperature and pressure
4. Adequate graphic or mathematical procedures to represent the leakage flow process; appropriate awareness when sonic velocity is reached, or of 2-phase flow conditions where liquid and gas co-exist.

In addition, if very small clearances are used, the ability of the materials to withstand rubbing is very important.

Stability

This subject will not be treated here. However, it will be noted that this is a very important consideration in some labyrinth seal applications. References [5] and [6] deal with this problem.

[5] Experience has shown that the noncontacting dynamic seals of high-speed engines require special mechanical design considerations for long life and good performance. Dimensional stability and structural integrity are compromised if certain mechanical criteria are not incorporated into the design.

[6] Labyrinth seals and associated structural elements have been noted to sustain fatigue failure in high powered, high-speed rotating machinery under circumstances that preclude blaming standing wave resonances or "stick-slip" excitation from rotor/stator rubs. It has been hypothesized that such failures may be caused by self-excited aeroelastic vibration of the seal.

References

1. Arnold and Stair, The Labyrinth Seal--Theory and Design, The University of Tennessee, Department of Mechanical Engineering, March, 1962.
2. Sternlicht and Zabriskie, Labyrinth Seal Leakage Analysis, Gen. Elec. Report 57GL279, Sept. 26, 1957, Gen. Engineering Laboratory, Schenectady, New York.

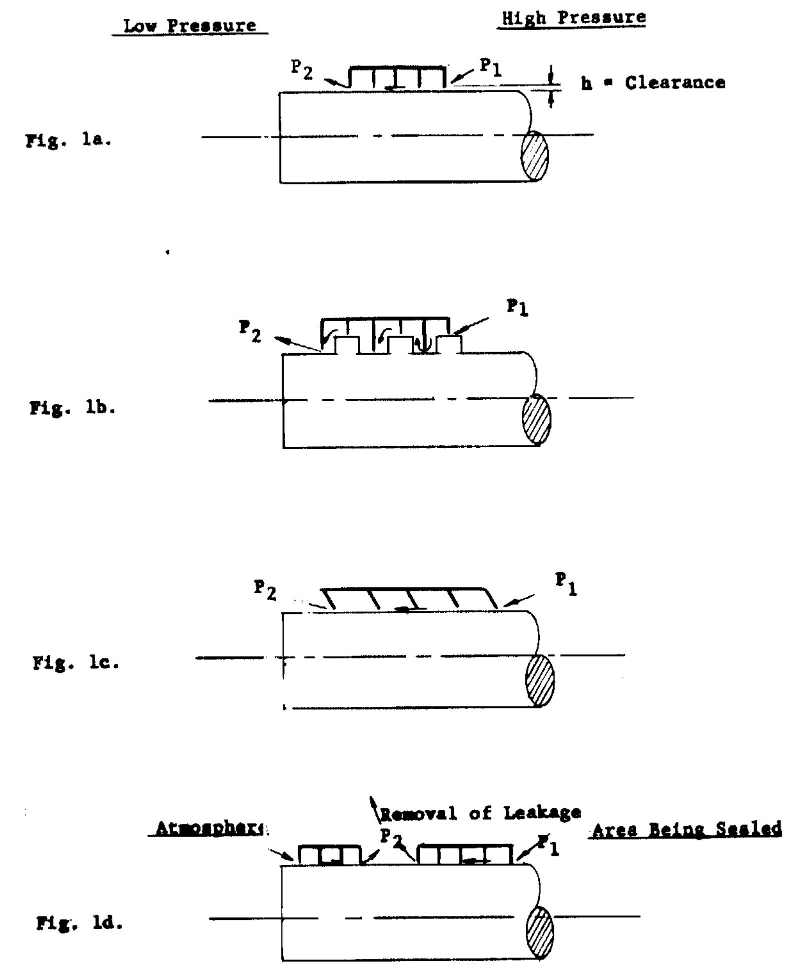
   Zabriskie and Sterlicht, Labyrinth-Seal Leakage Analysis, Trans. ASME, Vol. 81, Series D, 3, pp. 332-340.
3. Egli, The Leakage of Steam Through Labyrinth Seals, ASME Transactions, Vol. 57, 1935, p. 115-122.

References (con't.)

4. Tao and Donovan, Through Flow in Concentric and Eccentric Annuli of Fine Clearance With and Without Relative Motion of the Boundaries, Trans. ASME, Vol. 77, 1955, p. 1291

5. Alford, J. S. and Lawson, G. W., Dimensional Stability and Structural Integrity of Labyrinth Seals.

6. Ehrich, F., Aeroelastic Instability in Labyrinth Seals, ASME Paper 68-GT-32.



Pressure Drop = $\Delta p = P_1 - P_2$

Initial Pressure = P_1

Final Pressure = P_2

Pressure Ratio = $\frac{P_2}{P_1}$

Clearance Between Seal and Shaft - h

Figure 1. Labyrinth Seal

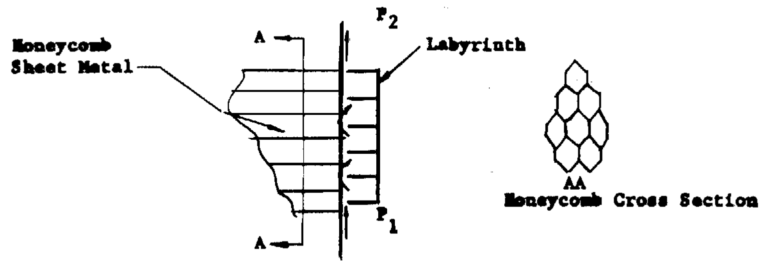


Figure 2. Honeycomb wearing surface in Labyrinth Seal

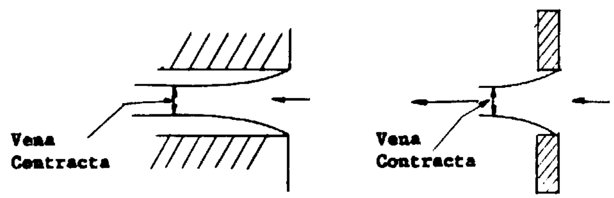


Figure 3a. Nozzle and Orifice

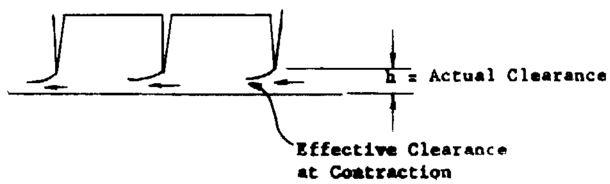
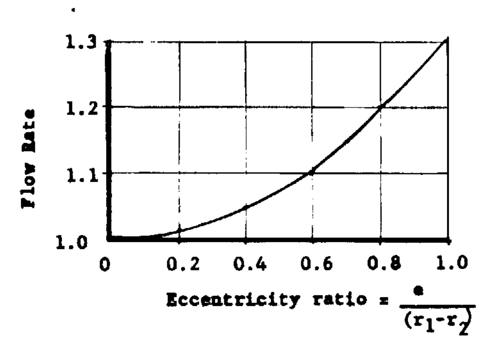


Figure 3b. Seal

Figure 3. Flow Contraction Effects



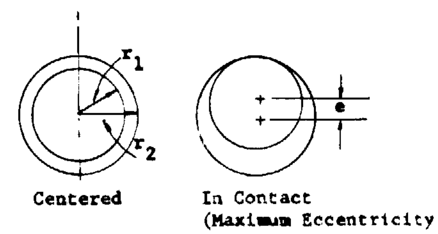


Figure 4. Flow Rate vs. Eccentricity - Turbulent Flow in Annulus

Reference (4)

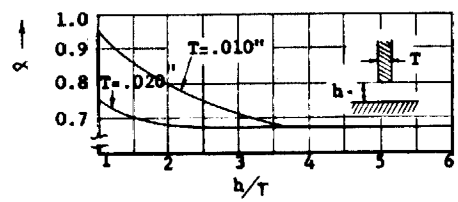
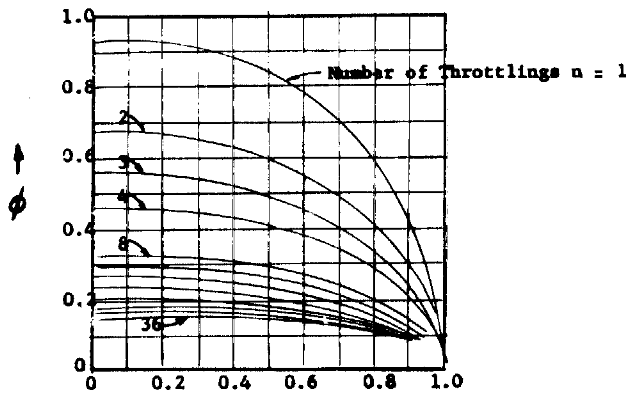
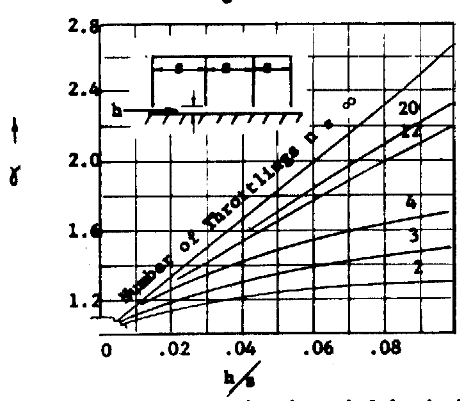


Figure 5a.



Pressure Ratio P_2/P_1

Figure 5b.



For Straight-Through Labyrinth

Figure 5c.

Figure 5. Labyrinth Seal Flow Factors Egli Reference (3)

H. LABYRINTH SEALS FOR EXPLOSIVE SQUIBS

I. Introduction

II. Description of the Labyrinths Seal

III. Problem Modeling

- Combustion Process
- Analytical Descriptions of the Flow and the Computational Procedure Flow from the Combustion Chamber into the First Land
- Flow in the First Land
- Flow into the First Groove
- Flow into Second Land

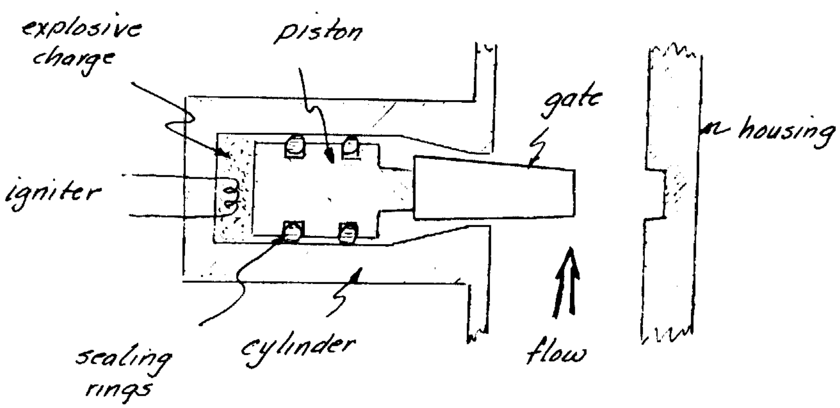
IV. Results

V. Summary

Introduction

Explosively operated hardware have been widely used in space applications where compactness, reliability and zero leakage are essential. The particular applications considered here are explosively operated values which have been operated successfully in space as presently designed. These valves are typically of the one-shot, hermetically sealed variety, where the explosive charge is used to drive a piston,and the attached valve,steam plus gate open or closed.

The accompanying sketch shows the schematic layout of such a valve in an open position ready to be fired. When the igniter



is triggered, the explosive charge burns rapidly driving the gate across the flow path stopping the flow. The housing is tapered so that the gate itself wedges in the housing around its perimeter, fusing the gate and housing together to form a leak-proof barrier to the flow.

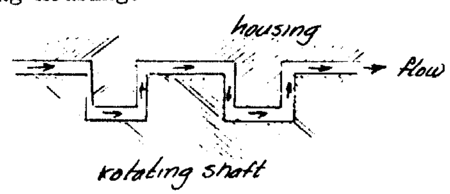
In addition to sealing off the through-flow, the gate or piston is usually designed to fuse with the cylinder at its base to prevent any contamination of the fluid by combustion products which might leak past the sealing rings. Leakage of this kind is a major concern, particularly when these valves are used in systems where the purity of fluid being controlled is important, such as in on-board scientific experiments. Here, even the slightest trace of combustion products in the controlled fluid could nullify the results of the experiment performed.

As it stands the device is relatively uncomplicated. It has been suggested, however, that if the sealing rings could be eliminated one of the less reliable components would be removed, and the design further simplified while retaining the all important zero leakage feature. It was noted earlier that the function of these sealing rings was to prevent combustion products from reaching the controlled fluid before the gate and cylinder fused together. The sealing device between the piston and cylinder, whatever it may be, is not required to be an absolute barrier to the flow of combustion products, only an impedance which delays the initiation of leakage. If leakage passed the piston occurs after the gate and cylinder fuse together, contamination of the controlled fluid is not possible and sealing device has served its purpose.

There are other methods which might be employed to restrict the flow for the necessary period of time. One of them, a non-contacting labyrinth seal,is the subject of this investigation.

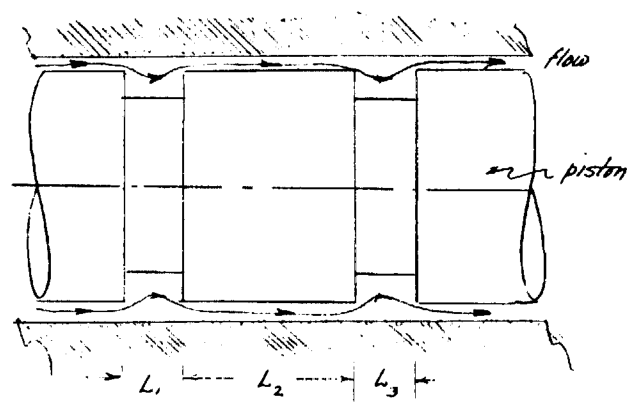
Description of the Labyrinth Seal

The labyrinth seal usually consists of a series of interlocking lands and grooves running circumferentially around a shaft and mating housing.



The tortuous flow path formed by these lands and grooves serves to restrict the axial flow of the fluid to be sealed. Such arrangements have been used successfully on rotating steam turbine shafts.

Because the piston must move axially to actuate the gate, an interlocking arrangement is not feasible here in the explosive squib application. If, however, grooves are cut circumferentially only around the piston it is possible that some of the labyrinth's tortuosity can be retained while allowing for the axial movement of the piston.



From the design standpoint it would be desirable to know just how the geometry of the annular clearance space affects the initiation of leakage flow, i.e., how many grooves should be used, how deep should they be, how wide should the lands and grooves be, etc.

It is not inconceivable that the effect of these parameters could be determined experimentally, and certainly sooner or later some testing must be done to determine their influence. Analysis, on the other hand, can establish trends and at least orders of magnitude, both of which are important to determining conceptual feasibility. As in any analysis certain assumptions are made to obtain a viable model of the actual process. These are described in the following section.

Problem Modelling

Combustion Process

The motion of the piston is initiated of course, by the explosion which is at best a complicated process. The final pressures following combustion reach 20,000 to 30,000 psi while the final temperatures range from 2500°F to 3000 °F. The actual explosion is not instantaneous so that the pressures cited above are not reached until somewhat after the flow of combustion products has started into the annular clearance space. For the purposes of this analysis it will be assumed, however, that the combustion process does occur instantaneously. This simplification will naturally result in predicted leakage initiation which is faster than can be actually realized in the real explosion process.

The combustion process results in products containing hydrogen, water vapor, carbon dioxide and vaporized metal salts. A good approximation for the specific heat ratio of the combustion products, especially at higher temperatures is $c_p/c_v = 4/3$. If it is estimated that the average $c_p = 0.28$ and $c_v = 0.21$, then the entropy rise of the gases during the combustion process (assuming an ideal gas) is

$$S_E - S_o = c_p \ln \frac{T_E}{T_o} - \frac{R}{J} \ln \frac{p_E}{p_o}$$

$$= 0.28 \ln \frac{T_E}{T_o} - 0.07 \ln \frac{p_E}{p_o}$$

Using typical values for the pressure and temperature ratios of

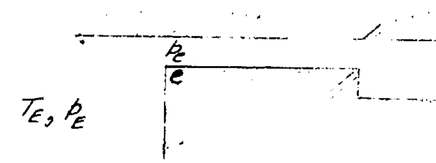
$$\frac{T_E}{T_o} = 6 \ , \ \frac{p_E}{p_o} = 1300$$

$$S_E - S_o = 0.28\ (1.79) - 0.07\ (7.17)$$

$$= 0.501 - 0.502 = - 0.001 \ \frac{BTU}{lb_m\ {}^\circ F}$$

Computations using other pressure ratios and associated temperature ratios yield similarily small entropy changes. It seems reasonable, therefore, to characterize the combustion products as undergoing a rapid isentropic process in the combustion chamber starting at ambient pressure and temperature.

Flow from the Combustion Chamber to the Labyrinth

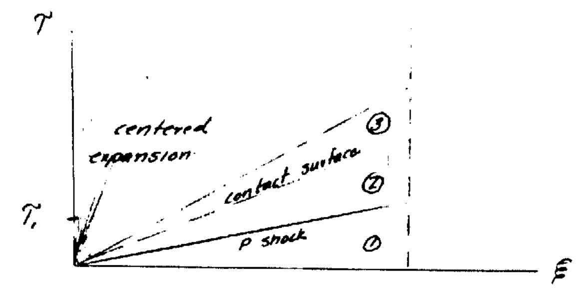


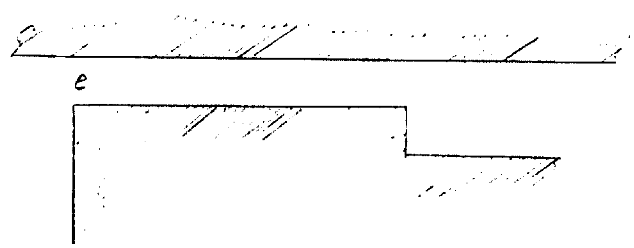
Assuming that the combustion process takes place very rapidly yielding almost instantaneously a very high temperature and high pressure gas (T_E, p_E), the flow from the combustion chamber into the labyrinth is essentially the flow from a large reservoir through at small restriction. The final pressure ratio across the restriction is well on excess of the critical value.

Actually the critical pressure ratio (approximately two) is surpassed very early in the combustion process. Once this ratio is reached the flow at the entrance becomes choked at Mach Number equal one. This does not mean that the flow velocity at the entrance to the first land does not change. It increases as the speed of sound increases and that increases in proportion to the square root of the absolute temperature at the entrance.

Flow Through the First Land of the Labyrinth

The sudden burst of combustion products through the constriction at (e) results in the generation of a shock wave propagating into gases already residing in the land clearance space.



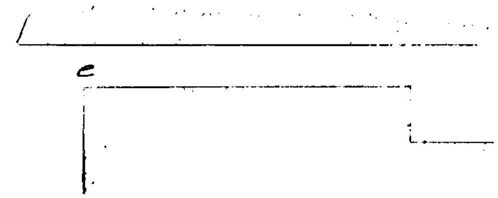


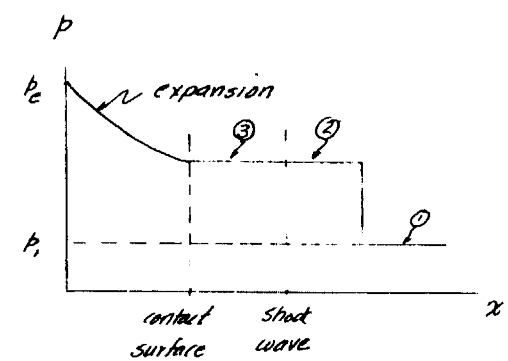
This shock front propagates into the uncontaminated gas with a velocity which depends on the strength of the shock, i.e., the pressure ratio across the shock.

Behind the shock front is region of steady flow consisting of the uncontaminated fluid which has been set in motion by the shock wave. This uncontaminated fluid is separated from the combustion products by the "so called" contact surface. Behind it is another region of steady flow this time of the combustion products.

The velocities and pressures in both of these steady flow regions are the same. Their temperatures are different, however, since they have vastly different histories starting from time-zero.

Behind this last steady flow regime and extending all the way back to the entrance (e) is a region of unsteady flow. The pressure distribution along the first land at a typical early time τ_1 (see τ - ξ figure) is shown in the accompanying sketch.





The mathematical description of the flow over the first land is based on the assumptions that the flow is

1. homentropic
2. one-dimensional
3. frictionless

These three assumptions permit the continuity and momentum equations to be written in the following form for unsteady flow

$$\frac{\delta_+ P}{\delta \tau} = \frac{\delta_- Q}{\delta \tau} = 0$$

where

$$P = \left(\frac{2}{\gamma-1}\right) a + u$$

$$Q = \left(\frac{2}{\gamma-1}\right) a - u$$

$$\frac{\delta_+}{\delta\tau} \equiv \frac{\partial}{\partial\tau} + (u+a)\frac{\partial}{\partial\xi}$$

$$\frac{\delta_-}{\delta\tau} \equiv \frac{\partial}{\partial\tau} + (u-a)\frac{\partial}{\partial\xi} \qquad *$$

These equations imply that there is no change of:

1. P along curves given by $\frac{d\xi}{d\tau} = u+a$ within regions where P and Q are continuous
2. Q along curves given by $\frac{d\xi}{d\tau} = u-a$ within regions where P and Q are continuous

Across shock waves or contact surfaces, special matching techniques must be applied, since these equations cannot be used when crossing from one region to another.

These non-linear equations show that changes in the quantities P and Q propagate throughout the space-time plane along the curves $\frac{d\xi}{d\tau}$, called characteristics. These disturbances in velocity and temperature, of course, give rise to disturbances in the remaining dependent variables, pressure and density. The simplifying assumptions mentioned above make it possible to solve the above equations without resorting to the clumbersome graphical procedures usually employed. The reason for this is that all of the characteristic curves needed to determine the position of the contact surface are straight lines.

* See Wave Diagrams for Nonsteady Flow in Ducts by Rudinger p.18

Fortunately, the location of the contact surface can be determined without knowing the details of the complex flow patterns which result from the interaction of shock waves at later times.

In practice, these equations are only used occasionally. Most of the analysis of unsteady flow deals with the matching of the various regions in which these equations apply, one to the other. The centered expansion at entrance to the first land is one of the regions where the equations for P and Q yield needed information. This unsteady expansion region results from the explosion in the combustion chamber. Since the explosion is rapid the values of P and Q rise very rapidly while the slopes of the characteristics also change drastically. The centered expansion is a consequence of assuming that the explosion is instantaneous. Since $u_e = a_e$ when choking occurs, the slope of the Q characteristic goes to zero very early in the combustion process. The slope of the P characteristic, on the other hand, continues to increase as u_e and a_e do. It is these characteristics which carry the disturbance caused by the explosion across the centered expansion region in the form of a P wave such that

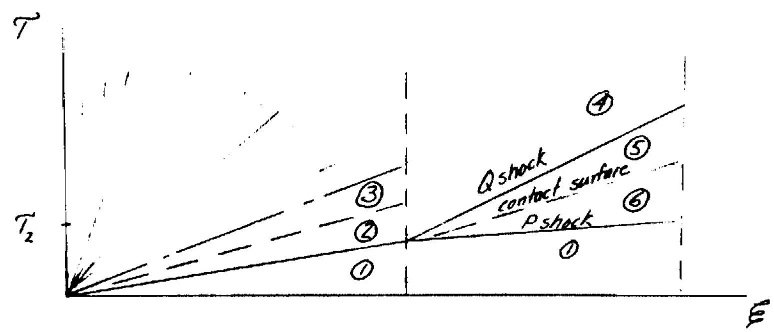
$$P_e = P_3 = \left(\frac{2}{\gamma_E - 1}\right) a_e + u_e = \left(\frac{\gamma_E + 1}{\gamma_E - 1}\right) a_e$$

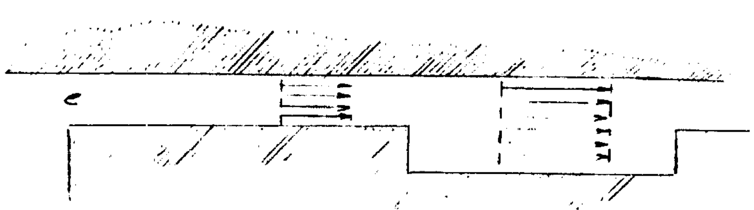
The boundary between the expansion and region 3 is found by matching the velocity change across the shock wave with the velocity in region 3. This matching is accomplished by a trial and error process, guessing the pressure ratio p_2/p_1

and using this guess in the Rankine-Huginiot equations for moving shock waves. These relations are the same as those used for stationary shock waves except that they are modified to take into account the motion of the shock wave. Once the velocities in regions (2) and (3) are equalized by trial and error, the speed of the combustion products through the first land is determined.

Flow Across on Area Discontinuity - Widening

When the shock wave reaches the area discontinuity it generates a transmitted P shock, a Q shock and a new contact surface. This contact surface is the interface between the ambient gas originally in the first





SHALLOW GROOVE

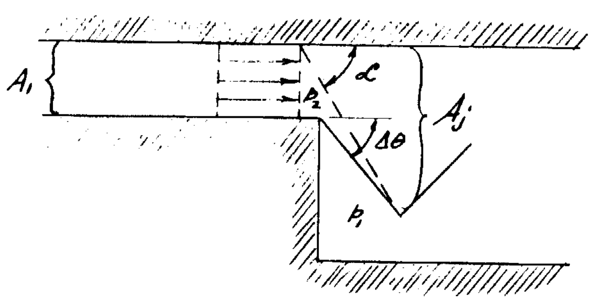
land area and the ambient gas originally in the first groove area. From the point of view of contamination this contact surface is of little interest since the combustion products are in region (3) behind the previously formed contact surface between regions, (2) and (3).

Preliminary calculations show that the contact surface travels with a velocity which just slightly less than the shock wave so that region (2) is very small. In order to avoid the rather complex calculations associated with the wave interactions which occur in the groove when the shock wave and the contact surfaces arrive at different times, it will be assumed in the following calculations that the shock and contact surface are coincident and only the combustion products flow into the groove.

It should be pointed-out that the generation of the shock waves and contact surface in the groove as shown in the sketch occurs only if one-dimensional flow is re-established in the groove. For "deep grooves" this may not happen. It is obvious, therefore, that before the analysis is carried any further it will be necessary to establish criteria for "deep and shallow" grooves.

If the inertia of the high velocity gas leaving the land area prevents the gas from expanding to completely fill the groove area, the result is analogous to what happens in an under-expanded jet (see sketch). The pressure turning angle $\Delta\theta$ for such jets is given by

$$\Delta\theta = \frac{\left(1-\frac{p_1}{p_2}\right)\sqrt{M_3^2-1}}{\gamma_E\, M_3^2}$$



DEEP GROOVE

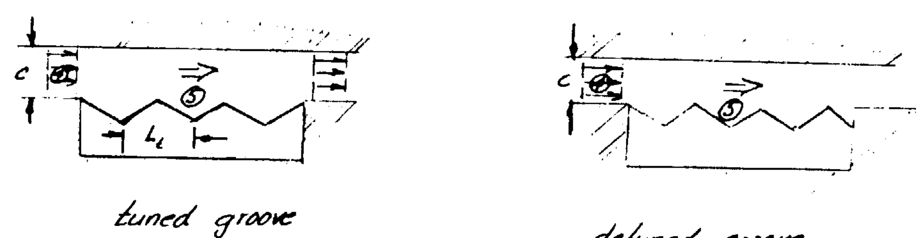
The angle of the Mach line (α) is given by

$$\tan \alpha = \frac{1}{\sqrt{M_3^2 - 1}}$$

The ratio of the area A_1 to be expanded jet area A_j is given by

$$A_c = \frac{A_1}{A_j} = 1 - \sqrt{M_3^2 - 1}\ \tan\left[\frac{\left(1 - \frac{p_1}{p_2}\right)\sqrt{M_3^2 - 1}}{\gamma_E M_3^2}\right]$$

If $A_c < A_R$ then the jet expands to fill the groove and one-dimensional flow results. If, on the other hand, $A_j < A_2$ so that $A_c > A_R$ the jet continues to maintain its identity for some distance down the groove before viscous effects cause it to diffuse. If the jet can maintain its identity reasonably well for the entire length of the groove it is possible that the groove may have "tuning lengths" which



DEEP GROOVE

permit a large carry-over of the energy from one land to another. A tuned groove would have a length which is a multiple of the basic tuning length L_t given by

$$L_t = c\left\{\frac{2\sqrt{M_3^2-1}}{1-\sqrt{M_3^2-1}\,\tan\left[\frac{\left(1-\frac{p_1}{p_2}\right)\sqrt{M_3^2-1}}{\gamma_E M_3^2}\right]}\right\}$$

For the large anticipated values of M_3 and $\frac{P_2}{P_1}$

$$\frac{L_t}{c} \approx \left(\frac{2\gamma_E}{\gamma_E - 1}\right)M_2 \sim 10 \text{ to } 20$$

Since c is of the order 10^{-4} inches, L_t will be of the order 10^{-3} inches. For large values of M_3 and P_2/P_1, A_c is of the order 1/4 so that the maximum width of the jet is of the order $4c$ or $4(10^{-4})$ inches.

From the standpoint of reducing leakage a "detuned" groove is certainly most advantageous. The exact hydrodynamic consequences of having a detuned groove are not clear except to say that the gas does not carry -over smoothly from one land to the next. From the leakage standpoint the tuned groove is the least desirable but the easiest to analyze, at least approximately. Computations based on this model will certainly yield transit times for the contact surface which are the shortest possible, with the result that the contact surface speeds computed for the tuned groove are higher than any actually realized.

The computer program, which will be described later in this report, will branch at the entrance to the first groove depending on whether the test described above shows the groove to be "deep"or "shallow". If the groove is "deep" the flow from the land to the tuned groove can be considered to be steady and isentropic, expanding from the groove area to the area A_j. Since the Mach number in region 3 is supersonic, the flow across the discontinuity is analogous to the flow in the diverging portion of a nozzle when the Mach number is unity at the throat. This means that the flow on the groove has a higher Mach than the land area Mach number. This Mach number may be easily computed using the steady, isentropic relation of gas dynamics

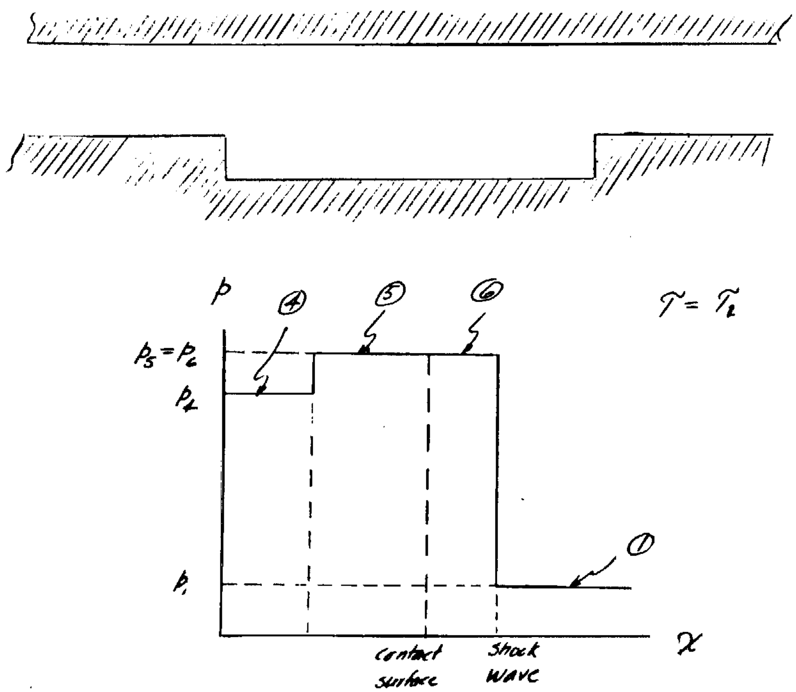
$$AD(M) = \frac{AM}{\left[1+\left(\frac{\gamma+1}{2}\right)M^2\right]^{\frac{\gamma+1}{2(\gamma-1)}}} = \text{constant}$$

which relates the groove Mach number to the land Mach number. Once the groove Mach number is determined all of the other dependent variables in the carry-over flow can be determined using the isentropic path functions.

$$\frac{P}{\rho^{\gamma}} = \text{constant}$$

$$\frac{T}{p^{\frac{\gamma-1}{\gamma}}} = \text{constant}$$

If on the other hand it is found that the groove is shallow, the computations are much more complicated as indicated by the earlier sketch. The Mach number in region 4 is calculated using the steady flow relation AD(M) = constant, the same as in the deep groove case. In contrast to the deep groove case, however, two shocks form in the groove. One shock, a P shock, advances into the motionless gas already in the groove. The



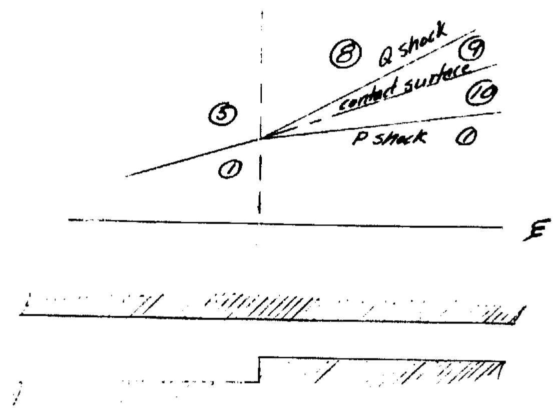
other shock, a Q shock, tries to advance upstream into the combustion products but is swept downstream because its velocity of propagation into the combustion products is less than the velocity of the combustion products.

The pressures in regions 5 and 6 (see sketch for $\tau = \tau_1$) are determined by a trial and error process, guessing the pressure in these regions (noting that $p_5 = p_6$) and calculating the velocities $\mathcal{U}_5$ and $\mathcal{U}_6$ from the Rankine Huginiot equations applied to the P and Q shocks. When $\mathcal{U}_5$ and $\mathcal{U}_6$ are equal the guess for $p_5 = p_6$ is correct and all the other dependent variables in regions 5 and 6 can be determined. The velocity of the contact surface is equal to $\mathcal{U}_5$ which is the most important output computed.

Flow Across Area Discontinuity - Contracting

Once again order of magnitude calculations show that the region between the shock wave front and the contact surface (region 6) is very small. When calculating the flow from the first groove into the second land it will again be convenient to assume that the contact surface and shock wave are coincident.

There are three ways in which the flow may enter the second land area from region 5. The simplest transition occurs when the area change is very small. If the flow in region 5 is steady and supersonic the flow through the area contraction is analogous to that in a converging nozzle, where a reduction in area leads to a reduction in Mach number toward unity. In order to reduce the Mach number below unity the cross-sectional area must first decrease and then increase, which it does not do in this case. This means that there will be a range of area ratios, depending on the Mach number in region (5), for which an M_8 can be found from AD(M) = constant such that $1.0 \leq M_8 \leq M_5$. When an M_8 in this range can be found, the calculations in the second land area are performed in exactly the same way as they were for the first groove

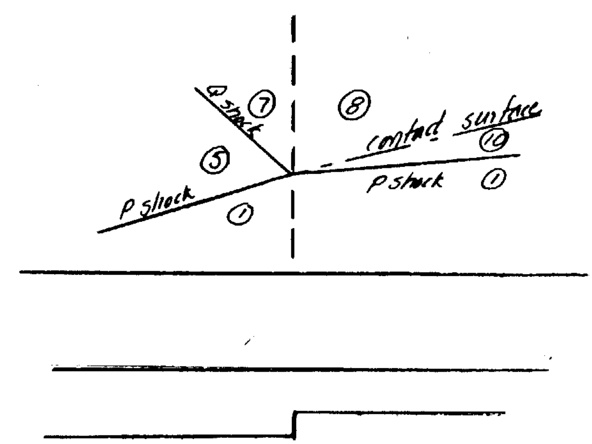


when it is classified as shallow. The pressure distribution in the second land also resembles that in the shallow groove.

If it was determined that the first groove is deep so that one-dimensional flow does not fill the groove area, M_8 will be equal to M_3, since no shocks form within the carry-over flow. All the other important dependent variables in region (8) can also be computed using the isentropic path functions and the values of the dependent variables in region (3). Velocities and pressures in regions (9) and (10) are found in the same way as described above.

It is quite possible that no value of the Mach number in region (8) can be found for the shallow groove case such that $1.0 \leq M_8 \leq M_5$. The situation is once again analogous to supersonic flow in a converging nozzle when the pressure ratio across the nozzle is not equal to that required for isentropic flow with a sonic or subsonic exit. In order to accomodate the imposed pressure ratio, the supersonic flow will undergo a Q shock to subsonic flow and then accelerate once again toward Mach number equal one at the minimum cross-section. The velocity of the Q shock must be larger than the velocity in region (5) so that the Q shock will propagate upstream into the combustion products.

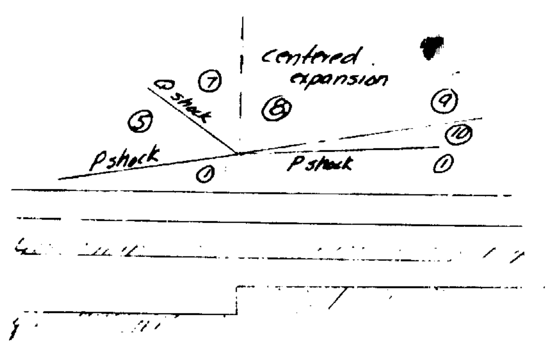
Once again the steady flow isentropic relation AD(M) = constant imposes a limit on the Mach number for converging flows. This time $M_7 \leq M_8 \leq 1.0$. Computations begin by guessing the



pressure in region (7) in such a way as to yield a sonic or subsonic M_8. The guess for p_7 will probably have to be altered several times depending on the method by which the flow establishes itself in the second land.

It is possible that steady flow is established in region (8) which in turn generates a P shock and a contact surface. If this happens the pressure in region (8) is calculated from the pressure in region (7) using the isentropic path functions. This pressure determines the shock strength and the velocity in region (10) which must be equal to the velocity in region (8). When a solution of this type exists it will be found by a trial and error process based on guessing p_7 and matching u_8 to u_{10}.

It is quite possible that a solution of the type just described cannot be found either. The reason for this is the upper limitation placed on $M_8 \leq 1.0$. If, in the process of trying to match the velocities in regions (8) and (10) it should be necessary to set M_8 equal to unity, the flow at the entrance to the second land becomes exactly analogous to the choked flow at the entrance to the first land. The flow in region (8) is then unsteady and can be approximated by a centered expansion. The flows on either side of the contact surface are steady with the same



velocity, while a P shock propagates into originally stationary gas in the second land area.

When the flow in the second land area is determined, by what ever means, enough information is then available with which to decide whether the labyrinth is sufficiently restrictive to warrant further computations or not.

Analytical Descriptions of the Flow and the Computational Procedure

Flow from the Combustion Chamber into the First Land

The analysis begins by specifying the following inputs:

1. The temperature of the gas initially in the labyrinth (T_o).
2. The final temperature of the combustion products in the combustion chamber (T_E).
3. The specific heat ratios of the combustion products (γ_E) and the gas initially in the labyrinth (γ_1).
4. The area ratio (clearance ratio) of the land to the groove (A_R).
5. The lengths of the land and grooves L_1, L_2, L_3, etc.

Since the combustion process was found to be almost isentropic starting at the initial temperature (T_o), this means that the combustion products have the same entropy as the stagnant gas ahead of it in the labyrinth. The passage of the combustion products from the combustion chamber is also assumed to be an isentropic process, which means that the dimensionless sound speed at the entrance (e) can be computed from the formula

$$a_e = \sqrt{\left(\frac{2}{\gamma_E+1}\right)\frac{T_E}{T_o}}$$

1. The Mach number of shock front

$$M_{S_{12}} = \sqrt{\frac{\frac{p_2}{p_1} + \frac{\gamma_1 - 1}{\gamma_1 + 1}}{\frac{2\gamma_1}{\gamma_1 + 1}}}$$

2. The density ratio across the shock front

$$\frac{\rho_2}{\rho_1} = \frac{1 + \left(\frac{\gamma_1 + 1}{\gamma_1 - 1}\right)\frac{p_2}{p_1}}{\frac{\gamma_1 + 1}{\gamma_1 - 1} + \frac{p_2}{p_1}}$$

3. The dimensionless sound speed ratio across the shock front

$$\frac{a_2}{a_1} = \frac{\rho_1}{\rho_2} \frac{M_{S_{12}}}{\sqrt{\frac{1 + \left(\frac{\gamma_1 - 1}{2}\right) M_{S_{12}}^2}{\gamma_1 M_{S_{12}}^2 - \left(\frac{\gamma_1 - 1}{2}\right)}}}$$

4. The dimensionless velocity change across the shock front

$$\frac{u_{12}}{a_1} = M_{S_{12}} - \frac{a_2}{a_1}\sqrt{\frac{1 + \left(\frac{\gamma_1 - 1}{2}\right) M_{S_{12}}^2}{\gamma_1 M_{S_{12}}^2 - \left(\frac{\gamma_1 - 1}{2}\right)}}$$

In these equations a_1 is the dimensionless sound speed of the stagnant gas initially in the clearance space. Having established the temperature (T_o) and the entropy of this gas as the reference values, this means that $a_1 = 1.0$ so that $a_1 = a_o$, where a_o is the sound speed of the undisturbed gas in the labyrinth. The result is that the velocity of the contact surface is equal to $\frac{u_{12}}{a_1}$ times the speed of sound in the stagnant gas a head of the shock wave.

Since the value of P_e is constant across the centered expansion, that means that $P_e = P_3$, so that

$$P_3 = \left(\frac{2}{\gamma_E - 1}\right) a_3 + u_3 = P_e$$

The matching conditions for regions (2) and (3) are

$$p_2 = p_3$$

$$u_{12} = u_3$$

The temperatures in regions (2) and (3) are not necessarily equal since the histories of the combustion products and the uncontaminated gas are quite different. This means that even if the combustion products and the undisturbed gas in the land area were the same gas, there would still be a contact surface which in this case would be a thermal or temperature interface.

Combining and solving the above equations for the value of a_3 yields

$$a_3 = \frac{P_e - \frac{u_{12}}{a_1}}{\left(\frac{2}{\gamma_E - 1}\right)}$$

Noting that the entropy of the gas in region (3) is still the same as the entropy of the gas in region (1), the pressure ratio across the shock is therefore given by

$$\frac{P_2}{P_1} = \frac{P_3}{P_1} = \left(a_3\right)^{\frac{2\gamma_E}{\gamma_E - 1}}$$

If the ratio p_2/p_1 was guessed correctly this calculation should yield that same guessed value. Usually it will be necessary to adjust the initial guess for p_2/p_1 until agreement is reached, and this calls for a trial and error computational procedure.

The most important outputs upon successful completion of the trial and error calculation, are the values of $\mathcal{U}_{12}/a_1$, $M_3 = \dfrac{\mathcal{U}_{12}/a_1}{a_3}$ and p_2/p_1. The last two quantities will be useful when computing the flow across the area discontinuity from the land to the groove.

Flow into the First Groove

At this point it must be decided whether the groove is "shallow" or "deep". The critical area ratio (A_c) quite naturally depends on how fast the combustion products flow out of the first land area. The maximum cross-sectional area occupied by the expanded flow in the groove is determined by locating where the pressure turning line and the Mach line intersect. When $A_c > A_R$ the expanded flow does not fill the groove but instead forms its own boundary. The maximum velocity with which the combustion products could advance across the groove (assuming complete carry-over in a tuned groove) can be estimated by assuming that the flow is roughly one-dimensional and shockless within this boundary. Since the flow in region 3 is steady, the flow across the discontinuity can be calculated using the steady, isentropic relation A D(M) = constant. For the problem at hand, the application of this equation leads to

$$\frac{M_5}{\left[1+\left(\frac{\gamma_E-1}{2}\right)M_5^2\right]^{\frac{\gamma_E+1}{2(\gamma_E-1)}}} = \frac{A_c M_3}{\left[1+\left(\frac{\gamma_E-1}{2}\right)M_3^2\right]^{\frac{\gamma_E+1}{2(\gamma_E-1)}}}$$

The value of M_3 is known from the previous land computations, so that it is only necessary to compute M_5 by a trial and error process. Once M_5 is found, energy considerations yield

$$\frac{a_5}{a_3} = \sqrt{\frac{1+\left(\frac{\gamma_E-1}{2}\right)M_3^2}{1+\left(\frac{\gamma_E-1}{2}\right)M_5^2}}$$

from which

$$\mathcal{U}_5 = M_5\left(\frac{a_5}{a_3}\right)a_3$$

The flow into and out of the groove is isentropic so that it is also possible to write the conditions at the entrance to the next groove as

$$\mathcal{U}_8 = \mathcal{U}_3$$
$$M_8 = M_3$$
$$a_8 = a_3$$
$$p_8/p_1 = p_3/p_1$$

These values will serve as inputs to the calculations for the next land which will be described somewhat later.

If it happens that $A_c < A_R$, then the flow expands to fill the groove and one-dimensional flow is established across the entire groove clearance. Two shock waves are formed in the groove, one propagating into the stagnant gas already in the groove and one propagating into the combustion products. The Mach number in region (4) is determined from the steady flow, isentropic relationship

$$\frac{M_4}{\left[1+\left(\frac{\gamma_E-1}{2}\right)M_4^2\right]^{\frac{\gamma_E+1}{2(\gamma_E-1)}}} = \frac{A_R M_3}{\left[1+\left(\frac{\gamma_E-1}{2}\right)M_3^2\right]^{\frac{\gamma_E+1}{2(\gamma_E-1)}}}$$

Once again, a trial and error procedure is necessary to find M_4. Once it is found then the independent variables in this region are computed from steady, isentropic relations

$$\frac{p_4}{p_1} = \frac{p_3}{p_1}\left[\frac{1+\left(\frac{\gamma_E-1}{2}\right)M_3^2}{1+\left(\frac{\gamma_E-1}{2}\right)M_4^2}\right]^{\frac{\gamma_E}{\gamma_E-1}}$$

$$a_4 = a_3\sqrt{\frac{1+\left(\frac{\gamma_E-1}{2}\right)M_3^2}{1+\left(\frac{\gamma_E-1}{2}\right)M_4^2}}$$

The strengths of the two shock waves are established by first estimating the value of the pressure ratio from the formula

$$\frac{p_6}{p_1} \approx \frac{\gamma_1(\gamma_1-1)}{2}\left[\frac{a_4 M_4}{1+\frac{a_4}{\sqrt{p_4/p_1}}}\right]^2$$

With this estimate the Rankine-Huginiot equations yield for the P shock

$$M_{S_{16}} = \sqrt{\frac{\frac{p_6}{p_1}+\frac{\gamma_1-1}{\gamma_1+1}}{\frac{2\gamma_1}{\gamma_1+1}}}$$

$$\frac{\rho_6}{\rho_1} = \frac{1+\left(\frac{\gamma_1+1}{\gamma_1-1}\right)\frac{p_6}{p_1}}{\left(\frac{\gamma_1+1}{\gamma_1-1}\right)+\frac{p_6}{p_1}}$$

$$\frac{a_6}{a_1} = \frac{\rho_1}{\rho_6}\left[\frac{M_{S_{16}}}{\sqrt{\frac{1+\left(\frac{\gamma_1-1}{2}\right)M_{S_{16}}^2}{\gamma_1 M_{S_{16}}^2-\left(\frac{\gamma_1-1}{2}\right)}}}\right]$$

$$\frac{u_{16}}{a_1} = M_{S_{16}} - \frac{a_6}{a_1}\sqrt{\frac{1+\left(\frac{\gamma_1-1}{2}\right)M_{S_{16}}^2}{\gamma_1 M_{S_{16}}^2-\left(\frac{\gamma_1-1}{2}\right)}}$$

The pressure ratio for the Q shock can be determined from ie estimated value of p_6/p_1 and the previously calculated ılue p_4/p_1, i.e.,

$$\frac{p_5}{p_4} = \frac{\frac{p_5}{p_1}=\frac{p_6}{p_1}}{\frac{p_4}{p_1}}$$

The Rankine-Huginiot equations for the Q shock now yield

$$M_{s_{45}} = \sqrt{\frac{\frac{p_5}{p_4} + \frac{\gamma_E - 1}{\gamma_E + 1}}{\frac{2\gamma_E}{\gamma_E + 1}}}$$

$$\frac{\rho_5}{\rho_4} = \frac{1 + \left(\frac{\gamma_E+1}{\gamma_E-1}\right)\frac{p_5}{p_4}}{\left(\frac{\gamma_E+1}{\gamma_E-1}\right) + \frac{p_5}{p_4}}$$

$$\frac{a_5}{a_4} = \frac{\rho_4}{\rho_5}\left[\frac{M_{s_{45}}}{\sqrt{\dfrac{1+\left(\frac{\gamma_E-1}{2}\right)M_{s_{45}}^2}{\gamma_E M_{s_{45}}^2 - \left(\frac{\gamma_E-1}{2}\right)}}}\right]$$

$$\frac{\Delta U_{45}}{a_4} = M_{s_{45}} - \frac{a_5}{a_4}\sqrt{\frac{1+\left(\frac{\gamma_E-1}{2}\right)M_{s_{45}}^2}{\gamma_E M_{s_{45}}^2 - \left(\frac{\gamma_E-1}{2}\right)}}$$

The last of these equations can be used to compute the dimensionless velocity U_4, i.e.,

$$U_4 = \frac{U_{16}}{a_1} + \left(\frac{\Delta U_{45}}{a_4}\right)a_4$$

If the original estimate for $\frac{P_6}{P_1}$ was correct, this value of should equal the product $a_4 M_4$, where the values of a_4 and M_4 are those computed earlier from the steady flow relations applied across the area discontinuity. Obviously a trial and error process based on guessing P_6/P_1 must be followed until the values of U_4 and $a_4 M_4$ are equal. Once this is achieved, the important dependent variables which can be computed are P_6/P_1, $\frac{U_{16}}{a_1}$, and $M_5 = \frac{U_{16}/a_1}{(a_5/a_4)a_4}$.

Flow into the Second Land

The flow arriving at the end of the first groove is supersonic. It is possible for this flow to make the transition from the groove to the land without undergoing a shock. This flow is analogous to supersonic flow in a converging duct. A shockless transition is possible if a Mach number in region 8 can be found in the range $1.0 \leq M_8 \leq M_5$ which satisfies the equation

$$\frac{A_R M_8}{\left[1+\left(\frac{\gamma_E-1}{2}\right)M_8^2\right]^{\frac{\gamma_E+1}{2(\gamma_E-1)}}} = \frac{M_5}{\left[1+\left(\frac{\gamma_E-1}{2}\right)M_5^2\right]^{\frac{\gamma_E+1}{2(\gamma_E-1)}}}$$

If such an M_8 exists it is found by a trial and error process. Assuming that an acceptable M_8 is found this way, the flow in the second land will contain both a P and a Q shock and a contact surface similar to those formed in the shallow groove. The determination of these two shocks' strengths follows the same procedure as was used to find the shock's strength in the groove. A value of P_{10}/P_1 is estimated, in this case from the equation

$$\frac{p_{10}}{p_1} \approx \frac{\gamma_1(\gamma_1-1)}{2}\left[\frac{a_8 M_8}{1+\frac{a_8}{\sqrt{p_8/p_1}}}\right]^2$$

The Rankine-Huginiot equations are used to determine the velocity change across the P shock as follows

$$M_{S_{110}} = \sqrt{\frac{\frac{p_{10}}{p_1} + \frac{\gamma_1 - 1}{\gamma_1 + 1}}{\frac{2\gamma_1}{\gamma_1 + 1}}}$$

$$\frac{\rho_{10}}{\rho_1} = \frac{1 + \left(\frac{\gamma_1+1}{\gamma_1-1}\right)\left(\frac{p_{10}}{p_1}\right)}{\left(\frac{\gamma_1+1}{\gamma_1-1}\right) + \frac{p_{10}}{p_1}}$$

$$\frac{a_{10}}{a_1} = \frac{\rho_1}{\rho_{10}}\left[\frac{M_{S_{110}}}{\sqrt{\frac{1+\left(\frac{\gamma_1-1}{2}\right)M_{S_{110}}^2}{\gamma_1 M_{S_{110}}^2 - \left(\frac{\gamma_1-1}{2}\right)}}}\right]$$

$$\frac{u_{110}}{a_1} = M_{S_{110}} - \frac{a_6}{a_1}\sqrt{\frac{1+\left(\frac{\gamma_1-1}{2}\right)M_{S_{110}}^2}{\gamma_1 M_{S_{110}}^2 - \left(\frac{\gamma_1-1}{2}\right)}}$$

The pressure ratio across the Q shock can now be calculated as

$$\frac{p_9}{p_8} = \frac{p_{110}/p_1}{p_8/p_1}$$

where

$$\frac{p_8}{p_1} = \frac{p_6}{p_1}\left[\frac{1+\left(\frac{\gamma_E-1}{2}\right)M_5^2}{1+\left(\frac{\gamma_E-1}{2}\right)M_8^2}\right]^{\frac{\gamma_E}{\gamma_E-1}}$$

The Rankine Huginiot equations for the Q shock now yield

$$M_{S_{89}} = \sqrt{\frac{\frac{p_9}{p_8} + \frac{\gamma_E - 1}{\gamma_E + 1}}{\frac{2\gamma_E}{\gamma_E + 1}}}$$

$$\frac{\rho_9}{\rho_8} = \frac{1 + \left(\frac{\gamma_E+1}{\gamma_E-1}\right)\left(\frac{p_9}{p_8}\right)}{\left(\frac{\gamma_E+1}{\gamma_E-1}\right) + \frac{p_9}{p_8}}$$

$$\frac{a_9}{a_8} = \frac{\rho_8}{\rho_9}\left[\frac{M_{S_{89}}}{\sqrt{\frac{1+\left(\frac{\gamma_E-1}{2}\right)M_{S_{89}}^2}{\gamma_E M_{S_{89}}^2 - \left(\frac{\gamma_E-1}{2}\right)}}}\right]$$

$$\frac{\Delta u_{89}}{a_8} = M_{S_{89}} - \frac{a_9}{a_8}\sqrt{\frac{1+\left(\frac{\gamma_E-1}{2}\right)M_{S_{89}}^2}{\gamma_E M_{S_{89}}^2 - \left(\frac{\gamma_E-1}{2}\right)}}$$

From this the dimensionless velocity in region (8) can be computed as

$$\mathcal{U}_8 = \frac{\mathcal{U}_{110}}{a_1} + \frac{\Delta \mathcal{U}_{89}}{a_8} a_8$$

where

$$a_8 = \left(\frac{a_5}{a_4}\right) a_4 \sqrt{\frac{1+\left(\frac{\gamma_E - 1}{2}\right) M_5^2}{1+\left(\frac{\gamma_E - 1}{2}\right) M_8^2}}$$

If this value of $\mathcal{U}_8$ equals the product $a_8 M_8$ then the guessed value of $\frac{p_{110}}{p_1}$ was correct; otherwise a trial and error procedure must be followed until these two quantities are equal. When equality is achieved, the important quantities are $\frac{P_{110}}{P_1}$ and p_{110}/p_1.

It should be noted at this point that if the first groove is found to be "deep", there is no need to compute M_8 as described above since $M_8 = M_3 > 1.0$. In addition $a_8 = a_3$, $P_8/P_1 = P_3/P_1$. The flow in the second land in this case is calculated starting with the estimate for $\frac{P_{110}}{P_1}$ and proceeding with the trial and error process for matching $\mathcal{U}_8$ to $a_8 M_8$.

Calculations performed to-date indicate that it is highly unlikely that an M_8 in the range $1.0 < M_8 \leq M_5$ can be found except for extremely shallow grooves. When this occurs a reflected Q shock forms instead which reduces the flow velocity and increases the gas pressure in a new region (7). In order for this Q shock to propagate upstream into the oncoming combustion products

$$\frac{p_7}{p_5} > \left(\frac{2\gamma_E}{\gamma_E + 1}\right) M_5^2 - \frac{\gamma_E - 1}{\gamma_E + 1}$$

There is another restriction on p_7/p_5 which will be discussed shortly.

Estimating p_7/p_5 using the above equation, the Rankine-Huginiot equations can be used to compute the following quantities for the Q shock

$$M_{5_{57}} = \sqrt{\frac{\frac{p_7}{p_5} + \left(\frac{\gamma_E - 1}{\gamma_E + 1}\right)}{\frac{2\gamma_E}{\gamma_E + 1}}}$$

$$\frac{\rho_7}{\rho_5} = \frac{1 + \left(\frac{\gamma_E + 1}{\gamma_E - 1}\right)\left(\frac{p_7}{p_5}\right)}{\left(\frac{\gamma_E + 1}{\gamma_E - 1}\right) + \frac{p_7}{p_5}}$$

$$\frac{a_7}{a_5} = \frac{\rho_5}{\rho_7}\left[\frac{M_{5_{57}}}{\sqrt{\frac{1+\left(\frac{\gamma_E - 1}{2}\right) M_{5_{57}}^2}{\gamma_E M_{5_{57}}^2 - \left(\frac{\gamma_E - 1}{2}\right)}}}\right]$$

$$\frac{\Delta \mathcal{U}_{57}}{a_5} = M_{5_{57}} - \frac{a_7}{a_5}\sqrt{\frac{1+\left(\frac{\gamma_E - 1}{2}\right) M_{5_{57}}^2}{\gamma_E M_{5_{57}}^2 - \left(\frac{\gamma_E - 1}{2}\right)}}$$

The Mach number in region (7) can now be determined from

$$M_7 = \frac{a_5}{a_7}\left[\frac{\mathcal{U}_{16}}{a_1}\left(\frac{a_4}{a_5}\right)\frac{1}{a_4} - \frac{\Delta \mathcal{U}_{57}}{a_5}\right].$$

The steady flow through the contraction from region (7) to region (8) is determined by finding an $M_7 < M_8 \leq 1.0$ which satisfies the equation

$$\frac{A_R M_8}{\left[1+\left(\frac{\gamma_E-1}{2}\right)M_8^2\right]^{\frac{\gamma_E+1}{2(\gamma_E-1)}}} = \frac{M_7}{\left[1+\left(\frac{\gamma_E-1}{2}\right)M_7^2\right]^{\frac{\gamma_E+1}{2(\gamma_E-1)}}}$$

It is clear that the combination of M_7 and A_R may be such that no acceptable M_8 can be found which satisfies this equation for the value of p_7/p_5 chosen. This will generally be caused by choosing p_7/p_5 so small that $M_8 > 1.0$. This is the second restriction on p_7/p_5 i.e., it must be chosen large enough so that the M_8 which results is less than unity. If $M_8 > 1.0$, the guessed value of p_7/p_5 must be immediately adjusted upward until $M_8 \leq 1.0$ before any further computations are performed.

Once an acceptable M_8 is found, the steady flow isentropic equations for the discontinuity can be used to determine

$$a_8 = \frac{a_7}{a_5}\,\frac{a_5}{a_4}\,a_4\sqrt{\frac{1+\left(\frac{\gamma_E-1}{2}\right)M_7^2}{1+\left(\frac{\gamma_E-1}{2}\right)M_8^2}}$$

$$u_8 = a_8 M_8$$

$$\frac{p_8}{p_1} = \frac{p_7}{p_5}\,\frac{p_6}{p_1}\left[\frac{1+\left(\frac{\gamma_E-1}{2}\right)M_7^2}{1+\left(\frac{\gamma_E-1}{2}\right)M_8^2}\right]^{\frac{\gamma_E}{\gamma_E-1}}$$

This value of p_8/p_1 determines the strength of the P shock transmitted from the groove into the land area. The Rankine-Huginiot equations for this shock are

$$M_{s_{110}} = \sqrt{\frac{\frac{p_8}{p_1} + \frac{\gamma_1-1}{\gamma_1+1}}{\frac{2\gamma_1}{\gamma_1+1}}}$$

$$\frac{\rho_{10}}{\rho_1} = \frac{1+\left(\frac{\gamma_1+1}{\gamma_1-1}\right)\left(\frac{p_{10}}{p_1}\right)}{\left(\frac{\gamma_1+1}{\gamma_1-1}\right)+\frac{p_8}{p_1}}$$

$$\frac{a_{10}}{a_1} = \frac{\rho_1}{\rho_{10}}\left[\frac{M_{s_{110}}}{\sqrt{\frac{1+\left(\frac{\gamma_1-1}{2}\right)M_{s_{110}}^2}{\gamma_1 M_{s_{110}}^2-\left(\frac{\gamma_1-1}{2}\right)}}}\right]$$

$$\frac{u_{110}}{a_1} = M_{s_{110}} - \frac{a_{10}}{a_1}\sqrt{\frac{1+\left(\frac{\gamma_1-1}{2}\right)M_{s_{110}}^2}{\gamma_1 M_{s_{110}}^2-\left(\frac{\gamma_1-1}{2}\right)}}$$

If this value of u_{110}/a_1 equals $a_8 M_8$ computed from the steady flow equations, the guess for p_7/p_5 was correct. In all likelihood these two values will not match so that another guess will be necessary. When $a_8 M_8$ and u_8 are compared during the trial and error process, there are two possible outcomes:

The flow is choked at (e) so that $\mathcal{U}_e = a_e$

Flow in the First Land

The fan of Q characteristics comprising the centered expansion is generated during that portion of the combustion chamber pressure rise which is below the critical pressure ratio for choked flow at the entrance (e). Once $\mathcal{U}_e = a_e$ the slope of the Q characteristics becomes zero. As $\mathcal{U}_e$ and a_e rise in unison with increasing pressure in the combustion chamber the slope of the P characteristics becomes larger and larger. The result is that the P characteristics (which carry the larger and larger values of P) begin to intersect forming the P shock wave which propagates into the stationary gas already in the first land area. When the final pressure is reached in the combustion chamber the flow conditions across the centered expansion are described by the constant value of P_e, where

$$P_e = \left(\frac{\gamma_E + 1}{\gamma_E - 1}\right) a_e$$

The strength of the P shock formed in the first land can be roughly estimated from the formula

$$\frac{p_2}{p_1} \approx \left[\frac{P_e - \frac{4}{\gamma_1 - 1}}{\frac{2}{\gamma_1 (\gamma_1 + 1)}}\right]^2$$

This value of p_2/p_1 can now be used to compute (from the Rankine-Huginiot equations)

1. $a_8 M_8 > \mathcal{U}_8/a_1$ - When this happens increasing p_7/p_5 will reduce $a_8 M_8$ and increase $\mathcal{U}_8/a_1$. In addition M_8 will be reduced. Since M_8 has already been adjusted to be less than unity there is no danger here. Also, increasing p_7/p_5 above the minimum value means that the Q shock continues propagate upstream into the oncoming combustion products.
2. $\mathcal{U}_8/a_1 > a_8 M_8$ - Decreasing p_7/p_5 will reduce $\mathcal{U}_8/a_1$ and increase $a_8 M_8$. There are two problems associated with this trend. One is that p_7/p_5 must never be less than the minimum value given above, or Q shock will be swept downstream. The other problem is that reducing p_7/p_5 increases M_8 toward unity. If it happens that M_8 reaches unity and $\mathcal{U}_8/a_1$ is still larger than $a_8 M_8$, a different calculation procedure for the flow in the groove must be adopted.

Certainly the restrictions imposed on the procedures just described suggest that it is not often that the flow conditions in the groove allow a solution to be found in this way. Order of magnitude calculations confirm this suspicion. When it happens that $M_8 = 1.0$ and $\mathcal{U}_8/a_1$ is still larger than $a_8 M_8$ it is clear that the flow in region (8) cannot be at a constant velocity, but must be accelerating at the expense of a falling pressure. The situation at the entrance to the second land is therefore exactly the same as at the entrance to the first land (e), i.e., choked flow exiting into a centered expansion. The analogy is not complete in the sense that the flow does not accelerate from rest in the groove to $M_8 = 1.0$ at the entrance of the second land. The Q shock, however, does serve the purpose of reducing the velocity in the groove to a very low value, approaching zero.

Computations for the flow in the second groove proceed along the same lines as those used to find p_2 and u_2/a_1 in the first groove. Computations start by evaluating

$$P_8 = \left(\frac{2}{\gamma_E - 1}\right) a_8 + u_8$$

from which the P and Q shock waves and the contact surface can be located using a trial and error process based on guessing the P_{110}/P_1 used in the Rankine-Huginiot equations. It is expected that the majority of shallow groove solutions will be found taking this latter path.

Results

The flow through the first land area is very close to that expected in a shock tube with an open end. If there were no grooves to form a labyrinth the transit time of the combustion products would be calculated from the velocity behind the first P shock (u_2) generated in the first land area.

Preliminary estimates based on typical combustion pressures and temperatures show that this velocity is of the order 10,000 in./sec., so that the transit time for the combustion products through a smooth annulus is of the order 10^{-4} sec per inch of piston length. Piston closing times are of the order $2.5(10^{-4})$ to $75\ (10^{-4})$ seconds. As a consequence, in order for the labyrinth to be effective it must reduce the transit time by a factor of 10 to 100.

If the labyrinth does reduce the speed of the combustion products it will do so in steps as the gas passes from one groove to another across the lands. Assuming that the total transit time is to be reduced only by a factor of 10, the velocity behind the shock wave must drop rapidly to well below sonic velocity (relative to the undisturbed gas ahead of it) after the first few grooves. If this does not happen, the high early velocities cannot be compensated for over the total length of the labyrinth. For this reason the computer solution need not be carried beyond the second land if the velocity behind the shock wave in the second land is not considerably less than the comparable velocity in the first land.

The results of the computer calculations are shown in the accompanying table. As the area ratio (A_R) is decreased going from a shallow groove toward a deep groove configuration, the velocity in the first groove u_{16}/a_1 decreases as might be expected. This decrease continues until A_R is approximately 0.20 when the groove becomes so deep that the flow from the first land does not fill the groove. For area ratios smaller than 0.30 it is assumed that the carry-over velocity in the first groove remains constant as the flow crosses the groove without a shock.

As the area ratio A_R is decreased the velocity u_{110}/a_1

in the second land increases and is always higher than the velocity in the first land. The reason for this is that stronger reflected Q shocks are required to pass the combustion products into the second land as A_R decreases. These shocks result in higher pressure levels in the second land which in turn cause stronger shocks and the higher velocities in the second land. The velocity in the second land changes from about one and one-half times the velocity in the first land for very shallow grooves to almost twice the first land velocity when A_R is approximately 0.30.

The calculations show that the second land velocity increases as A_R decreases and then abruptly decreases to the same velocity as in the first land at $A_R = 0.20$. This is, of course, due to the assumption that the deep groove carry-over flow is shockless. Obviously no such sharp transition is expected in the real labyrinth when the flow changes from quasi one-dimensional to full fledged two-dimensional flow as A_R is decreased. The results do show, however, that if the groove clearance is not more than about four times the land clearance there is at best only a small reduction of velocity in the groove which is more than compensated for by the increased velocity in the next land. If the next groove-land combination were calculated, the computed velocities would be larger than in the first groove-land combination. It will be recalled that just the opposite trend is necessary for a successful design, i.e., the velocity must decrease drastically.

Having ruled out the possibility of using the shallow groove design because it shows no regions of substantial velocity reduction we next turn our attention to the deep groove design, $A_R < 0.3$. Unfortunately, the analysis of this design is not based on the same from foundation as the shallow groove analysis. The reason for this is quite simple, the two-dimensional flow which occurs under these circumstances is extremely complex, requiring a degree of analytical sophistication beyond the scope of this investigation.

It will be recalled that the analysis for the deep groove is based on the assumption of complete "carry-over". This is most certainly never achieved in practice. There are two important aspects of the deep groove flow which must be considered. One is concerned with how quickly any part of the contaminated flow (combustion products) reaches the exit leading from the groove into the next land. The other is concerned with how the flow in the groove affects the flow in the second land area.

With regard to the first question, there is little doubt that the high velocity with which the gas jets from the first land vigorously propels the combustion products across the groove space, placing them very quickly near the entrance to the second land. In that sense the approximation that the combustion products are carried-over in a jet-like fashion seems justifiable and a reasonable approximation.

The second question, dealing with the way in which the groove flow influences the flow in the second land, cannot be dealt with in such a clear-cut fashion. The analysis assumes that the flow from the deep groove behaves in very much the same way as the flow from the combustion chamber into the first land, i.e., there is no initial velocity and the flow is choked at the entrance to the land. Assuming that contaminants have traversed the groove length carried by a jet-like flow, it is only necessary for a large pressure pulse to reach the entrance to the second land in order to initiate contaminant flow in the second land area. Recalling that the shock wave preceeds the contact surface, it is correct to say that the pressure pulse sent into the groove by the first land reaches the groove exit before the contaminants do. This being the case the important time increment is the one associated with the contaminant carry-over via jetting, rather than the pressurization lag. It is for this reason that the carry-over Mach number M_5 and a_5/a_1 are computed and used to characterize the contaminant transit time across the groove.

The analysis also assumes that the presumed pulse is not weakened by two-dimensionality effects while traversing the groove's length. Herein lies the analysis greatest weakness, i.e., that the combustion chamber's temperature and pressure are assumed to be transmitted isentropically and undiminished to the next groove. For modestly deep grooves ($A_R > 0.3$) where the two-dimensional effects are not too severe this could be an acceptable approximation. For really deep grooves ($A_R < 0.1$) such is not the case.

It is possible, however, draw some conclusions in lieu of of a precise analysis. It was pointed-out earlier that the labyrinth would be a successful replacement for sealing rings only if it resulted in high early reductions in velocity of the order of one to two orders of magnitude. This means at least comparable order of magnitude reduction in pressure must be achieved in the grooves. It is hard to conceive of shock wave strengths (pressure ratios) weakening to that extent while traversing a groove. Every land in which the contact surface moves with supersonic velocity (due to high pressures in the grooves) requires additional lands downstream with subsonic velocity to compensate for it. From this point of view, it seems unlikely that the weakening of the pressure pulses in the groove can substantially reduce land velocities unless there are a very large number of deep grooves.

One additional point in favor of deep grooves is their ability to dilute the combustion products with uncontaminated gases. On a strictly statistical basis the larger the grooves and the larger the number of grooves, the smaller the chance that many contaminated particles will reach the end of the labyrinth. This aspect of labyrinth leakage control has not been considered at all in the analyses described but should be a consideration if the gases traverse very many grooves.

Summary

The basic problem investigated in this analysis was the unsteady axial flow of a compressible fluid through a form of labyrinth seal applied to the actuating cylinder of an explosively operated valve. The purpose of the analysis was to determine the transit time through the labyrinth of the leading edge of the combustion products. If this time is less than the closing time of the valve a labyrinth type seal could be used in place of the sealing rings now employed, with no degradation of the

valve's performance.

The analysis itself was based on the equations of unsteady flow gas dynamics, omitting heat transfer and frictional effects and assuming homentropic flow in regions where there are no shocks. These regions are matched together at the shocks using the Rankine-Huginiot shock equations. The flow across area discontinuities were analyzed using the isentropic equations of steady flow.

Since the motion of the leading edge of the contaminating combustion products through the labyrinth is necessarily an early time phenomena, only the initial history of the leakage need be examined analytically. This means that the complicated wave interactions which evolve at later times do not enter into the analysis.

For shallow grooves with area ratios (A_R) greater than about 0.3 the formation of shocks within the grooves generated larger land velocities downstream and no substantial reduction in velocities in the grooves. Since the velocity in the first area land is well in excess of that required for successful seal performance, labyrinth designs with area ratios in this range are not likely candidates.

On the basis of the analysis it was found that the land and groove velocities were unchanged proceeding downstream, remaining at a level well in excess of that required for successful operation. Although the deep groove analysis is highly approximate, on an order of magnitude basis it offers little hope of successful operation with deep grooves either.

One must conclude, therefore, that labyrinth seals have little likelihood of preventing contaminant leakage during the valve closing time.

Future studies in this area should concentrate on the deep groove design and particularly on the details of the flow in the deep groove. It is only here that substantial reductions in axial velocity may eventually be gained. Because of the complicated nature of the deep groove flow and the potentially beneficial contributions caused by dilution which are also difficult to analyze, there is much to recommend a modest experimental program before further costly analysis is performed. The results of such could well determine whether further analysis is really necessary.

NOMENCLATURE

a	= speed of sound
a_o	= speed of sound at the reference temperature
a_1	= speed of sound of the stagnant gas in the labyrinth
$\mathcal{a}$	= a/a_o
A	= area normal to the flow
A_c	= critical area ratio = land area/A_j
A_j	= expanded jet area
A_R	= land to groove area ratio
c	= labyrinth clearance measured at the land
c_P	= constant pressure specific heat of combustion products
c_v	= constant volume specific heat of combustion products.
$D(M)$	= $\dfrac{M}{\left[1+\left(\frac{\gamma-1}{2}\right)M^2\right]^{\frac{\gamma+1}{2(\gamma-1)}}}$
$L_1, L_2, \ldots, L_n$	= axial lengths of the land and grooves
L_t	= basic tuning length of the groove
M	= Mach number = $\mathcal{U}/\mathcal{a}$
M_s	= Mach number of the shock wave
p	= absolute pressure
p_e	= absolute pressure at the entrance to the first land
p_E	= absolute pressure after combustion in the combustion chamber
p_o	= absolute pressure at the reference temperature, T_o
P	= $\left(\frac{2}{\gamma-1}\right)\mathcal{a} + \mathcal{U}$
P_e	= $\left(\frac{2}{\gamma-1}\right)\mathcal{a}_e + \mathcal{U}_e$
Q	= $\left(\frac{2}{\gamma-1}\right)\mathcal{a} - \mathcal{U}$
R	= gas constant
S_E	= entropy of the combustion products in the combustion chamber following combustion
S_o	= initial entropy of the combustion products at P_o and T_o
t	= time
T	= absolute temperature
T_E	= absolute temperature of the combustion products in the combustion chamber following combustion
T_o	= initial absolute temperature of the gas in the labyrinth, reference temperature
$\mathcal{U}$	= dimensionless velocity = gas velocity/$\mathcal{a}_o$
$\Delta\theta$	= Mach angle for the expanding jet
α	= turning angle of the expanding jet boundary
γ_1	= specific heat ratio of the uncontaminated gas in the labyrinth
γ_E	= specific heat ratio of the combustion products
ξ	= dimensionless axial coordinate
ρ	= gas density $\equiv x/L_n$
τ	= dimensionless time coordinate $\equiv \dfrac{a_o t}{L_n}$

Appendix

Explantion of Computer Program Print-Out

Because the computer program language does not admit to subscript notation the symbols used in the text and in the program do not always correspond exactly. The following paragraphs will provide symbolic transliterations, as well as some explanatory comments on the computed output.

Since the flow through the first land is always computed the same way the print-out for it always follows the same format. The following symbolic equivalents are used.

P2/P1 $\equiv p_2/p_1$

MS12 $\equiv M_{s12}$

DU12/A1 $\equiv \mathcal{U}_{12}/a_1$

DS12 $\equiv$ specific entropy change across the P shock

The entropy change across the shocks was never considered in the text of the report. Itis included in the computer program as a check on the program itself since the entropy change across a shock must always be non-negative. The entropy rise across the shocks is also a measure of the major dissipative effects in the flow field, and in that sense the magnitude of the entropy change is an indicator the labyrinth's restrictiveness.

If the first groove is "deep" the combustion products are assumed to traverse the groove length without shock. The important parameters computed here are:

M6 $\equiv M_6$

P6/P1 $\equiv p_6/p_1$

A6/A1 $\equiv a_6/a_1$

DU16/A1 $\equiv \mathcal{U}_{16}/a_1$

The entropy change here is zero.

When the groove is shallow, shocks form in the flow field. The print-outs for this case are:

P6/P1 $\equiv p_6/p_1$

MS16 $\equiv M_{s16}$

DU16/A1 $\equiv \mathcal{U}_{16}/a_1$

DS16 $\equiv$ specific entropy change across the P shock

DS45 $\equiv$ specific entropy change across the Q shock

Once again the entropy changes must be positive. The computer print-out signals a "deep" or "shallow" groove calculation by the omission or inclusion of entropy changes.

There are three different ways in which the flow in the second land can form. The flow from a "deep" groove always forms both P and Q shocks. Occassionally the flow from a shallow groove follows the same path. The print-outs for these cases are:

P110/P1 $\equiv p_{110}/p_1$

DU110/A1 $\equiv \mathcal{U}_{110}/a_1$

MS110 $\equiv M_{S110}$

DS110 $\equiv$ specific entropy change across the P shock

DS89 ≡ specific entropy change across the Q shock

The two remaining paths start with the formation of a reflected Q shock in the groove. The distinguishing feature of these flows is that they do not result in a Q shock in the land area, and hence the print-out of DS89 is zero.

If the flow at the entrance to region (8) is subsonic the flow is steady and a P shock forms. If the flow at the entrance to region is sonic then the flow in region (8) is unsteady but it also forms a P shock in the land. The Mach number at the entrance to the land area (M_8) is printed so that these two cases can be distinguished, one from the other. The print-out for these two cases are:

P110/P1 ≡ p_{110}/p_1

MS110 ≡ M_{S110}

DU110/A1 ≡ u_{110}/a_1

M8 ≡ M_8

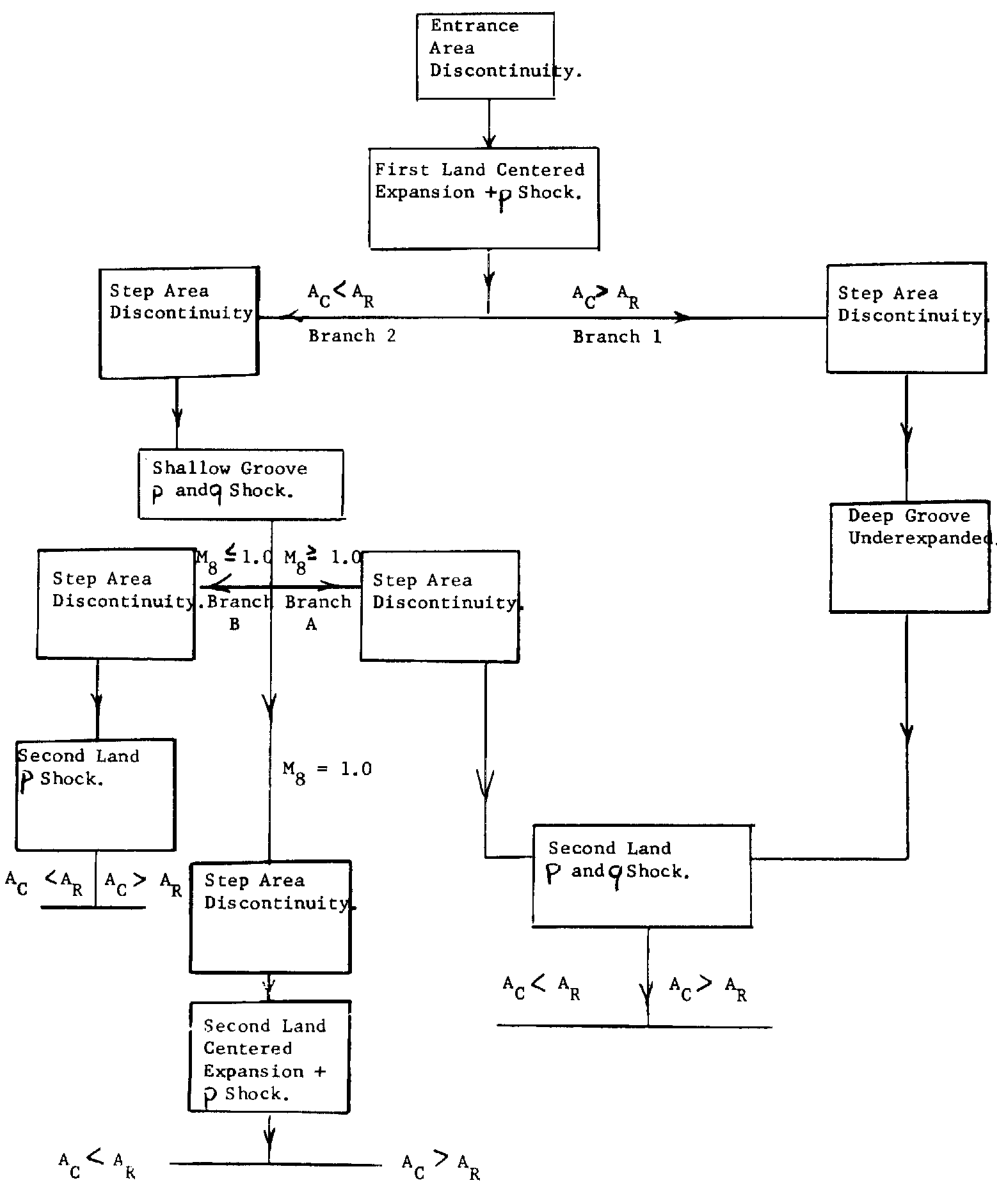
DS110 ≡ specific entropy change across the P shock.

If the recipe described above is followed it is not necessary to probe the inner workings of the computer program in order to determine the nature of flow in the various sections of the labyrinth. Each section of the print-out has been given its tell-tale trade-mark which immediately identifies the path followed by the program.

TABLE OF COMPUTATIONAL RESULTS

AR	p_2/p_1	$M_{S_{12}}$	U_{12}/a	$p_6/p_1 = p_5/p_1$	$M_{S_{16}}$	$U_{16}/a = U_{15}/a$	p_{110}/p_1	$M_{S_{110}}$	U_{110}/a_1	M_8
0.9	113.75	9.88	8.15	111.70	9.79	8.08	309.33	16.29	13.52	1.00
0.8	113.75	9.88	8.15	108.97	9.67	7.97	335.96	16.97	14.10	1.00
0.7	113.75	9.88	8.15	106.26	9.55	7.87	356.82	17.49	14.53	1.00
0.6	113.75	9.88	8.15	103.67	9.43	7.77	376.19	17.96	14.92	1.00
0.5	113.75	9.88	8.15	101.21	9.32	7.68	395.62	18.42	15.30	1.00
0.4	113.75	9.88	8.15	96.51	9.10	7.49	404.68	18.63	15.48	1.00
0.3	113.75	9.88	8.15	91.97	8.89	7.31	412.34	18.80	15.63	1.00
0.2	113.75	9.88	8.15	40.03	-	8.42	113.75	9.88	8.15	4.51
0.1	113.75	9.88	8.15	40.03	-	8.42	113.75	9.88	8.15	4.51

FLOW CHART



Computer Program Output

```
NWLAB3     16:14     ST MON 06/16/69

IN NSHOCK
IN .FIRST

INPUT VARIABLES

      TE          TO      GAMMA 1    GAMMA E
  4460.000    520.000      1.400      1.333

  AR =           .10

      P2/P1         MS12      DU12/A1        DS12
     113.751        9.881      8.150         4.099

     P5/P1     DU15/A1
    40.033      8.422

    P110/P1   DU110/A1     MS110       DS110        DS89         M8
    113.751     8.150      9.881       6.288        .000        4.513

  AR =           .20

      P2/P1         MS12      DU12/A1        DS12
     113.751        9.881      8.150         4.099

     P5/P1     DU15/A1
    40.033      8.422

    P110/P1   DU110/A1     MS110       DS110        DS89         M8
    113.751     8.150      9.881       6.288        .000        4.513

  AR =           .30

      P2/P1         MS12      DU12/A1        DS12
     113.751        9.881      8.150         4.099

     P6/P1      MS16      DU16/A1       DS16        DS45
    91.973      8.887      7.312        3.819       .112

     P110/P1      MS110      DU110/A1         DS110              M8
     412.344      18.804      15.625          6.309
                                                              1.000

  AR =           .40

      P2/P1         MS12      DU12/A1        DS12
     113.751        9.881      8.150         4.099

     P6/P1      MS16      DU16/A1       DS16        DS45
    96.509      9.103      7.494        3.898       .089

     P110/P1      MS110      DU110/A1         DS110              M8
     404.679      18.628      15.479          6.276           1.000
```

Output, continued

```
AR =          .50

   P2/P1        MS12      DU12/A1        DS12
  113.751       9.881       8.150        4.099

  P6/P1      MS16   DU16/A1      DS16       DS45
 101.214     9.322    7.679      3.976      .076

  P110/P1       MS110     DU110/A1       DS110          M8
  395.619      18.419      15.304        6.237       1.000

AR =          .60

   P2/P1        MS12      DU12/A1        DS12
  113.751       9.881       8.150        4.099

  P6/P1      MS16   DU16/A1      DS16       DS45
 103.670     9.434    7.773      4.016      .073

  P110/P1       MS110     DU110/A1       DS110          M8
  376.187      17.961      14.921        6.148       1.000

AR =          .70

   P2/P1        MS12      DU12/A1        DS12
  113.751       9.881       8.150        4.099

  P6/P1      MS16   DU16/A1      DS16       DS45
 106.256     9.551    7.872      4.057      .071

  P110/P1       MS110     DU110/A1       DS110          M8
  356.817      17.492      14.529        6.056       1.000

AR =          .80

   P2/P1        MS12      DU12/A1        DS12
  113.751       9.881       8.150        4.099

  P6/P1      MS16   DU16/A1      DS16       DS45
 108.967     9.672    7.974      4.098      .071

  P110/P1       MS110     DU110/A1       DS110          M8
  335.955      16.974      14.096        5.951       1.000

AR =          .90

   P2/P1        MS12      DU12/A1        DS12
  113.751       9.881       8.150        4.099

  P6/P1      MS16   DU16/A1      DS16       DS45
 111.699     9.792    8.075      4.140      .071

  P110/P1       MS110     DU110/A1       DS110          M8
  309.330      16.288      13.522        5.807       1.000
```

COMPUTER PROGRAM LISTING

```
NWLAB3     16:50    ST MON 06/16/69

10 COMMON GAM1,PE,M,DEL,A,UOA
15 REAL M,M2,M3,M4,M5,M7,M8,M10
17 PRINT,:"INPUT VARIABLES"
18 PRINT,:"      '  TE          TO    GAMMA 1    GAMMA E"
20 READ,TE,TO,GAM1,GAME
21 PRINT 355,TE,TO,GAM1,GAME
23 DO 9999 AR=.1,.9,.1
24 PRINT,::
25 TIM=TIME=J=0;XINC=2.
26 PRINT,:"  AR =",AR
30 GMIN1=GAME-1.
35 AE=SQRT(2./(GAME+1.)*TE/TO)
40 PE=(GAME+1.)/GMIN1*AE
42 42 CONTINUE
45 P2OP1=(PE-4./(GAM1-1.))**2/(2./(GAM1**2+GAM1))
50 25 CALL NSHOCK(P2OP1)
52 XM12=M
55 A3=(PE-UOA)/(2./GMIN1)
60 P3OP1=A3**(2.*GAME/GMIN1)
65 CK=P3OP1-P2OP1
70 SI=SIGN(1.,CK)
75 IF(ABS(CK)-.5)50,50,32
80 32 IF(TIM)33,36,33
85 33 IF(SI-SIGN(1.,SAVE))34,35,34
90 34 XINC=XINC/2.
91 P2OP1=P2OP1+SI*XINC
95 SAVE=CK;GO TO 25
100 35 P2OP1=P2OP1+SI*XINC
105 SAVE=CK;GOTO 25
110 36 P2OP1=P2OP1+SI*XINC
115 SAVE=CK
120 TIM=1.;GO TO 25
150 50 S2=(1./(GAM1*(GAM1-1.)))*LOG(P2OP1*((1.+(GAM1-1.)/2.*XM12
155 +**2)/((GAM1+1.)/2.*XM12**2))**GAM1)
157 IF(TIME)158,158,732
158 158 CONTINUE
160 PRINT,:"          P2/P1          MS12       DU12/A1          DS12"
165 PRINT 51,P2OP1,XM12,UOA,S2
170 51 FORMAT(5F12.3)
172 IF(TIME)175,175,9999
175 175 M2=UOA/A
180 M3=UOA/A3
185 ARG=(1.-(1./P2OP1))*SQRT(M3**2-1.)/(GAME*M3**2)
190 AC=1.-SQRT(M3**2-1.)*(SIN(ARG)/COS(ARG))
195 IF(AC-AR)300,300,200
200C   BRANCH 1
205 200 M5=M3
210 210 XL=M5/(1.+GMIN1/2.*M5**2)**((GAME+1.)/(2.*GMIN1))
215 XR=AC*M3/(1.+GMIN1/2.*M3**2)**((GAME+1.)/(2.*GMIN1))
220 IF(ABS(XL-XR)-.005)225,225,220
225 220 M5=M5+.1;GO TO 210
230 225 A5OA3=SQRT((1.+GMIN1/2.*M3**2)/(1.+GMIN1/2.*M5**2))
231 P5OP1=P2OP1*((1.+GMIN1/2.*M3**2)/(1.+GMIN1/2.*M5**2))**(GAME/GMIN1)
```

Listing, continued

```
U5=M5*A50A3*A3
M8=M3;U8=U5;A8=A3;P80P1=P30P1
PRINT,'"     P5/P1     DU15/A1"
PRINT 355,P50P1,U5
GO TO 391
C    BRANCH  2
300 M4=M3
XR=AR*M3/(1.+GMIN1/2.*M3**2)**((GAME+1.)/(2.*GMIN1))
310 XL=M4/(1.+GMIN1/2.*M4**2)**((GAME+1.)/(2.*GMIN1))
IF(XL-XR-.005)325,325,320
320 M4=M4+.1;GO TO 310
325 ARG=(1.+GMIN1/2.*M3**2)/(1.+GMIN1/2.*M4**2)
P40P1=P30P1*ARG**(GAME/GMIN1)
A4=A3*SQRT(ARG)
P60P1=GAM1*(GAM1+1.)/2.*(A4*M4/(1.+A4/SQRT(P40P1)))**2
CK4=M4*A4
330 CALL NSHOCK(P60P1)
U16=U0A
XM16=M
P50P4=P60P1/P40P1
CALL NSHOCK(P50P4)
A504=A
U4=U16+U0A*A4
IF(ABS(U4-CK4)-.1)350,350,335
335 IF(U4-CK4)340,350,345
340 P60P1=P60P1+.1;GO TO 330
345 P60P1=P60P1-.1;GO TO 330
350 PRINT,'"      P6/P1        MS16     DU16/A1        DS16         DS45"
M5=U16/(A*A4)
S16=(1./(GAM1*(GAM1-1.)))*LOG(P60P1*((1.+(GAM1-1.)/2.*XM16
+ **2)/((GAME+1.)/2.*XM16**2))**GAM1)
S45=(1./(GAM1*(GAM1-1.)))*LOG(P50P4*((1.+(GAM1-1.)/2.*M**2)/((
+ GAME+1.)/2.*M**2))**GAM1)
PRINT 355,P60P1,XM16,U16,S16,S45
355 FORMAT(6F10.3)
EX=(GAME+1.)/(2.*GMIN1)
DM5=M5/(1.+GMIN1/2.*M5**2)**(EX)
M8=1.
DM5AR=DM5/AR
358 XL=M8/(1.+GMIN1/2.*M8**2)**(EX)
IF(XL-DM5AR)370,360,360
PRINT,XL,DM5AR
360 IF(XL-DM5AR+.05)380,380,365
365 M8=M8+.1
IF(M8-M5)358,358,450
370 GO TO 450
C  BRANCH  A
380 A8=A*A4*SQRT((1.+GMIN1/2.*M5**2)/(1.+GMIN1/2.*M8**2))
P80P1=P60P1*((1.+GMIN1/2.*M5**2)/(1.+GMIN1/2.*M8**2))**(GAME/GMIN1)
391 CONTINUE
P100P1=P20P1
CK8=A8*M8
382 CALL NSHOCK(P100P1)
U110=U0A
M10=M
P90P8=P100P1/P80P1
CALL NSHOCK(P90P8)
U8=U110+U0A*A8
```

Listing, continued

```
IF(ABS(U8-CK8)-.03)400,400,385
385 IF(U8-CK8)390,390,395
390 P100P1=P100P1+.1;GO TO 382
395 P100P1=P100P1-.1;GO TO 382
400 S110=(1./(GAM1*(GAM1-1.)))*LOG(P100P1*((1.+(GAM1-1.)/2.*M10**2
+ )/((GAM+1.)/2.*M10**2))**GAM1)
S89=1./(GAM1*(GAM1-1.))*LOG(P90P8*((1.+(GAM1-1.)/2.*M**2)/((GAM1
++1.)/2.*M**2))**GAM1)
PRINT,'"     P110/P1   DU110/A1      MS110      DS110       DS89
+        M8"
PRINT 410,P100P1,U110,M10,S110,S89,M8
410 FORMAT(6F10.3)
GO TO 9999
C     BRANCH    B
450 P70P5=((2.*GAME/(GAME+1.))*M5**2)-GMIN1/(GAME+1.)
455 CALL NSHOCK(P70P5)
M7=(1./A)*(U16/(A504*A4)-U0A)
M8=M7
456 IF(M8-1.)457,4560,4560
4560 IF(J)487,487,490
457 XL=M8/((1.+GMIN1/2.*M8**2)**((GAME+1.)/(2.*GMIN1)))
458 XR=1./AR*(M7/(1.+GMIN1/2.*M7**2)**((GAME+1.)/(2.*GMIN1)))
XL1=1./((1.+GMIN1/2.)**((GAME+1.)/(2.+GMIN1)))
IF(XL1-XR)460,460,470
460 P70P5=P70P5+.1;GO TO 455
470 IF(ABS(XL-XR)-.02)495,495,475
475 IF(XL-XR)480,480,485
480 M8=M8+.05;GO TO 456
485 M8=M8-.05;GO TO 456
487 P70P5=P70P5+1.;GO TO 455
C BRANCH  C
490 M8=1.
A8=A*A504*A4*SQRT(1.+GMIN1/2.*M7**2/(1.+GMIN1/2.*M8**2))
PE=(GAME+1.)/GMIN1*A8
TIME=1.;GO TO 42
495 ARG=1.+GMIN1/2.*M7**2/(1.+GMIN1/2.*M8**2)
A8=A*A504*A4*SQRT(ARG)
U8=A8*M8
P80P1=P70P5*P60P1*ARG**(GAME/GMIN1)
CALL NSHOCK(P80P1)
IF(ABS(U8-U0A)-.25)525,525,500
500 IF(U8-U0A)505,525,510
505 P70P5=P70P5-.1;J=1;GO TO 455
510 P70P5=P70P5+.1;GO TO 455
525 S110=1./(GAME*GMIN1)*LOG(P80P1*((1.*(GAM1-1.)/2.*M**2/((
+ GAM1+1.)/2.*M**2))**GAM1))
PRINT,'"    P110/P1      MS110   DU110/A1       DS110          M8"
PRINT 550,P80P1,M,U0A,S110,M8
550 FORMAT(5F10.3)
732 PRINT,'"       P110/P1         MS110      DU110/A1          DS110
+           M8"
PRINT 51,P20P1,XM12,U0A,S2,M8
9999 CONTINUE
$USE NSHOCK
$DATA
4460,520,1.4,1.333,.6
```

CHAPTER III

FACE SEAL THEORY

HYDRODYNAMIC EQUATIONS FOR SEAL DESIGN

Introduction

In order to accurately analyze the hydrodynamic performance of face seals it is first necessary to establish what approximations may legitimately be made, and based on these approximations solve the appropriate equations with realistic boundary conditions. The literature contains a wealth of information on face seals both analytical and experimental. Much of this material is not pertinent to an investigation of the hydrodynamic behavior of seals for one reason or another. In this report the only sources cited are those which deal directly with the problem at hand and are of noteworthy merit.

Early analytical investigations of seal hydrodynamics started with the presumption that the well-developed theories of hydrodynamic bearing theory could be "carried-over" unaltered to seal applications. It was soon realized that this "lubrication-theory" was inadequate except in a few special cases. Surprisingly enough, the current literature still occasionally contains publications which proport to present a "generalized" seal analysis based on the Reynolds equation. Recognizing the inadequacy of "bearing theory" hydrodynamicists have recently developed analytical techniques which represent a substantial improvement in predictive accuracy over the older discredited methods. The references cited in the discussion to follow are concerned with these more recent developments.

Discussion of Available Data and Information on face seals

For non-rotating parallel face-seals the creeping motion solution, verified by Bremner for small flow Reynolds numbers, has been extended by Savage to include first order inertia effects at higher Reynolds numbers. The improved analysis agrees with the experimental data cited. An interesting feature of the more comprehensive analysis is its ability to "see", at least in part, the "entrance-effect" totally neglected by the over-simplified lubrication theory. It is also interesting to note that the entrance-effect has recently become of concern to lubrication technologists after having been neglected for so long. Savage's analysis does not completely account for the entrance effect, but it is certainly the first step in this direction.

When one of the disks is rotated, the flow situation becomes considerably more complicated. Daily and Nece have shown experimentally that even in the case of no net radial flow there are at least four possible combinations of flow regimes depending on the dimensions of the clearance space and the rotational Reynolds number.

Bayly and Conway attempted an investigation in which a radial pressure gradient and radial flow were imposed in addition to rotation. They were interested in larger clearances than are generally encountered in seals, so that the flow fields they encountered were quite complicated and probably similar to those investigated by Daily and Nece and others.

Dowson considered a more seal-like configuration, i.e., small clearances. He investigated the effects of radial and tangential flow when only the centrifugal force contribution to the acceleration is present. His results, restricted to large Euler numbers, are confirmed by comparison with experiment within the range of his approximation.

The effect of changes in radial geometry have been investigated analytically by Snapp using the Dowson solution for the pressure distribution. Since no experimental comparison is presented, the question of how far Dowson's solution can be stretched to analyze other geometries remains an open question. So long as the geometric deviations from parallel disks is not too severe the analysis is probably accurate for large Euler numbers.

The work of these investigators clearly indicates the necessity of including at least some non-linear inertia effects in any so-called general face seal analysis, whether the disks are stationary or rotating. Reasonably good analytical methods which include inertial effects are available for handling

disk or near-disk like configurations in laminar flow provided certain restrictions are satisfied. Actually, the limit of these restrictions has yet to be established experimentally. It is quite possible, for example, that as the Euler number is decreased more complicated flow fields will occur which are not covered by analyses, such as those found by Bayley and Conway. A more thorough experimental-analytical comparison than has been done to date is required to clear-up this point.

The problem of turbulent flow between parallel disks, because of its complexity, has been almost totally neglected to date. Daily and Nece, as well as others, have found that under the proper conditions turbulent flow could be established in the clearance space. The problem is somewhat more complicated than the laminar case, but several analytical schemes which have been successful in other configurations might be tried here. Once again it would be important to support the analysis with an experimental investigation.

So far, the discussion has focused on the preliminary considerations of the hydrodynamic problem. Establishing these concepts provides a "jumping-off-place" for improvements and refinements. Variance between basic theory and experiment are most generally attributed to the following causes:

1. Initially uncontrollable geometric variations, such as surface waviness and misalignment, in contradistinction to the intentional geometric variations already mentioned.
2. Temperature and pressure (in the case of compressible fluids) induced variations in the fluid properties.
3. Operationally induced geometric variations, such as thermal distortion and elasto-hydrodynamic distortions caused by the pressure distribution.
4. Entrance effects.

To a certain extent some of these problems have been investigated, but usually under the assumption that the techniques of lubrication theory are applicable (Osterle and Navandi, Orcutt and Cheng).

In summary, the basic features of face-seal hydrodynamics have been established. Analysis for laminar flow are available, but their limitations are not yet established experimentally. The analysis of turbulent flow is not available. The important possible variations from the idealized theory have been identified and await further analytical and experimental investigation.

<u>Non-Rotating Case-Laminar Flow</u>

<u>Ref</u>: Savage, S. B. "Laminar Radial Flow Between Parallel Plates" Jr. Applied Mech, Vol. 31, No. 4, Dec, '64, PP 594-596.

The analysis carried-out here is concerned with the purely radial, laminar out-flow of an incompressible fluid between two parallel disks. The solution obtained by a perturbation technique is an improvement over the "creeping-flow" solution of hydrodynamic lubrication theory in that it attempts to account for the inertial effects neglected in the creeping solution.

A comparison of the analytical results with very limited experimental evidence indicates excellent agreement with the measured radial pressure distribution except near the entrance. Here the analysis is in closer agreement with experiment than the creeping solution, but still somewhat in error. This error is attributed by the author to the influence of entrance effects.

Unfortunately there is no indication, either, analytical or experimental, as to the maximum Reynolds Number for which the derived approximate expression is expected to hold. From experimental comparison shown, it appears that the results are reasonably accurate up to at least Reynolds numbers of the order 3 x 10^4.

It should be emphasized that many seals have a very narrow radial land width so that the inability of this analysis to accurately predict entrance effects may severly limit its usefulness on an actual seal design.

The formula for the radial pressure distribution is as follows:

$$P - P_\infty = \frac{3\mu Q}{4\pi h^3} \ln \frac{R}{r} - \frac{27}{560} \frac{\rho Q^2}{h^2} \left[\frac{1}{r^2} - \frac{1}{R^2} \right]$$

where

P = pressure at radius r

P_∞ = pressure at radius R

R = outside redius of disks

r = ralial coordinate

μ = fluid visiosity

h = one-half the clearance

Q = volumetric flow rate

ρ = fluid density

Ref: Bremner, G. F. : "Experimental Study of Balance Conditions, Leakage Rates, and Pressure Profiles for Stationary Radial-Face Seals", Paper F2, Second International Conference on Fluid Sealing, BHRA, April 1964.

The results of the experiments reported in this paper indicate that the pressure distribution and flow-rate are adequately predicted by the equation:

$$P - P_\infty = \frac{3\mu Q}{4\pi h^3} \ln \frac{R}{r}$$

where

$$R_e^* = \frac{\rho \bar{U} h}{\mu} \frac{h}{R} \ll 1$$

Several different materials were used for the seal faces and it was found that the materials with a lower modulus of elasticity tended to distort more causing deviations between experiment and prediction. Temperature changes in viscosity were also found to influence correlation. The range of Reynolds numbers tested were $5 \times 10^{-5} < R_e^* < 5 \times 10^{-4}$, but no upper limiting R_e^* was determined for which this simple equation ceases to be valid.

<u>Rotating Case - No Net Radial Flow</u>

Ref: Daily, J. W. & Nece, R. E. "Chamber Dimension Effects on Induced Flow & Functional Resistance of Enclosed Disks." Jr. ot Basic Engrg., Vol. 82 pp 217 - 232 March 1960.

This investigation is primarily an extension of the work begun by Shultz-Grunow on the frictional drag of rotating enclosed disks. Those results are not directly related to through-flow seals, since the disk considered rotates in a completely enclosed housing and the measurements concerned with torque rather than flow and pressure. On the other hand, the investigators did find that four different flow regimes could be obtained (laminar flow with merged and separated boundary layers and turbulent flow with merged and separated boundary layers), each with its own torque characteristic. These results serve to illustrate the complex flows which are obtainable. Of particular interest is the presence of a boundary layer which would also be expected in narrow face seals operating at high rotational speeds.

These results are of basic importance in the design of seals whose geometry is significantly altered by shrouding.

<u>Rotating Case-Radial Pressure Gradient and Flow</u>

Ref: Bayley, F. J., Conway, L: Fluid Friction and Leakage Between a Stationary and Rotating Disk, Jr. of Mech. Engrg. Sci. Vol 6, No. 2, 1964, PP 164 - 172.

This investigation is an attempt to determine the combined effect of tangential and radial in-flow between rotating disks. The experimental results are conducted using dimensional analysis but no formal analysis is presented. The clearances used are also somewhat larger than normally used in seals.

It should be noted that the "creeping-flow" solutions for rotating disks do not predict any effect of rotation on the radial flow. The experimental results obtained here clearly demonstrate that a rotational effect is present when the tangential Reynolds number exceeds 10^5, where

$$R_e = \frac{\rho \omega R^2}{\mu}$$

ω = Angular Velocity of Disks

While the results of this purely experimental investigation are strictly applicable only to the geometric configurations tested and are therefore of limited value, the general conclusion is that this problem is by no means simple and that more thorough investigation is certainly required.

Re: Dowson, D.: "Inertia Effects in Hydrostatic Thrust Bearings"; Jr. of Basic Engrg., Vol. 83, No. 2, PP 227 - 234, June 1961.

Dowson, D.: "Discussion of Papers in F Session", Proceedings of the Second International Conference on Fluid Sealing, BHRA 1964, PP F81 - F83.

This paper includes retains the most predominate inertial terms in the analysis of a parallel disk face seal. This is a decided departure from the usual hydrodynamic lubrication theory which neglects these effects altogether. The influence of centripetal accelerations on the pressure distribution is considerable, indicating that solutions obtained using hydrodynamic lubrication theory are restricted to very low rotational speeds.

The results of the analysis indicate that the radial pressure distribution at the surface of the disks is geven by

$$P = \frac{3}{20}\rho r^2 \omega^2 + C \ln r + D$$

where C and D are constants. Applying this equation to a pocketed circular thrust bearing, the dimensionless flow-rate

$$\frac{6\mu Q}{\pi P_o h^3} = -\alpha^3 \left[\frac{1 + \frac{3}{20}\frac{\rho R^2 \omega^2}{P_o}(1 - r_o'^2)}{(\alpha^3 - 1)\ln r_1' + \ln r_o'} \right]$$

where

P_o = Supply pressure at the supply tube

R = outside radius of seal

$r_o' = r_o/R$

$r_1' = r_1/R$

α = ratio of pocket clearance to land clearance

r_o = Radius of fluid supply note

r_1 = radius of pocket

The assumptions made in this derivation require that the inertial parameter

$$\frac{\rho R^2 \omega^2}{P_o} \leq 1.0$$

Dowson cited successful experimental confirmation of his analysis with α = 1.0 (no pocket) up to values of the inertia parameter = 4.0 (BHRA Ref.)

A somewhat poorer correlation was obtained with deeper pockets, the discrepency being attributed by the author to 24°F rise in the fluid temperature.

Ref: Snapp, R. B., "Theoretical Analysis of Face-Type Seals with Varying Radial Face Profiles," ASME Preprint 64 WA/Lub6.

The analysis given here is actually an extension of the work by Dowson to non-parallel surfaces. While no experimental verification is given, the results are probably accurate if the geometry is a slight deviation from parallelism and the inertial parameter is of order unity or less.

Rotating Case - No Radial Flow - Misaligned

Ref: Osterle, F., Nahvandi, A., "The Effect of Vibration on the Load Carrying Capacity of Parallel Surface Thrust Bearings," 60 LubS-3, ASME Preprint 1960.

This paper presents an approximate analysis of the effect of the misalignment of the axes of rotation of two disks. It is assumed that the inertial effects are negligible and that there is no applied radial pressure difference. It is also assumed that the regions of subambient pressure cavitate and do not contribute to the load carrying capacity. No experimental comparison is given.

The equation for the pressure distribution within the clearance space is given by

$$P = \frac{3n\rho_m \sin\theta}{2(1 - n\rho_m \cos\theta)^3}\left[\rho^2 - (1+k)\rho + k\right]$$

where

θ = polar angle measured from the minimum film thickness

$\rho_m = \frac{1+k}{2}$

$k = r_i/r_o$

r_i = inside radius of disks

r_o = outside radius of disks

$$n = \frac{r_0 \lambda}{h_0}$$

h_0 = film thickness at radius = 0

λ = angle of axes inclination

Ω = absolute angular velocity of slider rotation

ω = absolute angular velocity of wobble

The dimensionless load capacity is given by

$$\bar{W} = \frac{n \rho_m (1-2k+2k^3-k^4)}{4(1-n^2 \rho_m^2)}$$

where

$$\bar{W} = \frac{w}{2\mu(\Omega-2\omega)\frac{r_0^4}{h_0^2}}$$

w = load capacity

Rotating Case - Hydrodynamic and Thermal Distortion Effects

Ref: Orcutt, F. K. and Cheng, H. S., "Preliminary Analytical Investigation of Radial Face Seals" U. S. Navy Marine Engrg. Lab. Rpt. 141/64.

In this paper hydrodynamic lubrication theory is used to analyze the effects of elastic and thermal distortion of face seals. The seal dynamic response characteristics were also studied analytically. A partial experimental comparison of the analytical results was made. Since the analysis neglects inertial effects altogether, it is probably limited to small rotational speeds.

Discussion of Available Data and Information on Shaft Seals

Fortunately, the hydrodynamics of shaft seals is considerably more advanced than the hydrodynamics of face seals. There are probably two reasons for this situation. First, the problem is somewhat simpler analytically and experimentally. Secondly, annular flow is closely related to journal bearings, so that its understanding has been of more immediate concern.

Probably the most useful papers, from the design standpoint, are the papers by Bell and Bergelin, and by Yamada. The first paper by Bell and Bergelin summarizes in one graph the results of many experimentors through the use of an orifice-type correlation coefficient which is accurate to $\pm 15\%$, according to the authors. They also furnish semi-empirical formulae which apply to various regions of the graph, but add little to the basic understanding of hydrodynamics. All of the results presented in their paper are for the non-rotating shaft case. In a practical sense, however, those parts of the plot which are in the laminar regime (Taylor numbers less than 40) are probably accurate even if the shaft rotates (see Yamada).

Some insight into the physical behavior of the fluid is given in a series of papers by Sparrow, one of which is cited here. For laminar flow in the entrance of an annulus, the fully developed pressure drop is increased by additional viscous forces and inertial forces present in the region of developing flow. The results of his analysis are in substantial agreement with those of Schlichting. Unfortunately, the pressure drop associated with the flow approach to the duct mouth is of the same order of magnitude as the one encountered within the duct and hence cannot be neglected. This pressure drop can only be estimated on the basis of simple theories for the individual viscous and inertial contributions. With this assumed correction added in, the analysis is in fair agreement for laminar flow when the ratio of length to clearance is greater than unity.

The literature contains almost nothing on the turbulent effects on approach or within the annulus, only a suggestion that the entrance length is probably longer then that in laminar flow. Brighton and Jones have investigated the fully established turbulent flow pressure drop in smooth annuli experimentally and semi-empirically using the Blasius form of the friction factor. Once again an approach correction must be added in order to compute the total pressure drop. Correlation with Bell and Bergelin is good for length to clearance ratios greater than ten, if the previously mentioned assumed approach correction is added in. For aspect ratios less than ten, entrance effects within the duct are probably not negligible - a point which requires further fundamental investigation.

The second outstanding investigation mentioned earlier, is that undertaken by Yamada. Using the method of "coupled orthogonal flows" Yamada derives an approximate analytical expression for the combined axial and circumferential flow between long coaxial rotating cylinders. With only slight modification this formulation is verified by experiment over a wide range of operating conditions depending on the axial and circumferential Reynolds numbers. The presence of tangential and axial flow coupling makes it doubtful that the non-rotating turbulent data of Bell and Bergelin can be used for the turbulent rotating annulus.

So far only concentric annuli have been discussed. In laminar flow, if the shaft and seal centerlines are eccentric but parallel, the change in flow rate is easily calculated from lubrication theory. Tao and Donovan have generalized this simple correction for turbulent flow showing that the leakage rate is less drastically altered by changes in eccentricity in turbulent flow than in laminar flow.

No information is available on the effects of a general misalignment. Misalignment torques are of interest to bearing technologists and this led Rice and Ausman to study the problem assuming that there was no through-flow in the annulus. These results could serve as a guide to the designer in ascertaining the self-alignment capabilities of flexurally mounted shaft seals.

It might be important to mention at this point that under certain circumstances the flow regime within a rotating annulus changes from simple Couette flow to toroidial vortices wrapped around the rotating shaft. Their existance was predicted and demonstrated experimentally by G. I. Taylor earlier in this century, and these vortices now bear his name. Since their discovery they have come under intensive analytical and experimental investigation. It is now known that at least four flow regimes are possible; laminar and turbulent Couette flow, and laminar and turbulent Taylor vortices.

Even though much effort has been invested toward understanding these vortices, much remains to be investigated, especially the effect of eccentricity and applied axial gradients. The presence of these vortices could very well be the reason why Yamada's analysis failed to correlate with experiment for certain ranges of rotational Reynolds numbers. DiPrima has made an exploratory investigation of the effects of eccentricity and axial pressure gradient, concluding that both effects tend to suppress the occurrence of the Taylor vortices. It should be stressed that these are only preliminary conclusions which are subject to experimental confirmation. Since most of the experiments with Taylor vortices has been with "long" cylinders, it is not certain they even exist in seal-like configurations.

Along these lines, Bjorklund and Kays experimentally investigated the frictional behavior of concentric rotating cylinders over a wide range of Reynolds numbers. They were able to distinguish at least two flow transitions and correlated semi-empirical friction factor formulae with the experimental data. There is some evidence that their friction factors are conservative if axial flow is superimposed on the axial flow.

In summary:

1. Design data is available for non-rotating, concentric shaft seals over a wide range of Reynolds numbers and aspect ratios. Analyses based on fundamental hydrodynamic principles are available for certain regions covered by the design data. A thorough

analytical understanding of approach effects and turbulent entrance effects is still lacking. In shaft seals, where these effects may very well predominate, this could be a serious analytical deficiency.

2. Design data is available for rotating, long, concentric shaft seals over a wide range of axial and circumferential Reynolds numbers and correlated in part with theory. For "short" seals, i.e., low aspect ratios, caution should be exercised in the use of this data since the neglected entrance effects may very well cause large discrepancies. This is a field of investigation which requires considerably more attention.
3. Techniques are available which permit the designer to estimate the effect of shaft eccentricity on the flow.
4. Numerical values of critical Reynolds and Taylor numbers are available which allow the designer to identify the probable flow regime present within the annulus.
5. Design information is also available which permits the designer to calculate the rotation torque and power consumption in a shaft seal.
6. The moments associated with shaft-seal misalignment can be roughly estimated.

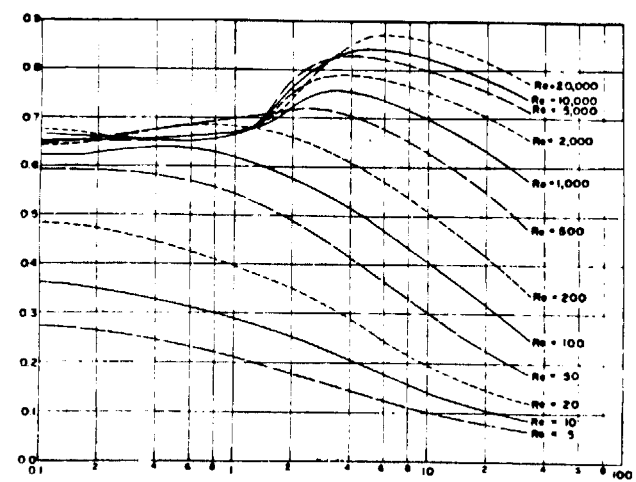
Non-Rotating Case - Laminar Flow and Turbulent Flow

Ref: Bell, K. J.; Bergelin, O. P.: "Flow Through, Annular Orifices" ASME Trans., Vol. 79, 1957, pg. 593.

This is probably the most comprehensive experimental investigation of annular, seal-like, restrictions available. The experimental results are correlated using three dimensionless parameters:

SUMMARY OF CONCENTRIC ORIFICE COEFFICIENTS

C
Orifice
Coefficient



L/c = annulus length/annulus radial clearance

After Reference:

Bell, K. J.; Bergelin, O. P.: "Flow Through Annular Orifices"

Length to clearance ratio = L/c

Reynolds Number = $R_e = \frac{D_e G}{\mu}$

Orifice Coefficient = $C = \frac{W}{A\sqrt{2\, g_o \rho\, \Delta P_T}}$

Where

L = annulus length

c = annulus radial clearance

$G = \rho V_{avg}$

ρ = fluid density

V_{avg} = average axial velocity of fluid

De = equivalent diameter = 2c

μ = fluid viscosity

W = axial mass flow rate of fluid

ΔP_T = total pressure drop across annulus

The various flow regimes are correlated with semi-empirical formulas. The curves plotted represent the mean values of the experimental data with a spread of $\pm$ 15%.

A particularly interesting feature of the graphical plot is the general concurrance of the data for $L/c < 1.0$ and $R_e > 100$.

Ref: Sparrow, E. M.; Lin. S. H.: "The Developing Laminar Flow and Pressure Drop in the Entrance Region of Annular Ducts", Journal of Basic Engineering, Vol. 86, No. 4, December, 1965, pp 827-834.

Schlichting, H: Boundary Layer Theory, McGraw-Hill, pp 171

This paper presents an analysis of the laminar flow entrance effects from the mouth of the annulus to any downstream point in the flow. The pressure variatio for seal-like configurations is given by the expression:

$$P_o - P = \left[\frac{96}{R_e}\frac{L}{D_e} + K\right] \frac{\rho V_{avg}^2}{2g_o}$$

The first term on the right hand side accounts for the fully developed viscous pressure drop. The second term is a function of the dimensionless ratio

$$\frac{L/c}{Gc/\mu}$$

and accounts for the inertial and viscous effects near the mouth of the annulus.

When this parameter is greater then 0.03 and the clearance is small, K is approximately 0.6. The value of K for other conditions may be determined from a graph given in the paper. This value of K is in close agreement with the result given by Schlichting.

In order to apply this equation to a shaft seal, the pressure drop between the upstream reservoir and the annulus mouth must be taken into account. This pressure drop is also composed of inertial and viscous contributions. The inertial contribution is one velocity-head, while the viscous contribution, predicted by the simplified sudden contraction analysis, is one-half of a velocity head. Adding these corrections to the formulation, yields

$$\Delta P_T = \left[\frac{96}{R_e}\frac{L}{D_e} + K + 1.5\right] \frac{\rho V_{avg}^2}{2\, g_o}$$

When L/c is greater than 100, the duct is long enough to mask the added corrective terms. For $1 < L/c < 10$ the equation is in reasonable agreement with the curves of Bell and Bergelin if $R_e > 100$. If L/c is of the order 0.1, the comparison is poor, indicating that the approach viscous head loss is not constant at one-half a velocity head.

Probably the most valuable feature of the analysis is its ability to predict the entrance length required to reach fully established laminar flow.

Non-Rotating Case - Turbulent Flow

Ref: Brighton, J. A.; Jones, J. B.: "Fully Developed Turbulent Flow in Annuli", Journal of Basic Engineering, Vol. 86, No. 4, December 1965, pp. 835 - 844.

Where:

$\Delta P/L$ = pressure drop gradient between two points in fully established flow regime

c = radial clearance

V_{avg} = average axial velocity

ρ = fluid density

Two Reynolds numbers are introduced, one for the axial flow given by

$$R_o = \frac{\rho c\, V_{avg}}{\mu}$$

and one for the circumferential flow

$$R_\omega = \frac{\rho c\, R\omega}{\mu}$$

Where:

R = radius of the shaft

ω = angular velocity of shaft

μ = viscosity

Without rotation, experimental data verifies that the friction factor is accurately predicted by

$$\lambda = \frac{48}{R_o} \qquad \text{laminar, } R_e < 10^3$$

$$\lambda = \frac{0.26}{R_o^{0.24}} \qquad \text{turbulent, } R_e > 10^3$$

Using an equivalent diameter Reynolds number, R_e, the laminar friction factor becomes

$$f = \frac{96}{R_e}$$

consistant with the references cited above. Converting the turbulent friction factor to match the previous definition

$$f = \frac{0.0766}{R_e^{0.24}}$$

which is in fair agreement with the results of Jones and Brighton.

Assuming a power-law velocity profile in the tangential direction the authors developed the following expression for the friction factors with rotation

$$\lambda = \frac{0.26}{R_o^{0.24}} \left[1 + \left(\frac{7}{8}\right)^2 \times \left(\frac{R_\omega}{2\, R_o}\right)^2 \right]^{0.38}$$

This expression was substantiated experimentally under the following conditions

$$R_w > 10^4, \; 250 < R_o < 25000$$
$$2(10^3) < R_\omega < 3(10^4), \; R_o \geq 5(10^3)$$

Graphs of the experimental results are available for those conditions not covered by the formulas.

The experimental results show that the sharp distinction between laminar and turbulent flow becomes less evident as R_ω increases from zero. For $R_o < 5(10^3)$ in the range $10^3 < R_\omega < 10^4$, transition to Taylor vortices causes large deviations between the experiment and the proposed sample theory.

<u>Rotating Case Laminar and Turbulent - Eccentric</u>

<u>Ref</u>: Donovan, W. F.; Tao, L. N.: "Through-Flow in Concentric and Eccentric Annuli of Fine Clearance With and Without Relative Motion of Boundaries"; <u>ASME</u> Trans., Vol. 77, 1955, pp 1291.

For the case of laminar flow between eccentric cylinders, the authors recall the well-known relationship

$$\frac{Q_e}{Q} = 1 + \frac{3}{2}\,\epsilon^2$$

Where:

Q_e = eccentric axial flow rate

Q = concentric axial flow rate

ϵ = ratio of center-line eccentricity to clearance

When the flow is turbulent they derive the approximate expression

$$\frac{Q_e}{Q} = \frac{1}{\pi} \int_0^{\pi} (1 + \epsilon \cos \theta)^{\frac{3}{2-n}} d\theta$$

Where n is the Reynolds number exponent in the Blasius friction factor formula ($0.21 < n < 0.25$). When $n = 1.0$, this equation reduces to the laminar prediction. Inspection of this equation indicates that the ratio Q_e/Q will decrease as $n \to 0$, a trend confirmed, at least qualititavely by the authors from experimental tests.

Rotating Case Misalignment Torques

Ref: Rice, J. R.: "Misalignment Torques of Hydrodynamic Gas Lubricated Bearings", Journal of Basic Engineering, Vol. 87, Series D, No. 1, March, 1965, pp. 193.

A numerical analysis is used to evaluate the misalignment torque in gas lubricated journal bearings when the midpoint of the shaft has a zero eccentricity. These results are compared with the earlier simplified results of Ausman.

The results, presented in the form of design curves, can be used for incompressible fluids by replacing the ambient pressure ($P_{amb.}$) with the liquid isothermal bulk modulus.

Rotating Case Frictional Effects

Ref: Bjorklund, I. S.; Kays, W. M.: "Heat Transfer Between Concentric Rotating Cylinders" Journal of Heat Transfer, Vol. 81, Series C, No. 3, August 1959, pp. 175.

The authors of this article are primarily interested in establishing an analogy between hear transfer and the frictional character of the circumferential flow between concentric cylinders. The results of their frictional investigation can

be summarized as follows:

Laminar $N_{Ta} < 40$

$$(fN_{Ta})_{lam.} = \frac{4(c/R)^{3/2}}{1 - \left(\frac{1}{1 + c/R}\right)^2}$$

Where:

c = radial clearance

R = shaft radius

N_{Ta} = Taylor number $= \sqrt{\frac{c}{R}} \left(\frac{\rho R\omega c}{\mu}\right)$

T = shear stress on rotating cylinder

f = friction factor $= \dfrac{T}{\rho \frac{(R\omega)^2}{2 g_o}}$

First Transition $40 < N_{Ta} < 90$

$$\frac{fN_{Ta}}{(fN_{Ta})_{lam.}} = 0.0388 \, (N_{Ta})^{0.877}$$

Second Transition $90 < N_{Ta} > 10{,}000$

$$\frac{fN_{Ta}}{(fN_{Ta})_{lam.}} = 0.19 \, (N_{Ta})^{0.522}$$

It should be emphasized that these results are for the case of no axial flow. Experiments indicate that the ratio $fN_{Ta}/(fN_{Ta})_{lam}$ is increased by the presence of the axial flow and the ratios given here are the minimum values.

Rotating Case - Stability - No Axial Flow

Ref: DiPrima, R. C.: "A Note on the Stability of Flow in Loaded Journal Bearings": ASLE Trans., Vol. 6, No. 3, July 1963 p. 249.

DiPrima, R. C.: "Viscous Flow Between Rotating Concentric Cylinders With a Circumferential Pressure Gradient at Speeds Above Critical": ASLE Trans., Vol. 7, No. 4, Oct. 1964, pp. 333.

The critical values of the Reynolds and Taylor numbers are compared for typical bearing clearance to radius ratios. For concentric cylinders with no axial flow, it is concluded that the critical Reynolds number for transition to Taylor vortices or turbulent flow is roughly the same, approximately 1400. It is also concluded that the effect of eccentricity or axial pressure gradient is in the direction of stabilizing flow, and that axial flow tends to retard the establishment of Taylor vortices.

In the second paper it is concluded that the results for concentric cylinders with no circumferential pressure gradient represent a lower bound for stability.

SECTION III B

LEAK DETECTION AND LEAKAGE MEASUREMENTS

Introduction

An integral part of any seal design is the detail consideration which must be given to the methods and techniques which will be employed to evaluate the integrity of the seal under simulated operational environments. Establishing the integrity of a seal involves the location of any leakage path which may be present plus the determination of the magnitude of flow involved. To accomplish this in an effective manner, a detailed understanding of the various flow modes which may be encountered and the various measurement methods which can be employed is very important. A knowledge of leakage calibration standards as well as an appreciation of the various leakage conversion and correlation techniques is also very helpful in defining the effectiveness of a seal. Considerable information and data are contained in reference (1), (3) and (4) on the general subject of leak detection and leakage measurement. The review included here is intended to give a few of the guiding principles taken from those reports and to indicate in rather broad terms the methods and techniques which are available for establishing a quantitative measure of seal effectiveness.

Modes of Flow in Gas Systems

The flow mode associated with a gas system may be free-molecular, laminar, turbulent, or it may lie in the vaguely defined slip flow and transition flow regimes. The important dimensionless parameter which characterizes the flow regime is the Knudsen Number (K_n) which is the ratio of the molecular mean free path (λ) to the clearance (δ) or hole size. For leakage detection work the Knudsen Number is not an entirely meaningful criteria since a direct measurement of the hole size is usually not feasible.

The flow of gases in leakage measurement work is most readily analyzed through the concept of conductance. Conductance has the units of pressure multiplied by volume divided by time multiplied by pressure or volume per time. The equation is as follows

$$C = \frac{Q}{\Delta P}$$

Where C = Conductance = $\frac{\text{volume}}{\text{sec}}$

Q = Flow rate = $\frac{\text{(volume) (pressure)}}{\text{sec}}$

ΔP = Pressure drop = pressure

It should be noted that C has the same units as S, the pump speed characteristically associated with vacuum pump systems widely used in leakage measurement analysis.

For the analysis of liquid flow, it is necessary to consider only two broad regimes, namely, laminar and turbulent. In the absence of two phase flow the classical Poiseuille and turbulent flow relationships are sufficient to define the flow of liquids.

Equations for analyzing the flow of gases and liquids in the various flow modes are treated rather completely in the references (1), (3), and (4) also presented are experimental correlations of the various flow theories and flow modes. The equations covering the permeation flow of gases is contained in section 11.0 of this guide.

Leakage Detection Equipment and Techniques

Leak detection equipment, like any other equipment design for critical measurements, should be used only within its design range. Unless this is done, one is apt to obtain meaningless results or to incur needlessly high costs. Reference (4) contains a complete description of instrumentation for leak detection and measurement. The following brief summary is taken in large measure from the same source. For detailed information regarding any particular method, the reference source should be consulted. It should be noted that the useful range of leakage measurement equipment extends through ten orders of magnitude. Consequently, a knowledge of the limitations of each method is important.

Leakage Measurement Methods

(Except Radioactive)

Type	Useful Minimum Range atm cc/sec
Gas Meter	10
Sonic	10^{-1}
Fluorescent dyes	Unknown
Bubble testing by Soap Solution	10^{-4}
Bubble testing by Immersion	10^{-5}
Halide Torch	10^{-5}
Pressure Change	(see discussion below)
Argon "getter" system	10^{-9}
Hydrogen sensitive ionization gage	10^{-8}
Electron emission (oxygen sensitive)	10^{-8}
Thermionic halogen	10^{-6} (10^{-9} is possible)
Mass spectrometer (helium sensitive)	10^{-10}
Condensation Nuclei	10^{-7}

An additional class of leak detection and leakage measurement devices is based on the use of nuclear techniques. Five broad areas of nuclear leak detection include: (a) Intrinsic radiotracers (b) Activation analysis (c) Radioactive gas system (d) Radioactive gage ionization systems and (e) Radioactive gas technology. Experimental sensitivities in the order of 10^{-4} cc/sec using radioactive gas techniques have been indicated for space-vehicle leak checking. A complete treatment of this subject matter is contained in reference (1).

Calibration of Leakage Testing Equipment

The inaccuracy of leak detectors is due primarily to three basic causes, namely:

1. calibration
2. non-linearity of detecting instrument
3. environmental variations

For calibration of test equipment it is generally convenient to employ commercially available calibrated leaks of the capillary or permeation type. The permeation type are available in the range 10^{-6} atm cc/sec to 10^{-10} atm cc/sec and most often have a self-contained gas supply. The capillary leaks are made from constricted glass tubing or collapsed metal tubing. These calibrated leaks can be made down to approximately 10^{-7} atm cc/sec. It is important that calibrated leaks obtained commercially should be checked against some absolute standard, if possible. There is some indication that considerable variation can be found in commercially available calibrated leaks. Table III on page 38 of reference 2 gives the results of experimental work performed on commercially available calibrated leaks.

Non-linearities present in the electronic apparatus must be understood and fully recognized. Consultation with manufacturers of test equipment will usually be sufficient to overcome problems in this area.

Environmental variations, for example, temperature, noise background and contaminates must be constantly monitored and fully compensated in order to obtain meaningful results.

Liquid-Gas Leakage Correlations

Correlations between the leakage of an operating liquid and leakage of some convenient test gas have been recognized as an important need for some considerable time. Preliminary work by Santeler and Moller (Reference 11) produced convenient graphs for establishing these correlations on an analytical basis. However, preliminary experimental evaluations have only recently been made. Reference 3 contains the results of these correlations. On a preliminary basis the work of

Santeler and Moller can be considered accurate to within 20%.

Leak Testing Problems

It is frequently observed that if a device schedule for leakage testing has been in contact with a liquid, no leak smaller than 10^{-7} atm cc/sec will be detected even with the use of a mass spectrometer. The effect is due to the liquid surface tension which in the case of water (72 dynes/cm) can sustain a pressure differential of several atmospheres under certain conditions. If the testing is done at one atmosphere, no leakage will be detected.

The check valve leak in which a sharp repeatable discontinuity is present in the leakage-pressure differential relationship has been noted but is not fully understood.

The self-cleaning leaks have been encountered in several test programs. This phenomena suggests that whenever possible, leak detection should be done at the proposed operating pressure.

GAS PERMEATION THROUGH SOLIDS

The permeation of a gas through a sound solid membrane has been described by various investigations as an adsorption - diffusion - desorption process. However, the actual mechanism is not clearly understood and perhaps varies for different gas-solid combinations. The following general statements can be made for metals and polymers.

1. No rare gas passes through any metal.
2. Halogen gas show no marked tendency.
3. H_2 permeates most metals especially Fe.
4. O_2 permeates Ag.
5. All gases permeate all polymers.

The Permeation Equations

The molecular permeation of a gas through a solid is described by the equation:

$$Q = PA \frac{\Delta P}{L} \qquad (1)$$

Where

Q = volume flow rate at STP

A = area normal to flow

ΔP = pressure drop along flow path

L = length of flow path

P = permeation rate

If the diffusion is atomic in nature, as with diatomic gases permeating through metals, the volume rate of flow is given by the following relationship:

$$Q \quad (P_1 - P_2) \qquad (2)$$

It should be noted that the permeation rate P is exponentially temperatu dependent as given by the following:

$$P = P_o e^{-\frac{E}{JRT}} \qquad (3)$$

P_o = constant, independent of temperature variation

E = activation energy of permeation

J = dissociation number (2 for diatomic gas permeating metals)

$R = 1.986\ cal/K^o$

T = degrees Kelvin

Sample Calculation

In order to estimate the flow losses attributable to permeation, consider the following example:

A gasket 0.1" thick 0.1" wide and 5" nominal diameter with a 1 atmosphere pressure difference across it.

$A = 5(2.54)\ \pi\ (0.1)\ (2.54) = 10.13\ cm^2$

$L = 0.254\ cm = 2.54\ mm$

$\Delta P = 1\ atm.$

Equation 1 becomes

$$Q = PA\frac{\Delta P}{L} = \frac{10.13}{2.54} P = 3.99\ P$$

For a natural rubber gasket at 25^oC

	P	Q	ρ	Loss
	$\frac{cm^3 - mm}{sec\text{-}cm^2 - atm}$	$\frac{cm^3}{sec}$	$\frac{lbs.}{cm^3}$	$\frac{lbs.}{year}$
H_2	4.0 (10^{-6})	15.96 (10^{-6})	1.83 (10^{-7})	92.0 (10^{-6})
O_2	1.8 (10^{-6})	7.18 (10^{-6})	29.2 (10^{-7})	661.0 (10^{-6})
N_2	1.0 (10^{-6})	3.99 (10^{-6})	25.5 (10^{-7})	321.0 (10^{-6})
He	3.0 (10^{-6})	11.97 (10^{-6})	3.65 (10^{-7})	137.7 (10^{-6})

For a Neoprene gasket at 24° C

	P $\frac{cm^3 - mm}{sec\text{-}cm^2 - atm}$	Q $\frac{cm^3}{sec}$	ρ $\frac{lbs.}{cm^3}$	Loss $\frac{lbs.}{year}$
H_2	10 (10^{-6})	3.99 (10^{-6})	1.83 (10^{-7})	2.30 (10^{-6})
O_2	0.3 (10^{-6})	1.2 (10^{-6})	29.2 (10^{-7})	110.0 (10^{-6})
N_2	0.1 (10^{-6})	0.399 (10^{-6})	25.5 (10^{-7})	32.1 (10^{-6})
He	0.6 (10^{-6})	2.4 (10^{-6})	3.65 (10^{-7})	27.6 (10^{-6})

By way of comparison, a 9-10 foot long steel pipe of the same diameter and thickness would lose Hydrogen at a rate comparable to those given above.

Summary

The calculation of the leakage rate, as given by equation (1), is based on a knowledge of the experimental determined permeation rate P. The data for polymers can usually be obtained from the literature and extrapolated to different temperatures by equation 3. For metals the diffusion process is atomic in nature and although the permeation rate is still exponential dependent on temperature, the diffusion flow is proportional to the difference of the square root of the pressures on each side of the metal solid. It is important that consistent dimensions be maintained in the use of permeation rates obtained from the literature.

Paper No.
67—WA/Lub-20

Cavitation in Mechanical Face Seals

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The characteristics of mechanical face seals operating with hydrodynamic films containing gas cavities were studied. Solutions to Reynolds' equation for hydrodynamic lubrication were obtained using both the short bearing approximation and numerical iterative methods. These two methods of solution satisfy continuity of flow throughout the seal film. Leakage rates and seal hydrodynamic loads were calculated. Some computed cavity shapes are also given.

Introduction

Hydrodynamic effects have been shown to exist in mechanical face seals [1].[1] Seal forces and leak rates are primary considerations in the design and application of face seals. Therefore, it is important to understand the basic principles of face seal operation. (See Fig. 1.)

The exact cause of hydrodynamic effects in mechanical face seals is not known. Four suggestions have been made by various investigators. They are:

1 Lubricant cavitation and nonparallel seal surfaces.
2 Axial seal vibration coupled with variable lubricant properties.
3 Variable lubricant properties and nonparallel surfaces.
4 Inertia effects.

Theoretically, each of these causes has been shown to be significant in producing net forces in face seals greater than the hydrostatic forces. However, inertia effects would not explain hydrodynamic forces of the magnitude measured by Denny [1]. There is a good possibility that variable fluid properties are a cause. However, experiments to measure this effect are difficult to control and accurate measurements are difficult to make. Another good prospect is lubricant cavitation. This has been observed by Nau [2] and Battelle [3] in transparent face seal tests.

Cavitation has been shown, by approximate analysis, to yield some of the same effects as those measured by Denny [1]. Good analytical solutions to the cavitation problem have not been obtained except for the journal bearing work of Jakobsson and Floberg [4]. This problem has been pursued in this paper so that cavitation can be studied in face seals from an analytical point of view.

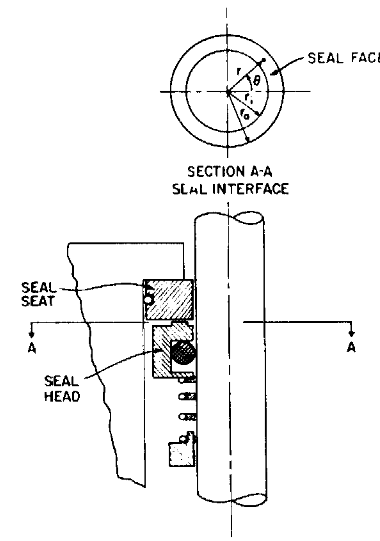

Fig. 1 Mechanical face seal geometry

Hydrodynamic forces are generated in face seals by lubricant cavitation because the cavitation prevents the generation of hydrodynamic pressures below a certain level while not restricting the upper bound of hydrodynamic pressure generation. If cavitation did not occur, the components of hydrodynamic

[1] Numbers in brackets designate References at end of paper.

Contributed by the Lubrication Division for presentation at the Winter Annual Meeting, Pittsburgh, Pa., November 12-17, 1967, of The American Society of Mechanical Engineers. Manuscript received at ASME Headquarters, August 17, 1967. Paper No. 67—WA/Lub-20.

Nomenclature

A = seal area, sq in.
A_f = flow area, sq in.
h = film thickness, in.
$\bar{h}_i$ = average film thickness at inside radius, in.
ρ = density, lb/cu in.
r = coordinate measured from center of seal, in.
θ = angular coordinate, rad
U = fluid velocity, in/sec
ω = rotational speed, rad/sec
Q = flow rate, cu in/sec
p = pressure, psi
τ = shear stress, psi
W = seal load capacity, lb_f
SM = shaft friction moment, in-lb_f
μ = dynamic viscosity, psi-sec

Dimensionless Terms

R = radius ratio = r/r_o
H = film thickness ratio = $h/\bar{h}_i$
P = pressure ratio = p/p_o
Λ = seal number = $\frac{6\mu\omega}{p_o}\left(\frac{r_o}{\bar{h}_i}\right)^2$
ϵ = waviness parameter
C_L = load coefficient = $\frac{W}{p_o r_o^2}$
C_f = friction coefficient = $\frac{SM}{\bar{h}_i{}^3 p_o r_o{}^3/6}$
q = flow coefficient = $\frac{12\mu Q}{\bar{h}_i{}^3 p_o}$
U' = pumping velocity variable = $\frac{U}{r_o\omega}$
n = number of circumferential waves
η = boundary function, equation (22)

Subscripts

0 = at the outside radius of the seal
i = at the inside radius of the seal
C = within the cavitated film

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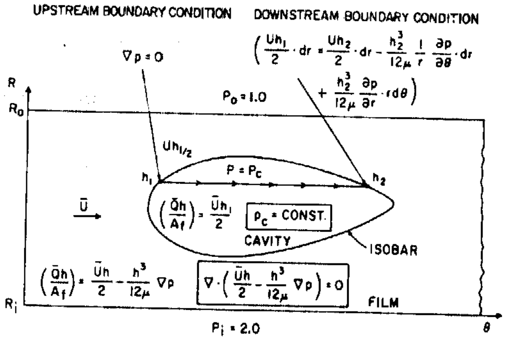

Fig. 2 Cavitation boundary conditions

pressure generation would balance. No net increase over the hydrostatic force would exist.

In lubricants, cavitation generally occurs when air or other gases dissolved in the lubricant separate from solution. The pressure at which this occurs is the pressure at which the lubricant is saturated with the particular gas. When the sealing fluid is other than an oil or the system is completely enclosed, the conditions that must exist for cavitation to occur are questionable. This would have to be investigated for the particular application.

The purpose of this study is to obtain solutions for the cavitating and noncavitating cases. Based on these solutions basic seal parameters may be studied. Solutions are not available which handle the cavitating condition accurately enough to conserve fluid flow rate in the seal. One of the basic seal parameters is leakage rate. It is important to be able to calculate it accurately along with calculations of net seal force and friction.

Cavitation Boundary Conditions [4]

Formulation of the boundary conditions, to be applied to Reynolds' equation for a lubricant film containing cavities, essentially involves a multiple region solution with flow rates matched at the region boundaries.

Within the cavity, the pressure is constant and the lubricant is carried in "streaks" or "ribbons" by the moving surface. Thus, within the cavity, Reynolds' equation reduces to

$$P_c = \text{const} \tag{1}$$

and the lubricant flow rate is given by

$$\frac{\bar{Q}}{A_f}h = \frac{\bar{U}}{2}h_1 \tag{2}$$

where h_1 = film thickness at upstream edge of the cavity.

Within the continuous film region, Reynolds' equation for incompressible flow is satisfied,

$$\nabla\cdot\left(\frac{Uh}{2} - \frac{h^3}{12\mu}\nabla p\right) = 0 \tag{3}$$

and the lubricant flow rate is given by

$$\left(\frac{\bar{Q}h}{A_f}\right) = \frac{Uh}{2} - \frac{h^3}{12\mu}\nabla p. \tag{4}$$

Since the cavity is at constant pressure, the boundary between the cavity and the continuous film region is a line of constant pressure. The pressure gradient in the continuous film is normal to this boundary. Therefore, matching the flow rates at the upstream edge of the cavity gives

$$\nabla p = 0;\ \text{(upstream cavity boundary condition)}. \tag{5}$$

Matching the flow rates at the downstream edge of the cavity gives the downstream cavity boundary condition. In dimensionless form this boundary condition is

$$H_1\Lambda = H_2\Lambda - H_2^3\left[\frac{1}{R^2}\frac{\partial P}{\partial \theta} - \frac{\partial P}{\partial R}\frac{d\theta}{dR}\right]. \tag{6}$$

Derivation of Short Bearing Cavitation Solution

Basic Equations

In the short bearing approximation, the pressure flow in the θ-direction is considered small compared to the pumping flow, $Uh/2$. Thus the Reynolds' equation reduces to

$$\frac{1}{R}\frac{d}{dR}\left(RH^3\frac{dP}{dR}\right) = \Lambda\frac{dH}{d\theta} \tag{7}$$

In the case where the film thickness, H, is not a function of the radius, R, the integration of this expression gives the following general solution,

$$P = \frac{\Lambda}{H^3}\frac{dH}{d\theta}\frac{R^2}{4} + \frac{C_1}{H^3}\ln R + C_2 \tag{8}$$

where C_1 and C_2 are constants of integration.

There are three separate regions and three sets of boundary conditions where the previous equations must be applied. These regions, shown in Fig. 19, are:

1 The full film region where the film extends across the face of the seal.

2 The outer film region where the film extends from the outer edge of the cavity to the outer edge of the seal.

3 The inner film region where the film extends from the inner edge of the seal to the inner edge of the cavity.

There is also the cavity region where the pressure is constant and equation (7) does not apply.

The general form of the boundary conditions is

$$P = P_m\ @\ R = R_m \text{ and } P = P_n\ @\ R = R_n \tag{9}$$

where the specific values for each region are given by the following table:

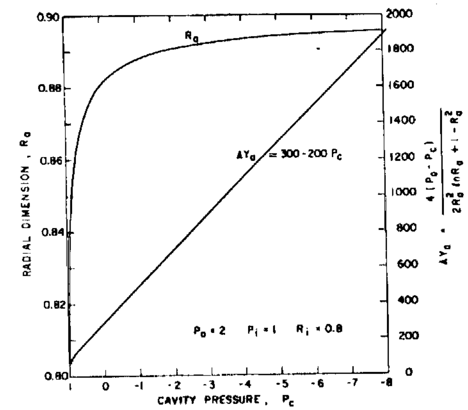

Fig. 3 Beginning of cavity radial dimension

	P_m	P_n	R_m	R_n	R_c
Full film region	P_i	P_o	R_i	1	—
Inner film region	P_i	P_c	R_i	R_1	R_1
Outer film region	P_c	P_o	R_2	1	R_2

The pressure and radial pressure gradient in any region have the following general form

$$(P - P_n) = \frac{\ln(R/R_n)}{\ln(R_m/R_n)}(P_m - P_n) + \frac{\Lambda}{4H^3}\frac{dH}{d\theta}$$
$$\times\left[(R^2 - R_n^2) + \frac{\ln(R/R_n)}{\ln(R_m - R_n)}(R_n^2/R_m^2)\right] \tag{10}$$

$$\frac{dP}{dR} = \frac{(P_m - P_n)}{R\ln(R_m/R_n)} + \frac{\Lambda}{4H^3}\frac{dH}{d\theta}\left[2R + \frac{(R_n^2 - R_m^2)}{R\ln(R_m/R_n)}\right] \tag{11}$$

Once the equation for the radial pressure gradient has been formulated, two equations can be determined for the cavity boundary shape from the cavity boundary conditions. For the upstream cavity boundary condition, the equation is

$$(\nabla P)_c = \left(\frac{dP}{dR}\right)_c = 0 \tag{12}$$

and for the downstream cavity boundary condition

$$(H_2 - H_1)\Lambda = -H_2^3\left(\frac{dP}{dR}\right)_c\frac{d\theta}{dR_c}. \tag{13}$$

The circumferential pressure gradient, $\partial P/\partial\theta$, has been neglected as in the original assumption.

The equation for the upstream cavity boundary shape is found by substituting equation (11) into equation (12),

$$\frac{1}{H^3}\frac{dH}{d\theta} = \frac{(P_n - P_m)}{\frac{\Lambda}{4}R_c\ln(R_m/R_n)\left[2R_c + \frac{(R_n^2 - R_m^2)}{R_c\ln(R_m/R_n)}\right]}. \tag{14}$$

The equation describing the downstream cavity boundary shape is obtained by substituting equation (11) into equation (13),

$$-\frac{dR_c}{d\theta} = \frac{H_2^3}{(H_2 - H_1)}\left\{\frac{(P_m - P_n)}{(\Lambda R_c\ln(R_m/R_n)}\right.$$
$$\left.+\frac{1}{4H^3}\frac{dH}{d\theta}\left[2R_c + \frac{(R_n^2 - R_m^2)}{R_c\ln(R_m/R_n)}\right]\right\}. \tag{15}$$

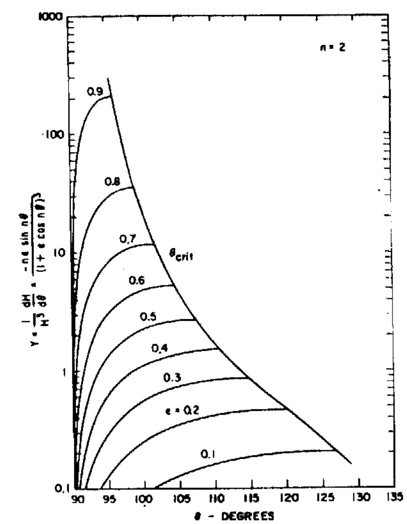

Fig. 4 Boundary function Y

Once the cavity shape ($R_1(\theta)$, $R_2(\theta)$) has been determined from equations (14) and (15), the film pressure and radial pressure gradient are obtained from equations (10) and (11). Then the seal load, leakage, and friction are calculated from the following expressions

$$C_L = \int_0^{2\pi}\int_{R_i}^{1} PRdRd\theta \tag{16}$$

$$(q_r)_i = \int_0^{2\pi} -H^3\left(\frac{dP}{dR}\right)_i R_i d\theta \tag{17}$$

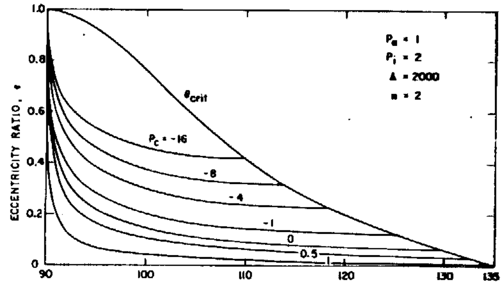

Fig. 5 Beginning of cavity

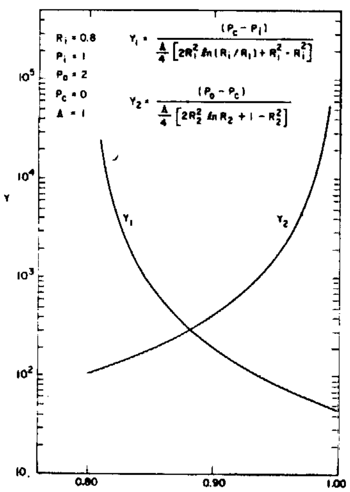

Fig. 6 Radial boundary functions

$$(q_r)_o = \int_0^{2\pi} -H^3 \left(\frac{dP}{dR}\right)_o d\theta \qquad (18)$$

$$C_f = \Lambda \int_0^{2\pi} \int_{R_i}^{1} \frac{R^2}{H}\, dR d\theta. \qquad (19)$$

The leakage rates at both the inside and outside radii are calculated and compared as a check on the accuracy of the calculations.

Method of Solution

The computations for this approximate solution were carried out on the General Electric Time Sharing Computer. The general procedure will be outlined in the following paragraphs.

Beginning of Cavity (R_a and θ_a). The first step in the calculation is to find at what radius the nose of the cavity is located. This is done by setting the two expressions for the upstream cavity boundary shape, equation (14), equal, and letting $R_1 = R_2 = R_a$. Thus

$$\frac{(P_o - P_c)}{(P_c - P_i)} = \frac{2R_a^2 \ln R_a + 1 - R_a^2}{2R_a^2 \ln (R_i/R_a) + R_a^2 - R_i^2} \qquad (20)$$

This equation gives R_a as a function of P_c/P_i, P_o/P_i, and R_i. Values of R_a as a function of P_c are given in Fig. 3.

Once R_a has been determined, θ_a can be determined from equation (14). Two more variables, Λ and ϵ, have been introduced in this equation. Therefore, θ_a is a function of P_c/P_i, P_o/P_i, R_i, Λ, and ϵ. A film thickness distribution given by

$$H = 1 + \epsilon \cos n\theta \qquad (21)$$

yields the following expression for θ_a

$$Y_a = \frac{1}{H^3}\frac{dH}{d\theta} = \frac{-n\epsilon \sin n\theta_a}{(1 + \epsilon \cos n\theta_a)^3} = \frac{4(P_o - P_c)}{\Lambda(2R_a^2 \ln R_a + 1 - R_a^2)} \qquad (22)$$

The right-hand side of equation (22) is plotted in Fig. 3 and the left-hand side in Fig. 4. A cross plot of these functions is given in Fig. 5 for $\Lambda = 2000$.

Intersection of Upstream and Downstream Boundaries (θ_{crit}, $R_{1\,crit}$, $R_{2\,crit}$)

Solutions to equation (14) give the upstream boundary shape of the cavity. This expression applies from (R_a, θ_a) to ($R_{1\,crit}$, θ_{crit}) and ($R_{2\,crit}$, θ_{crit}). The downstream limit of this equation occurs at the point where $dR/d\theta = 0$. This point can be determined by noting that equation (22) has a maximum at this point. Thus, from

$$\frac{d}{d\theta}\left[\frac{1}{H^3}\frac{dH}{d\theta}\right] = 0 \qquad (23)$$

the expression

$$\cos n\theta_{crit} = \frac{1 - \sqrt{1 + 24\epsilon^2}}{4\epsilon} \qquad (24)$$

is obtained for θ_{crit} as a function of ϵ. Once θ_{crit} had been obtained from equation (24) then

$$Y_{crit} = \left[\frac{1}{H^3}\frac{dH}{d\theta}\right]_{\theta_{crit}} = \frac{-n\epsilon \sin n\theta_{crit}}{(1 + \epsilon \cos n\theta_{crit})^3} \qquad (25)$$

can be evaluated. This function is plotted in Figs. 4 and 5. The corresponding values of radius, $R_{1\,crit}$ and $R_{2\,crit}$, are obtained by substituting equation (25) into equation (14) for regions 2 and 3 (see Fig. 6). Figs. 7 and 8 illustrate the radial extent of the cavities.

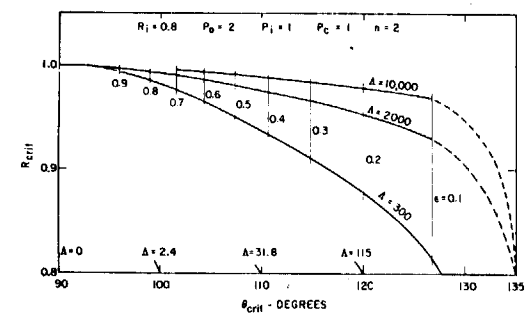

Fig. 7 Intersection of upstream and downstream boundaries (critical point)

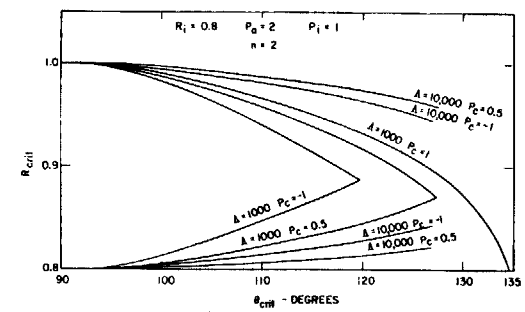

Fig. 8 Intersection of upstream and downstream boundaries (critical point)

Inception of the Cavity

It is important to know under what conditions and at what location a cavity will first appear. Since the cavity will first appear as a point, all the cavity conditions apply for the same value of R and θ. The required relationship can be determined by equating $Y_a(R_a)$ and Y_{crit} from equations (22) and (25). Therefore, the values of P_c, Λ, and ϵ, for which a cavity will first appear, can be determined. This is shown in Figs. 9 and 10.

Discussion of Results

Short Bearing Solutions

As has been noted, the conditions for a cavity to exist (inception of cavity), the location of the nose of the cavity, and the extreme radial dimensions of cavity (θ_{crit}, $R_{1\,crit}$, $R_{2\,crit}$), can be solved for directly. However, the downstream cavity boundary must be obtained by solving equation (15) stepwise. Fig. 11 shows the location of the tail of the cavity as a function of ϵ, Λ, and P_c for a given P_i and P_o. These results were obtained from the computer solutions.

The effect of cavity pressure on the load coefficient is illustrated in Figs. 12, 17, and 18 for $P_c = 1, 0.5$, and -1, respectively.

Several cavity shapes are shown in Fig. 19 for the case where there is no hydrostatic pressure differential across the seal.

As an example consider the following case:

$$P_o = 2 \quad P_i = 1 \quad P_c = -1$$

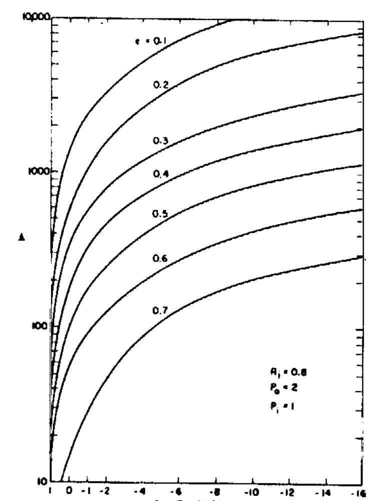

Fig. 9 Conditions for inception of cavity

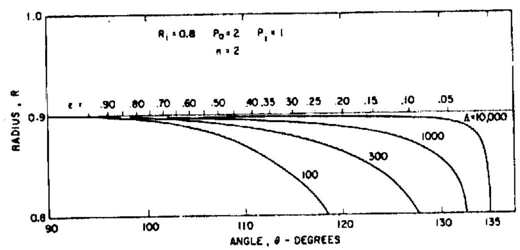

Fig. 10 Inception point of cavity

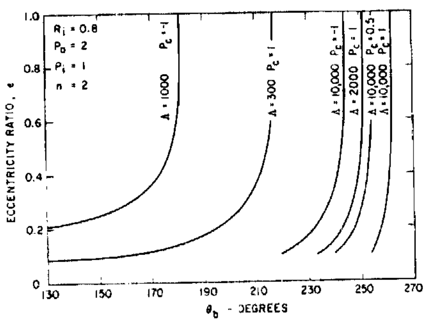

Fig. 11 End of cavity

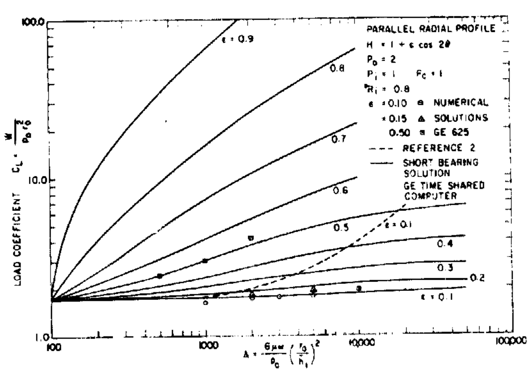

Fig. 12 Load capacity

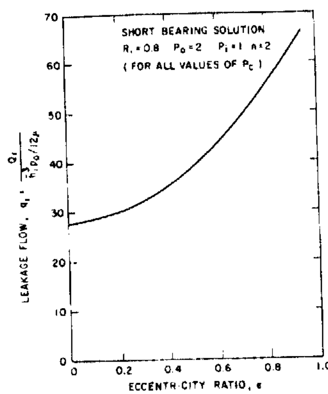

Fig. 13 Seal leakage

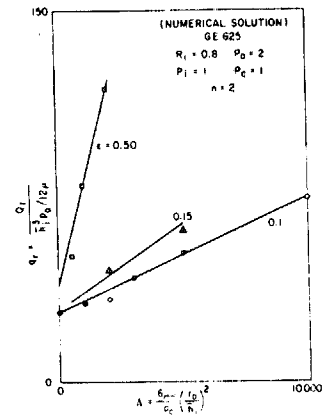

Fig. 14 Seal leakage

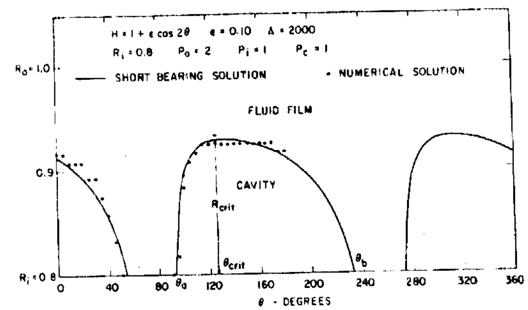

Fig. 15 Cavity shape

$\Lambda = 1000 \quad \epsilon = 0.3 \quad n = 2 \quad R_i = 0.8.$

The results are:

$R_a = 0.888$	Fig. 3, also Fig. 10 for $\Lambda = 575$
$\Lambda Y_a = 500$	Fig. 3
$Y_a = 500/1000 = 0.500$	
$\theta_a = 99$ deg	Fig. 4
$Y_{crit} = 0.87$	Fig. 4
$\theta_{crit} = 115$ deg	Fig. 4 and 5
$\Lambda_{min} = \frac{\Lambda Y_a}{Y_{crit}} = \frac{500}{0.87} = 575$	Also Fig. 9
Cavity first appears at	Figs. 9 and 10
$\Lambda = 575, \theta = 115$ deg	
$R = 0.888$	
$Y_1\Lambda = Y_2\Lambda = Y_{crit}\Lambda = 870$	
$R_{1\,crit} = 0.867$	Figs. 6 and 8
$R_{2\,crit} = 0.915$	Figs. 6 and 8
$\theta_b = 162$ deg	Fig. 11
$q = -32$	Fig. 13
$C_L = 1.85$	Fig. 18

Comparison With Numerical Solutions

Solutions to the two-dimensional Reynolds' equation, equation (3), were obtained using a numerical iterative program on the GE 625 computer. The boundary conditions, equations (5) and (6), were applied to these solutions. This required a double iterative approach (i.e., an iterative solution to Reynolds' equation for a given cavity shape and then an iterative solution for the correct cavity shape).

Convergence of these solutions has been limited to moderate values of ϵ and Λ. These solutions are compared to the short bearing solutions in Figs. 12, 15, and 16. Fig. 12 shows that the values of load coefficient given by the two methods agree quite closely. Also, Figs. 15 and 16 show that the cavity shape and leakage distribution are the same for moderate values of Λ and ϵ. However, at higher values of Λ, the leakage rates do not agree nearly as well. This can be seen from Figs. 13 and 14.

Fig. 13 shows that the short bearing solution gives leakage rates which are a function of P_b, P_0 and ϵ only. The cavity pressure, P_c, and the seal number, Λ, do not influence it.

The numerical solutions, where $\partial P/\partial \theta$ was not neglected, gave leakage rates agreeing with short bearing solutions for $\Lambda = 0$ but increased linearly with Λ as seen in Fig. 14.

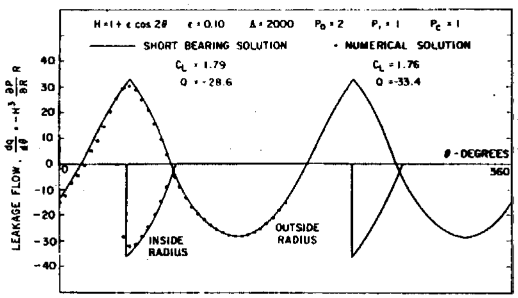

Fig. 16 Leakage flow distribution at edges of seal

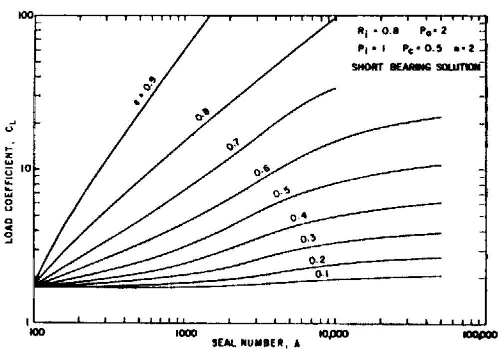

Fig. 17 Load capacity

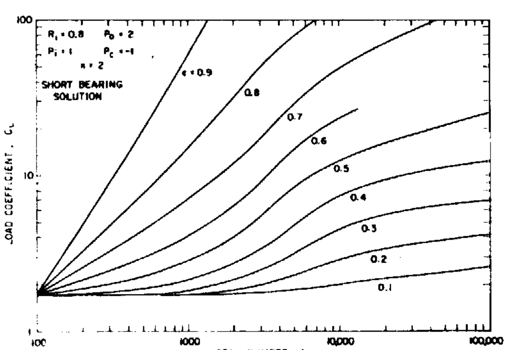

Fig. 18 Load capacity

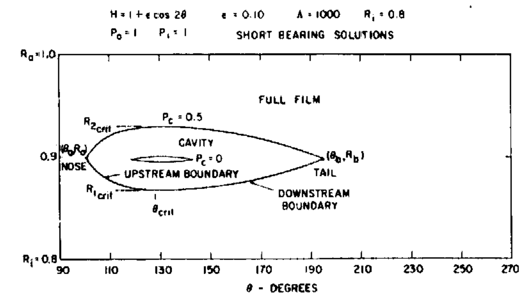

Fig. 19 Cavity shapes

Conclusions

The short bearing approximation provides a very useful solution to the problem of a mechanical face seal operating on a thin hydrodynamic film containing gas cavities.

These solutions compare well with numerical solutions to the two-dimensional Reynolds' equations in the moderate Λ, ε-range. The quality of the approximation at higher values of Λ and ε is not known.

The presence of a gas cavity in the film does affect the load on the seal. For the case studied, there would be no hydrodynamic effect on the seal load and leakage without the cavity.

The solutions presented here satisfy continuity throughout the seal film.

Future Work

An experimental program is being carried out under a follow-on to NASA Contract No. NAS 7-434. The purpose of this study will be to demonstrate the existence of cavities of the type calculated in this paper and to determine to what extent this theory represents actual face seal behavior.

A more complete discussion of the work in this paper is given in references [5 and 6].

Acknowledgment

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III. E

Paper No. 68—Lub-2

Inward Pumping in Mechanical Face Seals

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The hydrodynamic characteristics which produce inward pumping in mechanical face seals are studied. Solutions to Reynolds' equation for hydrodynamic lubrication are obtained using both the short-bearing approximation and numerical iterative methods. It is found that pumping in face seals results from combined radial eccentricity and misalignment, or one circumferential surface wave. Experimental results confirm the theory.

Introduction

Hydrodynamic effects have been shown to exist in mechanical face seals [1].[1] Seal forces and leak rates are primary considerations in the design and application of face seals. Therefore, it is important to understand the basic principles of face seal operation (see Fig. 1).

An anomalous behavior observed in face seal experiments by Denny [1], Nau [2], and others is that of "inward pumping." This is a condition where fluid or air leaks through the seal in the opposite direction to the hydrostatic pressure drop across the seal. The term inward pumping was attached to this phenomenon because in the seal experiments where it was observed the inside of the seal was pressurized. Therefore, any leakage in the outward direction was attributed to the hydrostatic pressure differential while observed leakage toward the center of the seal and against the pressure was attributed to a pumping action in the seal.

So far, no satisfactory explanation of this phenomenon has been set forth. Nau [3], in a recent paper based on experiments using a transparent seal apparatus, confirmed his earlier postulate [4] that lubricant cavitation was the cause for inward pumping. He also concluded that this pumping action occurred only in the inward direction. Lyman and Saibel [5], using an approximate solution to the face seal hydrodynamic problem with gas cavities, were able to calculate an inward pumping rate for the case of no pressure difference across the seal. Findlay [6—8] showed analytically that cavity solutions using the Swift-Floberg [16, 17] boundary conditions—i.e., conservation of flow at the cavity boundaries—did not result in any inward pumping. Subsequent experimental results reported here and in [9] confirm this conclusion and substantiate the short-bearing theory put forth in [6].

A theory is developed here which shows that a combination of radial eccentricity and misalignment, or one circumferential film-thickness wave, will result in a "pumping" leakage which may be either inward or outward. Experimental results reported here and in [9] confirm this theory, which was developed for a full film, i.e., no gas cavities. Experiments and theory for the eccentric seal with gas cavities show that rather than causing inward pumping the cavities reduce the pumping effect.

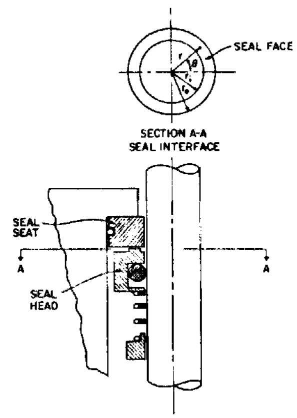

Fig. 1 Mechanical face seal geometry

[1] Numbers in brackets designate References at end of paper.

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Derivation of Short-Bearing Pumping Solution

General Equations for Wavy Seal

Reynolds' equation for the radially eccentric face seal is

$$\frac{\partial}{\partial R}\left[RH^3\frac{\partial P}{\partial R}\right] + \frac{1}{R}\frac{\partial}{\partial \theta}\left[H^3\frac{\partial P}{\partial \theta}\right] = \Lambda\left\{\frac{\partial}{\partial R}\left[RHU'_r\right] + \frac{\partial}{\partial \theta}\left[HU'_\theta\right]\right\} \quad (1)$$

where, from Fig. 2, the relative seal surface velocities are given by the following expressions:

Nomenclature

A = seal area, sq in.
A_f = flow area, sq in.
h = film thickness, in.
$\bar{h}_i$ = average film thickness at inside radius, in.
ρ = density, lb/cu in.
r = coordinate measured from center of seal, in.
θ = angular coordinate, rad
U = fluid velocity, ips
ω = rotational speed, rad/sec
Q = flow rate, cu in/sec
p = pressure, psi
τ = shear stress, psi
W = seal load capacity, lb$_f$
SM = shaft friction moment, in-lb$_f$
μ = dynamic viscosity, psi-sec

Dimensionless Terms

R = radius ratio = r/r_0
H = film thickness ratio = $h/\bar{h}_i$
P = pressure ratio = p/p_0
Λ = seal number = $\frac{6\mu\omega}{p_0}\left(\frac{r_0}{\bar{h}_i}\right)^2$
ϵ = waviness parameter
C_L = load coefficient = $\frac{W}{p_0 r_0^2}$

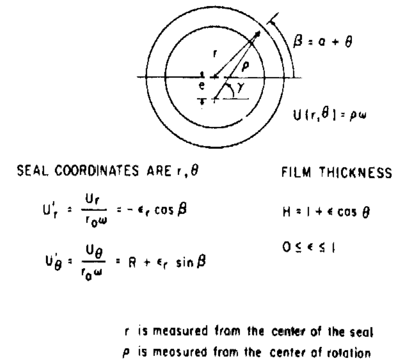

Fig. 2 Fixed seal, rotating shaft

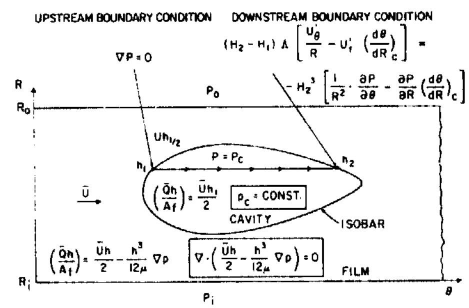

Fig. 3 Cavity boundary conditions

$$U'_r = -\epsilon_r \cos\beta \quad (2)$$

$$U'_\theta = R + \epsilon_r \sin\beta \quad (3)$$

Using the short-bearing approximation, where the $\partial p/\partial\theta$ term is neglected, and assuming that the film thickness is a function of θ only, as in the wavy seal case, equation (1) reduces to

$$\frac{d}{dR}\left[RH^3\frac{dP}{dR}\right] = \Lambda(R + \epsilon_r \sin\beta)\frac{dH}{d\theta} \quad (4)$$

Integration of this expression gives the following general solution:

$$P = \frac{\Lambda}{H^3}\left(\frac{R^2}{4} + R\epsilon_r \sin\beta\right)\frac{dH}{d\theta} + \frac{C_1}{H^3}\ln R + C_2 \quad (5)$$

where C_1 and C_2 are constants of integration which are determined from the boundary conditions on the pressure.

For the case where gas cavities may be present in the fluid film [6], there are three separate film regions and three sets of boundary conditions where equation (5) must be applied. These regions, shown in Fig. 3, are:

1 The full film region, where the film extends across the face of the seal.

2 The outer film region, where the film extends from the outer edge of the cavity to the outer edge of the seal.

3 The inner film region, where the film extends from the inner edge of the seal to the inner edge of the cavity.

There is also the cavity region, where the pressure is constant and equations (4) and (5) do not apply.

The general form of the boundary conditions is

$$P = P_m \text{ at } R = R_m,\; P = P_n \text{ at } R = R_n \quad (6)$$

where the specific values for each region are given as follows:

	P_m	P_n	R_m	R_n	R_c
Full film region	P_i	P_o	R_i	1	—
Inner film region	P_i	P_c	R_i	R_1	R_1
Outer film region	P_c	P_o	R_2	1	R_2

The pressure and radial pressure gradient in any region have the following general form:

$$(P - P_n) = \frac{\ln(R/R_n)}{\ln(R_m/R_n)}(P_m - P_n) + \frac{\Lambda}{4H^3}\frac{dH}{d\theta}\Big\{(R^2 - R^2_n) + 4(R - R_n)\epsilon_r \sin\beta + \frac{\ln(R/R_n)}{\ln(R_m/R_n)}[(R^2_m - R^2_n) + 4(R_m - R_n)\epsilon_r \sin\beta]\Big\} \quad (7)$$

$$\frac{dP}{dR} = \frac{(P_m - P_n)}{R\ln R_m/R_n} + \frac{\Lambda}{4H^3}\frac{dH}{d\theta}\left\{\left[2R + \frac{(R^2_n - R^2_m)}{R\ln(R_m/R_n)}\right] + 4\left[1 + \frac{(R_n - R_m)}{R\ln(R_m/R_n)}\right]\epsilon_r \sin\beta\right\} \quad (8)$$

Full Film—Single Fluid Case

If it is assumed that the same fluid is present on both sides of the seal and that the fluid film is complete, i.e., no gas cavities, then, equation (8) becomes

$$\frac{dP}{dR} = \frac{(P_i - P_o)}{R\ln R_i} + \frac{\Lambda}{4H^3}\frac{dH}{d\theta}\left\{\left[2R + \frac{(1 - R^2_i)}{R\ln R_i}\right] + 4\epsilon_r \sin\beta\left[1 + \frac{(1 - R_i)}{R\ln R_i}\right]\right\} \quad (9)$$

The leakage coefficient for the radially eccentric seal is defined as

$$q = \int_0^{2\pi}\left[\Lambda HU'_r - H^3\frac{dP}{dR}\right]Rd\theta \quad (10)$$

Therefore, integrating at $R = R_o = 1$, we have

$$q = \int_0^{2\pi}\Big\{-\Lambda H\epsilon_r \cos\beta - H^3\frac{(P_i - P_o)}{\ln R_i} - \frac{\Lambda}{4}\frac{dH}{d\theta}\left[2 + \frac{1 - R^2_i}{\ln R_i}\right] - \Lambda\frac{dH}{d\theta}\epsilon_r \sin\beta - \Lambda\frac{dH}{d\theta}\epsilon_r \sin\beta\frac{(1 - R_i)}{\ln R_i}\Big\}d\theta$$

$$= -\frac{(P_i - P_o)}{\ln R_i}\int_0^{2\pi}H^3d\theta - \Lambda\epsilon_r\frac{(1 - R_i)}{\ln R_i}\int_0^{2\pi}\frac{dH}{d\theta}\sin\beta d\theta - \frac{\Lambda}{4}\left[2 + \frac{1 - R^2_i}{\ln R_i}\right]\int_0^{2\pi}\left(\frac{dH}{d\theta}\right)d\theta - \Lambda\epsilon_r\int_0^{2\pi}\frac{d}{d\theta}[H\sin\beta]d\theta \quad (11)$$

If a wavy seal surface is assumed, the film thickness is given by

$$H = 1 + \epsilon\cos n\theta \quad (12)$$

For this case, the leakage coefficient becomes

$$q = -\frac{(P_i - P_o)}{\ln R_i}\int_0^{\pi}(H\epsilon\cos n\theta)^3d\theta - \Lambda\epsilon_r\frac{(1 - R_i)}{\ln R_i} \quad (13)$$

$$\times \int_0^{2\pi} (-en \sin n\theta) \sin (\alpha + \theta) d\theta$$

$$= -\frac{(P_i - P_o)}{\ln R_i} 2\pi(1 + 1.5 e^2) + \Lambda \epsilon_r \epsilon \pi \frac{(1 - R_i)}{\ln R_i} \cos \alpha \quad (13) \; (Cont.)$$

for $n = 1$. It will be noted that for $n > 1$ the hydrostatic leakage is unchanged but the pumping leakage becomes zero.

The first term in equation (13) is the seal leakage due to the hydrostatic pressure differential. The second term is the leakage due to the radial eccentricity of the seal coupled with the circumferential variation in the film thickness. This second leakage term may either be inward or outward, depending upon α, the phase angle between the surface waviness and the eccentricity vector (Fig. 2), or upon the direction of rotation, ω, which appears in Λ.

The shaft-seal radial eccentricity that will make the net leakage zero is

$$e = \epsilon_r r_o = \frac{(p_i - p_o)\bar{h}^2_i(1 + 1.5\epsilon^2)}{3\mu\omega(r_o - r_i)\epsilon \cos \alpha} \quad (14)$$

For the minimum value of e in equation (14), we find that

$$\cos \alpha = 1; \alpha = 0 \text{ deg} \quad (15)$$

$$(1 + 1.5\epsilon^2)/\epsilon = 2.45 \quad \epsilon = 0.817 \quad (16)$$

Note that $0 \le \epsilon \le 1$ and, at $\epsilon = 1$, $(1 + 1.5\epsilon^2)/\epsilon = 2.50$.

Substituting (15) and (16) into (14) and using the test seal conditions—i.e., $r_o = 1.75$ in., $r_i = 1.125$ in., $\mu = 4.95 \times 10^{-6}$ psi-sec (350 centistokes)—the result is

$$e = 2.52 \times 10^6(p_i - p_o)\bar{h}^2_i/N(\text{rpm}) \quad (17)$$

In order to estimate the order of magnitude of e, let the following conditions be assumed:

$$(p_i - p_o) = 5 \text{ psi}$$

$$\bar{h}_i = 10^{-3} \text{ in.}$$

$$N = 10^3 \text{ rpm}$$

Therefore, for this numerical example

$$e = 1.26 \times 10^{-3} \text{ in.} \quad (18)$$

It would not be unusual to have a shaft-seal radial eccentricity of this magnitude. Even though the viscosity assumed in this calculation is quite high, the film thickness is much larger than normally would occur in a face seal. Further, if the equation (13) for the seal leakage is written in dimensional terms, i.e.,

$$Q = -\frac{2\pi(1 + 1.5\epsilon^2)\bar{h}^3_i(p_i - p_o)}{12\mu \ln R_i} + 6\pi\omega r^2_o\epsilon_r\epsilon \frac{(1 - R_i)}{12 \ln R_i} \bar{h}_i \cos \alpha \quad (19)$$

$$= Q_s + Q_{pump}$$

then it will be noted that

$$Q_s \propto \bar{h}^3_i$$

$$Q_{pump} \propto \bar{h}_i \quad (20)$$

Therefore, as the average seal film thickness $\bar{h}_i$ decreases, the pumping component of the leakage becomes relatively more significant.

Thus it has been shown that, for mechanical face seals, the pumping leakage due to radial eccentricity can be a very important factor and, for the small film thicknesses that normally occur, only a small radial eccentricity is needed. In fact, it becomes a question whether or not this phenomenon can be completely eliminated from face seal experiments and applications.

Since at this time no numerical solutions have been obtained for the case where gas cavities are present in the film of the radially eccentric seal, the equations and boundary conditions for this problem are presented in Appendix 1.

Misaligned Eccentric Seal

As will be explained later, the experimental results were obtained for the misaligned, radially eccentric seal rather than the wavy, radially eccentric seal. Therefore, an analysis of this problem is of importance in verifying the seal hydrodynamic theory developed herein.

The wavy and misaligned cases approach one another as the seal radius ratio, R_i, approaches unity. The wavy solution is presented because it is much easier to obtain and the form of the final expression for leakage is simple and, thus, more readily understood.

It was found to be more convenient to obtain solutions to the misaligned, full-film problem using numerical computer solutions. The FORTRAN IV program used on the GE 635 computer for this purpose is explained in more detail in Appendix 2. The approach used shows that the "short bearing" approximation is very good.

For the misaligned seal, the film thickness distribution is given by

$$H = 1 + R\epsilon_t \cos \theta \quad (21)$$

and, therefore, is no longer a function of θ only. Thus equation (1) becomes

$$\frac{\partial}{\partial R}\left[RH^3 \frac{\partial P}{\partial R}\right] + \frac{1}{R}\frac{\partial}{\partial \theta}\left[H^3 \frac{\partial P}{\partial R}\right] = \Lambda \left\{RU'_r \frac{\partial H}{\partial R} + U'_\theta \frac{\partial H}{\partial \theta}\right\}$$

$$= \Lambda \{(-R\epsilon_r \cos \beta)(\epsilon_t \cos \theta) + (R + \epsilon_r \sin \beta)(-R\epsilon_t \sin \theta)\}$$

$$= \Lambda \{-R\epsilon_r\epsilon_t \cos \alpha - R^2\epsilon_t \sin \theta\} \quad (22)$$

The short-bearing approximation could be used here to obtain a solution. However, this approach requires using some series approximations. Since it was already available, a numerical computer program was used instead. Equation (22) was solved in two parts. The first was for the hydrostatic leakage where ϵ_r, the radial eccentricity ratio, was zero and the boundary conditions were $P = P_i$ at $R = R_i$, $P = P_o$ at $R = 1$. These solutions are plotted in Fig. 4 and take the following form,

$$-\frac{q_s \ln R_i}{2\pi(P_i - P_o)} = -\frac{Q_s 12\mu \ln R_i}{2\pi\bar{h}^3_i(p_i - p_o)} = 1 + a\epsilon^2_t \quad (23)$$

where the slope, a, is a function of R_i and is shown in Fig. 5.

The second part of the solution to equation (22) is for the pumping component and is obtained by using a finite value for ϵ_r and applying the boundary conditions $P_i = P_o = 0$. As can be seen from equation (22), the result will be in terms of the constant factor $\Lambda\epsilon_r\epsilon_t \cos \alpha$, the same as for the wavy case. Values for

$$\frac{q_{pump}}{\Lambda\epsilon_r\epsilon_t \cos \alpha} = \frac{2Q_{pump}}{\epsilon_r\epsilon_t \cos \alpha \, \bar{h}_i\omega r_o^2} = f(R_i, \epsilon_t) \quad (24)$$

are plotted in Fig. 6.

Experimental Results

The experimental apparatus used is described in [9, 10]. Fig. 7 shows the comparison between theory and experiment for the full-film outward pumping case. Fig. 8 shows the comparison for the inward pumping case. In this latter set of data it was difficult to supply enough oil to the reservoir surrounding the outside of the seal because of the net inward pumping taking place. Therefore, only low-speed data were obtained.

When the outside of the seal is not flooded, gas cavities appear

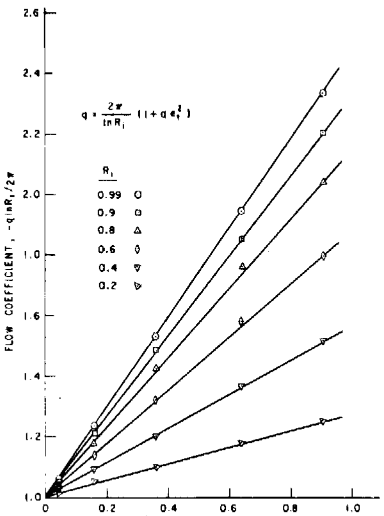

Fig. 4 Hydrostatic flow coefficient for misaligned seal

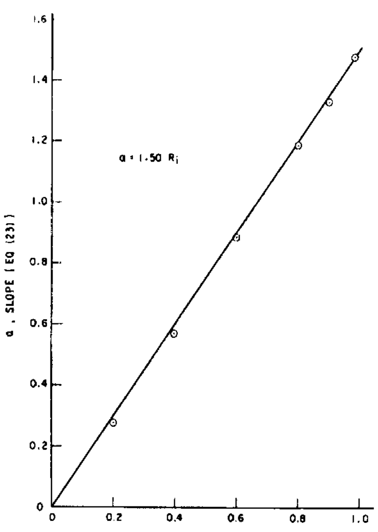

Fig. 5 Hydrostatic flow constant for misaligned seal

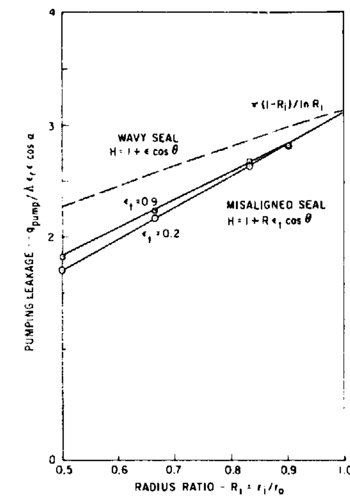

Fig. 6 Seal pumping leakage due to radial eccentricity

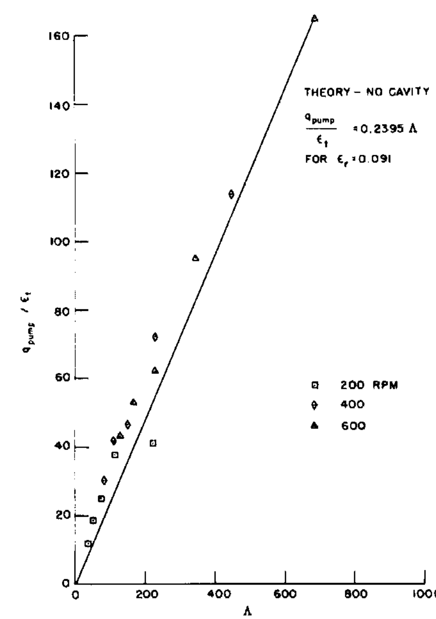

Fig. 7 Outward pumping due to radial eccentricity

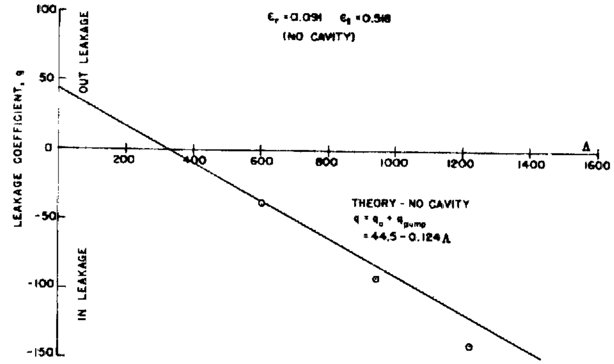

Fig. 8 Leakage coefficient showing the effects of inward pumping

Fig. 9 Concentric, transparent face seal showing gas cavity

Fig. 10 Concentric, transparent face seal showing gas cavity (smaller cavity, lower speed)

in the seal interface as shown in Figs. 9-12. When this occurs, the inward pumping effect is diminished, as can be seen by the data plotted in Figs. 13-16.

Data are currently being taken (*a*) at higher speeds to show the effects of fluid inertia [11, 12], and (*b*) for different viscosity fluids on either side of the seal. These data will be presented in [10, 13].

Discussion

Correlation of Theory

For the full-film case the correlation between experiment and theory was quite good. To date, numerical results have not been obtained from the theory developed for the radially eccentric seal containing gas cavities. The equations for the full-film, two-fluid problem have been derived but not set up for solution as yet. It is hoped that these two solutions can be pursued further in [10].

Previous Investigations

The work of Denny [1] demonstrated the existence of hydrodynamic forces in mechanical face seals. He and Nau [3] both observed the inward pumping phenomenon. Nau [2, 3] also observed gas cavities in transparent-seal experiments. Because he observed these two phenomena simultaneously, he was led to the conclusion that the cavitation produced the pumping leakage. Lyman and Saibel [5], in their approximate cavitation seal analysis, tried to show theoretically that gas cavities did cause inward pumping in addition to hydrodynamic lift forces. Orcutt and Cheng [14] and Pape [15] also used approximate

Fig. 11 Eccentric, transparent face seal showing gas cavity and inward pumping of air (note spiral of bubbles in center)

Fig. 12 Eccentric, transparent face seal showing gas cavity (note many bubbles in center due to higher speed)

cavity solutions to demonstrate the relationship between this phenomenon and the hydrodynamic lifting force observed by Denny [1].

All these solutions [5, 14, 15], neglected to use conservation of flow boundary conditions (Swift-Floberg) at the edges of the cavities. Thus continuity of flow was not maintained. Pape [15] recognized this fact and Orcutt and Cheng [14] did not use their solution to calculate flow rates; for, if they had, the same leakage rate would not have been obtained at both the inner and outer radii. This was the mistake that Lyman and Saibel [5] made. They integrated the local, radial flow rates at the inside radius of the seal and obtained an inward leakage when cavities were assumed. If, however, they had done the same at the outer radius they would have obtained an outward leakage for the same case. Thus a basic inconsistency existed in their solution and they were misled to believe that cavitation produces inward pumping in face seals.

Conclusions

1 The short-bearing cavity solution presented in [6] has been confirmed experimentally.

2 The pumping theory developed in this paper has been confirmed experimentally.

3 In concentric face seals of fixed geometry, cavitation produces a net hydrodynamic lifting force and moment, but does not alter the leakage rate; i.e., it does not produce an inward pumping leakage.

4 Pumping leakage is produced by a combination of radial eccentricity and seal misalignment (tilt) or one-wave waviness.

5 The aforementioned pumping effect can be either inward or outward, depending upon the relationship between the radial eccentricity and misalignment, and the direction of rotation.

6 As long as a complete fluid film exists in the seal interface, the hydrostatic pressure differential does not affect the pumping leakage and produces a leakage which is independent of speed and, of course, always in the direction of decreasing pressure.

7 The existence of gas cavities in the film reduces the pumping effect, in eccentric seals, rather than causing it as heretofore postulated.

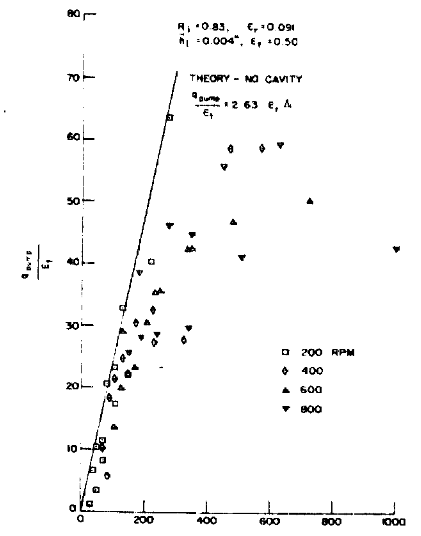

Fig. 13 Pumping coefficient for eccentric face seal with cavities

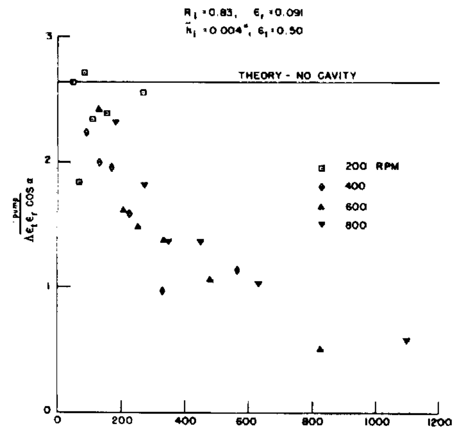

Fig. 14 Pumping coefficient for eccentric face seal with cavities

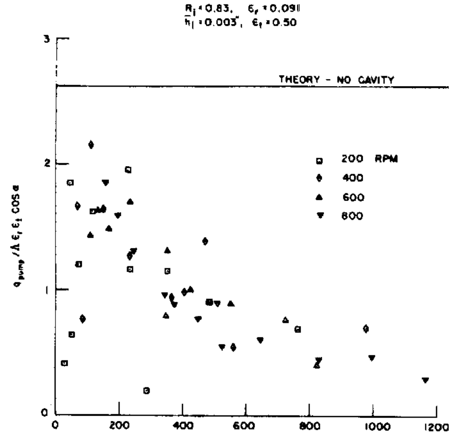

Fig. 15 Pumping coefficient for eccentric face seal with cavities

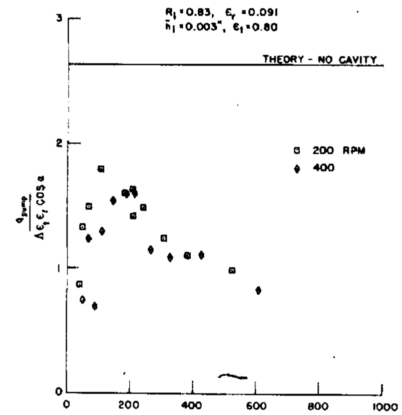

Fig. 16 Pumping coefficient for eccentric face seal with cavities

Future Work

An experimental program is now being completed to verify the predicted effects of fluid inertia [11,12], and to study the influence of having fluids of different viscosities on either side of the seal.

Also, the equations presented in Appendix 1 for the eccentric seal with cavities are being programmed. The two-fluid analysis is being pursued, but may not be completed under the current NASA contract.

Acknowledgment

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APPENDIX 1

Eccentric Seal With Cavity

Once equation (8) for the radial pressure gradient has been formulated, two equations can be determined for the cavity boundary shape from the cavity boundary conditions. Using the short-bearing approximation, the upstream cavity boundary condition is

$$(\nabla P)_c = \left(\frac{dP}{dR}\right)_c = 0 \tag{25}$$

and for the downstream cavity boundary condition

$$(H_2 - H_1)\Lambda\left[\frac{U'_\theta}{R} - U'_r\left(\frac{d\theta}{dR}\right)_c\right] = -H^3_2\left[\frac{1}{R^2}\frac{\partial P}{\partial\theta} - \frac{\partial P}{\partial R}\left(\frac{d\theta}{dR}\right)_c\right] \tag{26}$$

$$= H^3_2\left[\left(\frac{dP}{dR}\right)_c\left(\frac{d\theta}{dR}\right)_c\right] \tag{27}$$

The circumferential pressure gradient, $\partial P/\partial\theta$, has been neglected as in the original assumption.

The equation for the upstream cavity boundary shape is found by substituting equation (8) into equation (25):

$$\frac{1}{H^3}\frac{dH}{d\theta} = \frac{(P_m - P_n)}{\frac{\Lambda}{4}\left\{2R^2_c \ln R_n/R_m + (R^2_m - R^2_n) + 4\epsilon_r \sin\beta[R_c \ln R_n/R_m + (R_m - R_n)]\right\}} \tag{28}$$

The equation describing the downstream cavity boundary shape is obtained by substituting equation (8) into equation (27):

$$\left(\frac{dR}{d\theta}\right)_c = \frac{H^3_2R_c}{(H_1 - H_2)U'_\theta\Lambda}\left(\frac{dP}{dR}\right)_c + \frac{R_cU'_r}{U'_\theta} = \frac{H^3_2R_c}{(H_1 - H_2)U'_\theta\Lambda} \times \left\{\frac{P_m - P_n}{R_c \ln R_m/R_n} + \frac{\Lambda}{4H^3_2}\frac{dH_2}{d\theta}\left[2R_c + \frac{(R^2_n - R^2_m)}{R_c \ln R_m/R_n} + 4\epsilon_r \sin\beta\left[1 + \frac{(R_n - R_m)}{R_c \ln R_m/R_n}\right]\right]\right\} + \frac{R_cU'_r}{U'_\theta} \tag{29}$$

It should be noted that, when the appropriate substitutions are made for n and m, for the various regions discussed before, R_1 will be the inner cavity boundary, i.e., the boundary closest to R_i, and R_2 will be the outer cavity boundary, i.e., closest to R_o. This Nomenclature is not to be confused with H_1, which is the film thickness on the upstream boundary of the cavity, and H_2, the film thickness on the downstream boundary (Fig. 3). By definition, the upstream cavity boundary is where the fluid streamers leave the complete film region and enter the cavity region, and the downstream cavity boundary is where these same streamers reenter the full-film region.

The point where the cavity boundary is tangent to the relative surface velocity vector distinguishes the difference between the upstream and the downstream cavity boundaries. This point, therefore, is located where the following expression is satisfied:

$$\left(\frac{dR}{d\theta}\right)_c = \frac{R_cU'_r}{U'_\theta} \tag{30}$$

Note that equations (29) and (30) are consistent since at this point of intersection the upstream boundary condition, equation (25), also applies and equation (29) reduces to equation (30).

The remaining required expression is the one needed to solve for H_1. The H_1 which corresponds to a given H_2 in equation (29), i.e., to a specific point on the downstream boundary, is found by tracing the streamer path through the cavity. That is, the proper H_1 for a given H_2 is obtained by determining the departure point on the upstream boundary for the fluid streamer entering the downstream boundary at H_2 (see Fig. 3). For a given point on the downstream boundary of the cavity (R_d, θ_d), the required coordinates on the upstream boundary (R_u, θ_u) are obtained from the following equation:

$$R^2_u + 2\epsilon_rR_u \sin\beta_u = R^2_d + \epsilon_rR_d \sin\beta_d \tag{31}$$

These equations are sufficient to calculate the dimensions of the gas cavity and thus yield the solution for pressure in the seal film. From this solution, the load, flow, and torque coefficients may be calculated:

$$C_L = \int_0^{2\pi}\int_{R_i}^1 PRdRd\theta \tag{32}$$

$$q = \int_0^{2\pi}\left((\Lambda HU'_r) - H^3\frac{\partial P}{\partial R}\right)Rd\theta,\ R = \text{const} \tag{33}$$

$$C_f = \int_0^{2\pi}\int_{R_i}^1\left(\Lambda\frac{U'_\theta}{H} + \frac{3H}{R}\frac{\partial P}{\partial\theta}\right)R^2dRd\theta \tag{34}$$

Note that equation (34) represents the torque about the center of the seal.

APPENDIX 2

Seal Computer Program

Although it is nice to have closed-form analytical solutions to the various face seal problems, it sometimes becomes necessary to resort to numerical computer solutions. Once the basic problem has been set up on the computer, it will usually handle various seal film geometries equally well, whereas the analytical solutions hold for more limited conditions. Therefore, the GE 635 Seal Program has proven very useful. However, the computer solutions to the gas cavity problems require a large number of iterations and it is desirable to improve the numerical approach so as to reduce the computer time.

A method has been developed that not only conserves computer time but also gives a numerical solution to the short-bearing approximation as the first iteration in the process. This is a variation of a method described by Castelli and Privics [18]. In [18] the film pressures in a bearing are obtained by direct solution of a finite set of simultaneous equations. This approach still requires a large number of computations and in some cases may not reduce the computer time over that required by an iterative solution.

It has been recognized that in the seal problem the influence of the circumferential pressure gradient is much less than that of the radial pressure gradient. Thus the short-bearing approximation, wherein $\partial P/\partial\theta$ is neglected altogether, has resulted in very good solutions to the face seal problem.

The new method of solution used in the GE 635 Seal Program makes use of the fact that a very good approximation can be obtained by neglecting $\partial P/\partial\theta$, and solves the short-bearing approximation to Reynolds' equation numerically. However, this numerical solution is obtained by direct calculation of the simultaneous equations rather than by iteration. This is a very quick computation since the number of simultaneous equations is equal only to the number of radial grid points taken in the numerical approximation. Once the short-bearing solution is calculated and printed out, the $\partial P/\partial\theta$ terms in Reynolds' equation are included as correction terms. Several corrective calculations are then made in order to obtain the two-dimensional solution to Reynolds' equation.

The development of this new method of numerical solution starts from the numerical formulation of equation (1):

$$D_j P_{ij} = \frac{1}{2\Delta\theta^2}[P_{i+1,j} + P_{i-1,j}]$$
$$+ \frac{3}{4\Delta\theta H_{ij}}\left(\frac{\partial H}{\partial\theta}\right)_{ij}[P_{i+1,j} - P_{i-1,j}] + \frac{R^2_j}{2\Delta R^2}[P_{i,j+1} + P_{i,j-1}]$$
$$+ \frac{1}{4\Delta R}\left(\frac{3R^2_j}{H_{ij}}\frac{\partial H}{\partial R_{ij}} + R_j\right)[P_{i,j+1} - P_{i,j-1}]$$
$$- \frac{\Lambda R^2_j}{2H^2_{ij}}\left\{\frac{\partial}{\partial R}(U'_r H) + \left(\frac{U'_r H}{R}\right) + \frac{1}{R_j}\frac{\partial}{\partial\theta}(U'_\theta H)\right\}_{ij} \quad (35)$$
$$D_j = \left[\frac{1}{\Delta\theta^2} + R^2_j\frac{1}{\Delta R^2}\right] \quad (36)$$
$$H = (1 + a\xi + b\xi^2)^m + \epsilon\cos n\theta + \epsilon_t R\cos(\theta + \psi) + \epsilon_r \quad (37)$$

These equations can be combined as follows:

$$P_{ij} = A_{1j}P_{i+1,j} + A_{2j}P_{i-1,j} + A_{3j}P_{i,j+1} + A_{4j}P_{i,j-1} - A_{5j} \quad (38)$$

where

$$A_{1j} = [(1/2\Delta\theta^2) + (3/4\Delta\theta H_{ij})(\partial H/\partial\theta)_{ij}]/D_j \quad (39)$$
$$A_{2j} = [(1/2\Delta\theta^2) - (3/4\Delta\theta H_{ij})(\partial H/\partial\theta)_{ij}]/D_j \quad (40)$$
$$A_{3j} = \{(R^2_j/2\Delta R^2) + (1/4\,\Delta R) \times [(3R^2_j/H_{ij})(\partial H/\partial R)_{ij} + R_j]\}/D_j \quad (41)$$
$$A_{4j} = \{(R^2_j/2\Delta R^2) - (1/4\,\Delta R) \times [(3R^2_j/H_{ij})(\partial H/\partial R)_{ij} + R_j]\}/D_j \quad (42)$$
$$A_{5j} = (\Lambda R^2_j/2H^2_{ij})\left\{\frac{\partial}{\partial R}(U'_r H) + (U'_r H/R) + \frac{1}{R_j}\frac{\partial}{\partial\theta} \times (U'_\theta H)\right\}\Big/ D_j \quad (43)$$
$$D_j = 1/\Delta\theta^2 + R^2_j/\Delta R^2 \quad (44)$$

If it is assumed that the pressures in the circumferential direction ($P_{i+1,j}$, $P_{i-1,j}$) are known or assumed, then one may start at one of the boundaries of the seal and work across the seal obtaining expressions for P_{ij} in terms of the starting point boundary pressure and $P_{i,j+1}$. Thus, proceeding across the seal to the other boundary pressure, the last $P_{i,j+1}$ is known—i.e., it is the second boundary pressure and working back across the seal, all P_{ij}'s are determined on that radial line. Going on to the next radial line the procedure is repeated.

The equations used for this procedure

$$P_{ij} = X_j P_{i,j+1} + Y_j \quad (45)$$

where

$$X_j = A_{3j}/(1 - A_{4j}X_{j-1}) (j = 2, \ldots\ldots, j_{max} - 1) \quad (46)$$
$$X_1 = 0 \quad (47)$$
$$Y_j = (A_{1j}P_{i+1,j} + A_{2j}P_{i-1,j} - A_{5j} + A_{4j}Y_{j-1})/ \times (1 - A_{4j}X_{j-1});\ (j = 2, \ldots, j_{max}) \quad (48)$$
$$Y_1 = P_m \text{ (first boundary pressure)} \quad (49)$$
$$P_{i,j\,max} = P_n \text{ (second boundary pressure)} \quad (50)$$

The calculational procedure is as follows:

Step 1: Short-Bearing Approximation (i.e., $\partial P/\partial\theta = 0$)

(*a*) $A_{1j} = A_{2j} = 0$.
(*b*) $D_j = R^2_j/\Delta R^2$.
(*c*) Calculate X_j, Y_j, ($j = 2,3, \ldots, j_{max} - 1$); i fixed.
(*d*) Calculate P_{ij} ($j = j_{max-1}, \ldots, 3, 2$); i fixed.
(*e*) Repeat for all i's.

Step 2: First Correction

(*a*) Calculate A_{1j}, A_{2j} from equations (39) and (40).
(*b*) Use $P_{i+1,j}$ and $P_{i-1,j}$ from step 1.
(*c*) Calculate X_j, Y_j, ($j = 2, 3, \ldots, j_{max-1}$); i fixed.
(*d*) Calculate P_{ij}, [equation (45)], ($j = j_{max-1}, \ldots, 3, 2$); i fixed.
(*e*) Repeat for all i's.

Steps 3 to N: Additional Corrections

Repeat procedure under step 2, always using the $P_{i+1,j}$ and $P_{i-1,j}$-values from the previous step (or iteration).

This method is now in use on the GE 635 Seal Program and has proved to be very successful. The results obtained thus far indicate that for the full-film seal the short-bearing solution to Reynolds' equation is very close to the two-dimensional solution.

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III. F

Measurements of Leakage in Mechanical Face Seals

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The characteristics of mechanical face seals operating with hydrodynamic films were studied experimentally and analytically. Experimental and theoretical results are given showing the effects of gas cavities in the film and inward pumping. Photographs taken through the transparent test seal show the characteristics of the gas cavities. Experimental results are also given for the two fluid case, i.e., where the seal separates two different fluids; and for the case where inertia effects are important.

INTRODUCTION

Hydrodynamic effects in mechanical face seals have been shown to be of considerable importance (see Figure 1). Therefore, an extensive study of these effects was undertaken and the complete results published in (*1*) and (2).

A theoretical study of cavitation effects is given in (3). Experimental verification of that theory is given in this paper. Both experimental and theoretical results are given in (*4*) for the problem of inward pumping in mechanical face seals. The study reported in (*4*) includes both the full film-single fluid case and the single fluid

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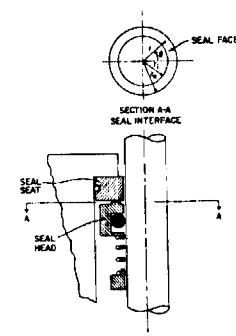

Fig. 1—Mechanical Face Seal Geometry.

NOMENCLATURE

h = film thickness, in.
$\bar{h}_i$ = average film thickness at inside radius, in.
e = displacement between seal center and shaft center, in.
ρ = coordinate measured from center of shaft, in.
ρ = density, lb/in.^{3}
r = radial coordinate measured from center of seal, in.
θ = angular coordinate, radians
U = fluid velocity, in./sec
α = phase angle between misalignment and eccentricity, radian
ω = rotational speed, radians/sec
Q = flow rate, in.^{3}/sec
p = pressure, psi
μ = dynamic viscosity, psi-sec

Dimensionless Terms

R = radius ratio = r/r_0
H = film thickness ratio = $h/\bar{h}_i$
P = pressure ratio = p/p_0
Λ = seal number = $\dfrac{6\mu\omega}{(p_i - p_0)}\left(\dfrac{r_0}{h_i}\right)^2$
ϵ = waviness parameter
ϵ_r = radial eccentricity
ϵ_t = tilt eccentricity
q = flow coefficient = $\dfrac{12\mu Q}{h_i^3(p_i - p_0)}$

Subscripts

o = at the outside radius of the seal
i = at the inside radius of the seal
c = within the gas cavity

with gas cavities case. That study is extended in this paper to give a correlation for the single fluid with gas cavities case and the full film-two fluid case.

Inertia effects in mechanical face seal hydrodynamic films are discussed theoretically in (5), (6) and (7) for concentrically rotating seals and in (8) and (9) for inward pumping seals. Experimental results confirming the theory given in (5) are presented in this paper.

All of the references given so far are part of the same study. Additional references derived from this study include a review of face seal theory (10) and thermal effects in face seals (11).

CAVITATION IN MECHANICAL FACE SEALS

A theory was derived in (3) for the case where gas cavities exist in the face seal hydrodynamic film. This theory assumed that the film thickness distribution is given by

$$H = 1 + \epsilon \cos n\theta \qquad [1]$$

the so-called "wavy seal."

Experimentally it was found in (2) that results could be obtained much more easily and in a more controlled fashion for the "misaligned seal" where the film thickness distribution is given by

$$H = 1 + R\epsilon_t \cos \theta \qquad [2]$$

The theory (3, 4) showed that for the concentrically rotating face seal the leakage rate is unaffected by the presence of gas cavities in the film. The distribution of fluid leakage around the circumference is altered but the total leakage is unchanged and, therefore, is the same as for the hydrostatic (nonrotational) case. It should be noted that these results are for a fixed geometry and since the gas cavities do alter the pressure distribution in the film they will tend to change the film thickness distribution and, therefore, the leakage rate.

For this case, then, the leakage rate is

$$q_0 = -\frac{(P_i - P_0)}{\ln R_i} 2\pi(1 + 1.5\epsilon^2) \qquad [3]$$

for the wavy seal, and

$$q_0 = -\frac{(P_i - P_0)}{\ln R_i} 2\pi(1 + 1.5R_i\epsilon_t^2) \qquad [4]$$

for the misaligned seal.

The basic problem in deriving an adequate theory for face seal behavior is the lack of accurate data. In general, this is a difficult experimental area because at the close clearances required for face seal operation, small, uncontrolled variations in surface geometry play a major role. Thus, the key to successful face seal experiments is the controlled, accurate measurement of the face seal geometry. The experiments discussed here have been

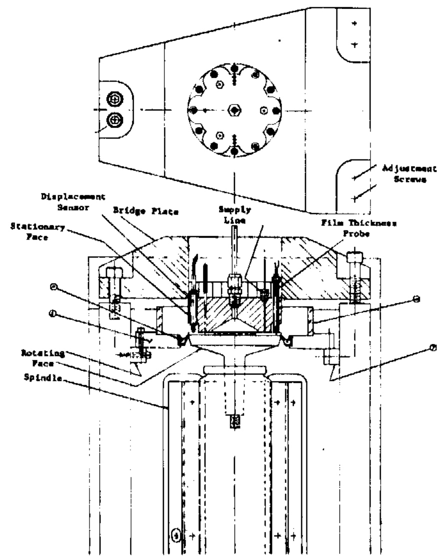

Fig. 2—Face Seal Experiment.

designed with this factor in mind. This allows accurate measurement of face seal performance characteristics, such as, leakage, normal force and power loss, as a function of known face seal geometries. Thus, the objective of obtaining data adequate for precise correlation and understanding of the physics of face seal behavior may be attained. The situation of having what appears to be comprehensive seal data, when actually the seal interface geometry is unknown, will thereby be avoided.

A cross-section of the face seal experiment is shown in Figure 2. The center feed pipe and film thickness probes are shown. The Sheffield displacement sensors were used to set the initial film thickness. Three sensors were used and from these three readings any combination of average film thickness, $\bar{h}_i$, and tilt eccentricity, ϵ_t, could be set. During the dynamic tests the Wayne-Kerr capacitance probe was used.

The transparent stationary face of the test seal is shown in Figures 3 and 4. It is made from 7740 pyrex glass optical polished plates. The glass face is mounted on lucite, a transparent, heat resistent acrylic compound. The flatness of the glass seal surface was within one sodium light band.

The rotating metal seal face is B6H3 steel which was heat treated before grinding. After grinding, the measured runout was ±0.000025 inch and the flatness was within 0.000005 inch.

In these experiments oil was fed to the inside of the seal under pressures from 1 to 40 psig and the outside diameter of the seal was exposed to air. The stationary

Fig. 3—Transparent Stationary Seal Face Mounted on the Bridge Plate.

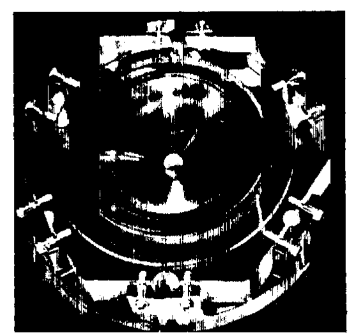

Fig. 4—Transparent Stationary Seal Face.

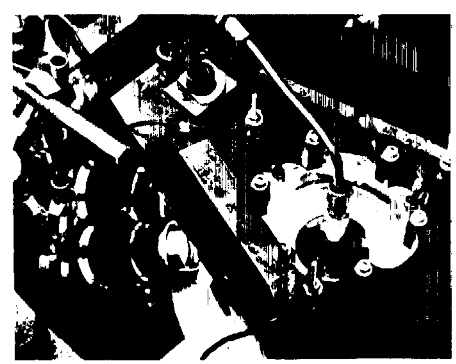

Fig. 5—Seal Face Through Transparent Mounting.

seal member was held at a fixed tilt to the rotating member so that a "converging-diverging" film shape was obtained between the seal faces. For these conditions air was sucked into the seal interface in the diverging film region forming a gas cavity at ambient pressure. Leakage measurements were obtained for conditions ranging from hydrostatic up to $\Lambda = 1000$.

The experimental results are shown correlated with equation [4] in Figures 6, 7, 8 and 9. Most of the data

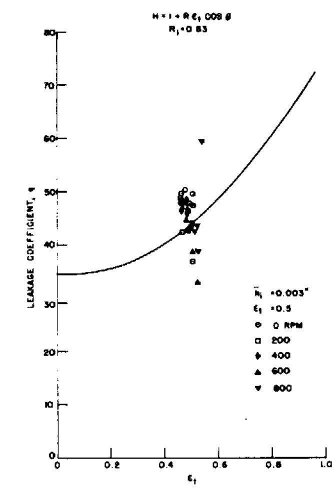

Fig. 6—Misaligned Seal Leakage.

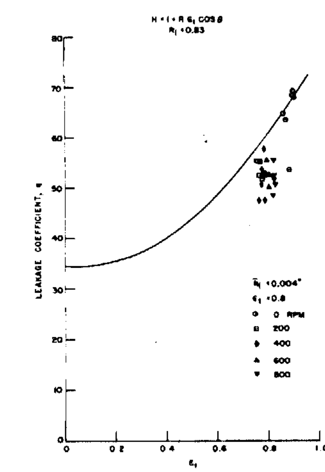

Fig. 7—Misaligned Seal Leakage.

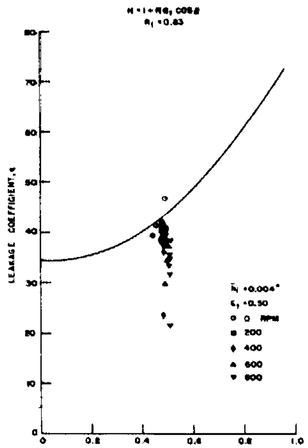

Fig. 8—Misaligned Seal Leakage.

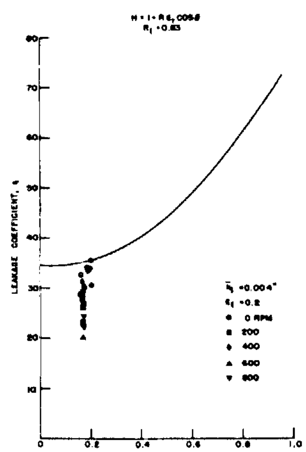

Fig. 9—Misaligned Seal Leakage.

correlated within 25% which represents an error in the film thickness measurement of only 8%. The important thing to note is that the rotational speed had no measurable effect on the leakage. Since the gas cavities, as shown in Figure 10, increased in size with speed it can be concluded that the presence and size of the gas cavities had no effect on the leakage. The pressure level did effect the accuracy of the correlation. Since the pressure level also changed the film thickness, the indication is

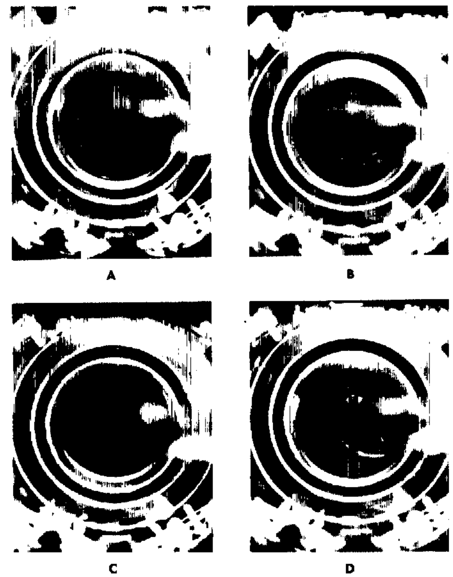

Fig. 10—Transparent Face Seal Showing Gas Cavity (A, B, and C are radially eccentric with increasing rpm. D is concentric).

that an error in film thickness measurement prevented a closer correlation.

INWARD PUMPING IN MECHANICAL FACE SEALS

Full Film—Single Fluid

This problem is discussed in detail in (4). It was found that a pumping leakage occurred in face seals due to a combination of radial eccentricity and misalignment (see Figure 11). The pumping leakage is added to the leakage given by equation [3] or [4] to give the total leakage. The pumping leakage is

$$q_{\text{pump}} = \Lambda \epsilon_r \epsilon \cos\alpha \frac{\pi(1 - R_i)}{\ln R_i} \qquad [5]$$

for the wavy seal and

$$q_{\text{pump}} = \Lambda \epsilon_r \epsilon_i \cos\alpha\, f(R_i, \epsilon_i) \qquad [6]$$

for the misaligned seal, where $f(R_i, \epsilon_i)$ is given in Figure 12 and is approximately linear with R_i and varies only slightly with ϵ_i. The analytical results given in Figure 12 were obtained using a computer solution to Reynolds equation for the misaligned seal.

For the radially eccentric seal, then, the total leakage is

$$q = q_0 + q_{\text{pump}} \qquad [7]$$

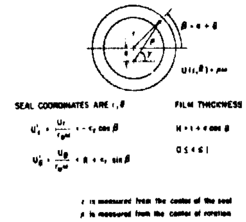

Fig. 11—Eccentric Seal Geometry.

for the full film case. It will be seen that for the seal containing gas cavities equation [7] does not apply even though q_0 given by equations [3] and [4] does apply to either the cavity case or no cavity case for the concentric seal.

In order to obtain experimental verification of equation [6] two different experiments were made. The first was with the radial eccentricity oriented in such a way as to give outward pumping which increased the hydrostatic leakage and maintained a full film. The results of this experiment are correlated in Figure 13. The second experiment was for inward pumping. In order to maintain a full film in this case it was necessary to flood the outside of the seal with oil so that no cavities could form. The results of this oil-oil experiment are correlated in Figure 14 and are shown as a linear function of Λ. The other curve is for cavity experiments and will be discussed in the section below. It can be seen that in both cases the theory predicted the experimental behavior very accurately. In the second case at low speeds the hydrostatic leakage was reduced in magnitude while at high speeds the leakage was completely reversed so that the net leakage was inward against the hydrostatic

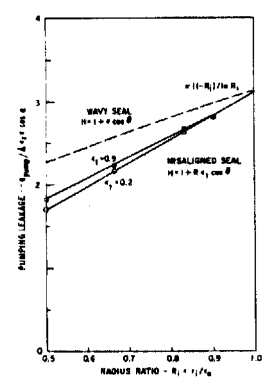

Fig. 12—Seal Pumping Leakage Due to Radial Eccentricity.

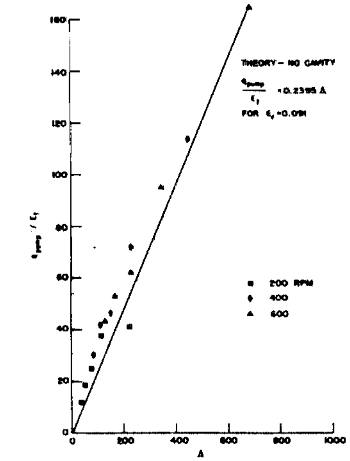

Fig. 13—Outward Pumping Due to Radial Eccentricity (No Cavity).

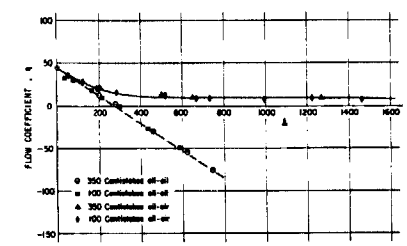

Fig. 14—Inward Pumping Experiments With and Without Cavities.

pressure drop. It is interesting to note that the pumping effect can be either inward or outward depending upon the sign of cos α and the direction of ω, the rotational velocity. Also the occurrence of a pumping leakage is dependent upon the simultaneous presence of misalignment, ϵ_i, and a radial eccentricity, ϵ_r. The term "inward pumping", therefore, is not an accurate description of the phenomenon, but the result of early experimental observations.

Single Fluid—With Cavities

When the direction of pumping was inward and the outer diameter of the seal was not flooded with oil, gas cavities appeared at the higher speeds. These cavities can be seen in Figure 10. As can be seen by the solid curve in Figure 14, the presence of cavities reduced the magnitude of the inward pumping component of leakage.

The equation and boundary conditions for this problem are discussed, but not solved, in (4). For the radially eccentric misaligned seal Reynolds equation is

$$\frac{\partial}{\partial R}\left[RH^3\frac{\partial P}{\partial R}\right]+\frac{1}{R}\frac{\partial}{\partial\theta}\left[H^3\frac{\partial P}{\partial\theta}\right] = -\Lambda\left\{R\epsilon_r\epsilon_t\cos\alpha - R^2\epsilon_t\sin\theta\right\} \quad [8]$$

with the boundary conditions $P = P_i @ R = R_i$, $P = P_0 @ R = 1$, $P = P_c @ R = R_c$. For the oil-air experimental results given in Figure 14 $P_c = P_0$. Therefore, if Λ is calculated using $(p_i - p_0)$ as the pressure term, then,

$$P = P(R, \theta, \Lambda, \epsilon_r, \epsilon_t, \cos\alpha, p_c) \quad [9]$$

and

$$q = q(\Lambda, \epsilon_r, \epsilon_t, \cos\alpha, R_i, p_c) \quad [10]$$

The results given in Figure 14 are for $p_c = p_0$, $\epsilon_r = 0.091$, $\epsilon_t = 0.57$, $\cos\alpha = 1$ (inward pumping), $R_i = 0.83$. Note that since the viscosity, μ, appears in Λ its value does not affect the correlation.

A general theoretical correlation for this case is not given. The purpose of measuring leakage for only one set of ϵ_r, ϵ_t values was to have data to check the theory developed in (4). Even though equation [8] was solved for the wavy seal case and set up for the misaligned case, numerical values have not yet been obtained for the latter. It is hoped that this can be done under an extension of the current study.

Full Film—Two Fluids

A logical extension of the seal pumping study is the case where the face seal separates two different fluids. Experimental results for this case are given in Figures 16 and 17 with the viscosity ratio μ_0/μ_i as a parameter.

It will be noted that the data showed a certain amount of scatter. This was due mainly to the fact that the two fluids leaked through the seal and mixed with each other in the reservoirs surrounding the seal. That is, at low Λ the inside fluid leaked to the outside reservoir and mixed with the other oil there. Thus, the viscosity of the oil on the outside of the seal tended to be variable. At high Λ the outer fluid was pumped into the center of the seal and mixed with the oil there modifying its viscosity. Therefore, periodically the experiments were stopped and the two reservoirs flushed before continuing. However, since each leakage measurement took a finite length of time, the mixing effect was never eliminated.

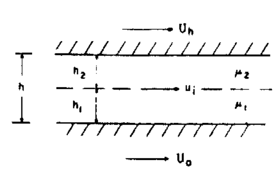

Fig. 15—Fluid Film Geometry for the Two Fluid Problem.

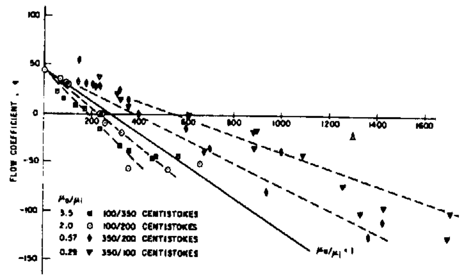

Fig. 16—Inward Pumping Experiments—Two Fluids.

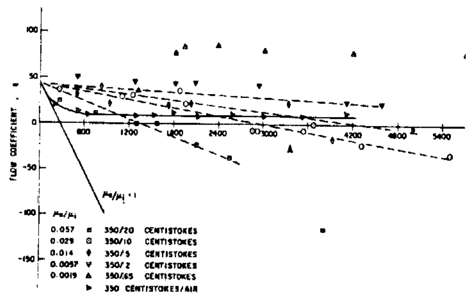

Fig. 17—Inward Pumping Experiments—Two Fluids.

Viscosity ratios, μ_0/μ_i, ranging from 3.5 to 0.002 are reported in Figures 16 and 17. Although for the single fluid case, $\mu_0/\mu_i = 1$, the data showed very little scatter, and lay on a straight line; the two fluid data did scatter and it is not certain that the correlations should be straight lines or not.

Theoretical solutions have not yet been obtained for the two fluid case, but again it is hoped that this can be accomplished in an extension of this study.

The fact that μ_0/μ_i should be used as a correlating parameter can be seen by looking at the equations for this problem which were developed in (2). Referring to Figure 15 and assuming that the two fluids exist in two distinct layers, the seal equations become

$$\frac{\partial}{\partial R}\left[RH_1^3\frac{\partial P}{\partial R}\right]+\frac{1}{R}\frac{\partial}{\partial\theta}\left[H_1^3\frac{\partial P}{\partial\theta}\right] = \Lambda_1\frac{\partial}{\partial R}\left\{RH_1\frac{\left[\frac{-3H}{\Lambda_1}\frac{\partial P}{\partial R} - \left(\frac{\mu_2}{\mu_1}\right)\frac{\epsilon_r\cos(\alpha+\theta)}{(H-H_1)}\right]}{\left[\frac{1}{H_1}+\frac{(\mu_2/\mu_1)}{H-H_1}\right]}\right\}$$

$$+\ \Lambda_1\frac{\partial}{\partial\theta}\left\{H_1\frac{\left[\frac{-3H}{\Lambda_1}\cdot\frac{1}{R}\frac{\partial P}{\partial\theta} + \left(\frac{\mu_2}{\mu_1}\right)\cdot\frac{R+\epsilon_r\sin(\alpha+\theta)}{(H-H_1)}\right]}{\left[\frac{1}{H_1}+\frac{(\mu_2/\mu_1)}{(H-H_1)}\right]}\right\} \quad [11]$$

for region 1, and

$$\frac{\partial}{\partial R}\left[R(H-H_1)^3\frac{\partial P}{\partial R}\right]+\frac{1}{R}\frac{\partial}{\partial\theta}\left[(H-H_1)^3\frac{\partial P}{\partial\theta}\right]$$

$$= \left(\frac{\mu_2}{\mu_1}\right)\Lambda_1\frac{\partial}{\partial R}\left\{R(H-H_1)\left[-\epsilon_r\cos(\alpha+\theta) + \frac{\frac{-3H}{\Lambda_1}\frac{\partial P}{\partial R} - \left(\frac{\mu_2}{\mu_1}\right)\frac{\epsilon_r\cos(\alpha+\theta)}{(H-H_1)}}{\frac{1}{H_1}+\frac{(\mu_2/\mu_1)}{(H-H_1)}}\right]\right\}$$

$$+\left(\frac{\mu_2}{\mu_1}\right)\Lambda_1\frac{\partial}{\partial\theta}\left\{(H-H_1)\left[(R+\epsilon_r\sin(\alpha+\theta)) + \frac{\frac{-3H}{\Lambda_1}\cdot\frac{1}{R}\frac{\partial P}{\partial\theta} + \left(\frac{\mu_2}{\mu_1}\right)\cdot\frac{R+\epsilon_r\sin(\alpha+\theta)}{(H-H_1)}}{\frac{1}{H_1}+\frac{(\mu_2/\mu_1)}{(H-H_1)}}\right]\right\} \quad [12]$$

for region 2. The boundary conditions are $P = P_i @ R = R_i$, $P = P_0 @ R = 1$. Solving these two equations simultaneously gives

$$P = P\left(R, \theta, \Lambda_1, \frac{\mu_2}{\mu_1}, \epsilon_r, \epsilon_t, \cos\alpha\right) \quad [13]$$

$$H_1 = H_1\left(R, \theta, \Lambda_1, \frac{\mu_2}{\mu_1}, \epsilon_r, \epsilon_t, \cos\alpha\right) \quad [14]$$

Therefore,

$$q = q\left(\Lambda_1, \frac{\mu_2}{\mu_1}, \epsilon_r, \epsilon_t, \cos\alpha, R_i\right) \quad [15]$$

INERTIA EFFECTS IN MECHANICAL FACE SEALS

This topic is considered in some detail in (5), (6), (7), (8), and (9). In (5) and (2) the concentric, misaligned seal leakage coefficient is given by

$$q = \frac{12\mu Q}{\bar{h}_i^3\left[(p_i - p_0) + \frac{3}{20}\rho\omega^2(r_0^2 - r_i^2)\right]} = -\frac{2\pi(1 + 1.5R_i\epsilon_t^2)}{\ln R_i}. \quad [16]$$

That is, the hydrostatic pressure differential is modified by the centrifugal inertia term and the theoretical result is otherwise the same as the laminar flow, full film case where inertia effects are neglected.

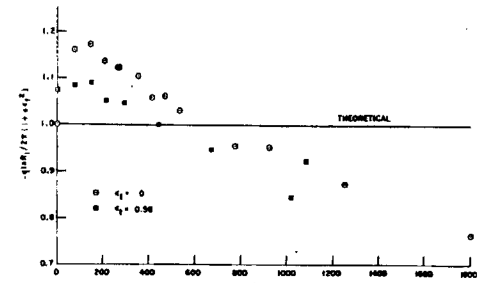

Fig. 18—Inertia Experiments.

The experiments performed up to this point have been at low speeds and with high viscosity oils. Thus, high hydrodynamic pressures could be generated while the inertia terms were minimized. In order to obtain significant inertia effects, it was necessary to run the seal experiments at high speeds. To do this, a low viscosity oil (5–10 centistokes) had to be used in order to reduce the friction and, therefore, the torque required and the heat generated. A compromise between oil viscosity and average film thickness had to be arrived at so that the leakage rates would not be too great or the film thickness measurement error too large. Values of 10 centistokes and 0.003 inches were used for the oil viscosity and average film thickness. Both aligned tests and a tilt eccentricity, ϵ_t, of 0.56 were used.

The experimental results are correlated with the theory in Figure 18. The theory was modified somewhat from that given in equation [16]. The inertia term

$$\frac{3}{20}\rho\omega^2(r_0^2 - r_i^2) \quad [17]$$

assumes that all the inertia effects take place within the seal interface and that p_i is the pressure at the inner diameter of the seal. However, it was quite obvious that because of the rotating disk, in this experiment, that the fluid in the inner reservoir was spinning around. Therefore, the pressure p_i measured at the center of rotation was not the pressure felt at the inside diameter of the seal. The final correlation showed that if an inertia term of

$$\frac{3}{20}\rho\omega^2 r_0^2 \quad [18]$$

was used, the correlation was quite good. That is, the oil in the inner reservoir experienced rotational inertia forces of the same type as the oil in the seal interface.

CONCLUSIONS

Although some scatter was found in certain of the data, there was sufficient consistency in most of the data to confirm the misaligned seal leakage theories, with and without inertia effects, with cavities and no inertia effects, and the eccentric seal pumping theory.

Since the eccentric, misaligned seal theories for pumping with cavities and with two fluids have not been completed, no verification of these theories has been made. However, sufficient data has been obtained to evaluate these effects and give a basis for substantiating the theories.

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III. G

The Effects of Geometry and Inertia on Face Seal Performance—Laminar Flow

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The "short bearing" equation of lubrication theory, modified to include inertial effects, is used to study the influence of geometric deviations from the ideal. It is found that the centrifugal force could be responsible for hydrodynamic features of the leakage flow which are theoretically unexplainable in the absence of this inertial effect. There is good evidence that the theory and the results are applicable over the entire laminar range of operation, provided the nominal clearance is small compared to the nominal radius.

Introduction

The earliest investigations of face seals were based on the assumption that face seals were very much like thrust bearings and could therefore be analyzed using lubrication theory, i.e., the Reynolds equation. That this is generally not the case has been quite convincingly demonstrated in the last decade by the work of Dowson, Snapp, and Soo [1], [2], [3],[1] all of whom have included the effects of inertia in their analyses. Their results show that inertia, specifically the centrifugal force term, can play a significant role in the performance characteristics of face seals and thrust bearings.

Dowson [9] has also verified experimentally the theoretical prediction of subambient pressures in the clearance space of a thrust bearing, the presence of which can only be explained on the basis of inertial effects.

This paper is concerned with:

(*a*) The application of the theory, including inertial effects, to those situations where the seal geometry deviates from the ideal aligned, flat surface, and an assessment of the relative effect of these deviations.

(*b*) The special hydrodynamic features of the flow in the clearance space which are directly attributable to inertial effects, e.g., separated flow and cavitation.

(*c*) An estimation of the limitations of the laminar theory, i.e., the maximum value of the Reynolds number for which the theory is valid.

Analysis of Misaligned Face Seal

Film Thickness Equation. The geometry of the seal is shown in Fig. 1(*a*). It will be convenient to adopt a coordinate system rotating with the precessional angular velocity $\bar{\Omega}$, since in this system the time-dependence at a fixed coordinate position is eliminated.

In Fig. 1(*b*) the x axis is chosen so that it always passes through the point of minimum film thickness. With this axis as a reference line for a rotating cylindrical coordinate system, the equation for the film thickness is given by

$$h = h_0 - r\cos\theta\tan\lambda \tag{1}$$

Equations of Motion. The dimensionless variables of the system will be defined by the following equations:

$$\begin{aligned} u &= u'U_0 & r &= r'R \\ v &= v'R\Omega_d & z &= z'h_0 \\ w &= w'\frac{h_0U_0}{R} & \theta &= \frac{\theta'}{\lambda} \\ & & p &= p'P_0 \end{aligned} \tag{2}$$

[1] Numbers in brackets designate References at end of paper.

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Nomenclature

a = the ratio of the waviness amplitude to the nominal clearance
c = nominal clearance
C_M = torque coefficient $= \dfrac{2T}{\rho\omega^2R_2^5/2}$
$\bar{e}_r, \bar{e}_\theta, \bar{e}_z$ = unit vectors
E_u = Euler number $= \dfrac{P_0}{\rho(R\omega)^2}$
F = total pressure force in clearance space tending to separate the seal surfaces
h = local film thickness
h_0 = disk clearance along the center line
$I_1(r,\theta) = \displaystyle\int \frac{r^2dr}{h^3}$
$I_2(r,\theta) = \displaystyle\int \frac{dr}{rh^3}$
k = an integer
p = pressure in the clearance space
P_0 = nominal pressure
P_1 = pressure at the inside radius of seal
P_2 = pressure at the outside radius of seal
P_c = cavity pressure
Q = volumetric leakage rate
r = radial coordinate
R_1 = inside radius of seal
R_2 = outside radius of seal
R_1^* = outside radius of cavity
R_2^* = inside radius of cavity
R = nominal radius
Re = Reynolds number $= \dfrac{c^2\omega}{\nu}$
T = total torque on a face seal
u = radial velocity component of fluid
U_0 = nominal radial velocity $= \dfrac{Q}{2\pi Rc}$
v = tangential velocity component of fluid
$\bar{V}$ = relative velocity vector
w = axial velocity component of fluid
z = axial coordinate
θ = tangential coordinate
λ = angle of axis misalignment
μ = viscosity of fluid
$\nu = \mu/\rho$
$\xi = 1 - \dfrac{z}{h}$
ρ = density of fluid
$\bar{\beta}$ = vector embedded in rotating disk
τ = shear stress
ψ = stream function
$\bar{\omega}$ = rotational angular velocity vector of upper disk
$\bar{\Omega}$ = precessional angular velocity of upper disk
Ω_d = characteristic angular velocity of the fluid

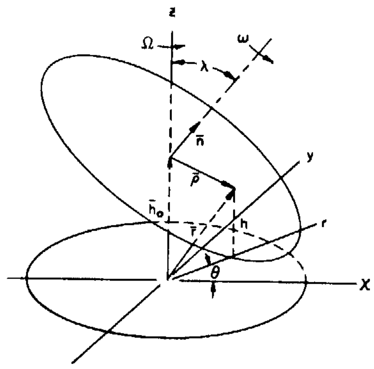

Fig. 1(a) Face seal, inertial coordinate system

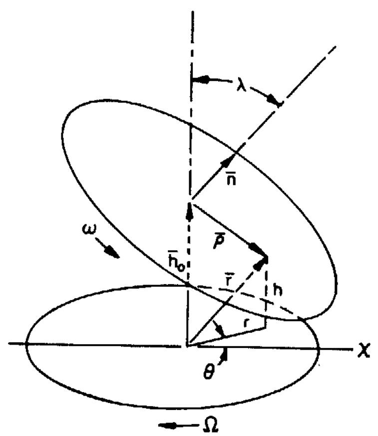

Fig. 1(b) Face seal, noninertial coordinate system

Using these variables, the continuity equation for incompressible flow can be expressed in the form

$$\frac{1}{r'}\frac{\partial(u'r')}{\partial r'} + \left(\frac{\lambda R\Omega_s}{U_s}\right)\frac{1}{r'}\frac{\partial v'}{\partial \theta'} + \frac{\partial w'}{\partial z'} = 0 \qquad (3)$$

where $\frac{\lambda R\Omega_s}{U_s}$ is of order unity for large rotational speeds and small leakage rates; i.e., $R\Omega_s \gg U_s$.

The dimensionless form of the radial momentum equation for laminar flow is

$$u'\frac{\partial u'}{\partial r'} + \left(\frac{\lambda R\,\Omega_s}{U_s}\right)\frac{v'}{r'}\frac{\partial u'}{\partial \theta'} - \left(\frac{R\Omega_s}{U_s}\right)^2\frac{v'^2}{r'}$$
$$+ w'\frac{\partial u'}{\partial z'} - r'\left(\frac{R\Omega_s}{U_s}\right)^2 - 2v'\left(\frac{R\Omega_s}{U_s}\right)^2$$
$$= -\left(\frac{P_s}{\rho U_s^2}\right)\frac{\partial p'}{\partial r'} + \frac{\nu}{RU_s}\left(\frac{R}{h_s}\right)^2\frac{\partial^2 u'}{\partial z'^2} \qquad (4)$$

The left-hand side of this equation contains the additional terms required when a noninertial reference frame is used. Under the assumption that $h_s \ll R$, the viscous forces on the right-hand side have been simplified, retaining only the most significant contributions.

For small leakage rates with

$$\frac{P_s}{\rho U_s^2} \sim \frac{\nu}{RU_s}\left(\frac{R}{h_s}\right)^2 \sim \left(\frac{R\Omega_s}{U_s}\right)^2 \gg 1 \qquad (5)$$

the radial momentum equation in the rotating coordinate system reduces to

$$-\rho\left(\frac{v^2}{r} + 2\Omega v + r\Omega^2\right) = -\frac{\partial p}{\partial r} + \mu\frac{\partial^2 u}{\partial z^2} \qquad (6)$$

The tangential momentum equation in dimensionless form is

$$u'\frac{\partial v'}{\partial r'} + \left(\frac{\lambda R\Omega_s}{U_s}\right)\frac{v'}{r'}\frac{\partial v'}{\partial \theta'} + \frac{u'v'}{r'} + w'\frac{\partial v'}{\partial z'} + 2u'$$
$$= -\left(\frac{\lambda P_s}{\rho U_s R\Omega_s}\right)\frac{1}{r'}\frac{\partial p'}{\partial \theta'} + \frac{\nu}{RU_s}\left(\frac{R}{h_s}\right)^2\frac{\partial^2 v'}{\partial z'^2} \qquad (7)$$

in which the inertial corrections and viscous force simplifications have already been made. For small leakage rates this equation reduces to

$$\frac{\partial^2 v}{\partial z^2} = 0 \qquad (8)$$

In a similar fashion, the axial momentum equation can be shown to simplify to

$$\frac{\partial p}{\partial z} = 0 \qquad (9)$$

Tangential and Radial Velocity Distributions. The surface velocity of a point on the lower disk is given by

$$V_\Omega = -r\Omega \qquad (10)$$

While the surface velocity of a point on the upper disk is given by

$$\bar{V} = \bar{\omega} \times \bar{p} = (r\omega\cos\lambda)\bar{e}_\theta + (r\omega\sin\theta\sin\lambda)\bar{e}_z \qquad (11)$$

neglecting terms of order λ^2.

Equation (8) has the solution

$$v = f(r, \theta)z + g(r, \theta) \qquad (12)$$

which, when made to satisfy the boundary conditions of equations (10) and (11) becomes

$$v = r(\omega\cos\lambda + \Omega)\frac{z}{h} - \Omega r \qquad (13)$$

When equation (13) is used in the right-hand side of equation (6), it simplifies to

$$\mu\frac{\partial^2 u}{\partial z^2} = \frac{\partial p}{\partial r} - \rho r(\omega\cos\lambda + \Omega)^2\left(\frac{z}{h}\right)^2 \qquad (14)$$

Equation (9) indicates that p is not a function of z, so that equation (14) may be integrated twice with respect to z to yield

$$u = \frac{z^2}{2\mu}\frac{\partial p}{\partial r} - \frac{\rho r z^4}{12\mu h^2}(\omega\cos\lambda + \Omega)^2 + F(r, \theta)z + G(r, \theta) \qquad (15)$$

Using the boundary conditions of equations (10) and (11) to evaluate the arbitrary functions of integration results in a radial velocity distribution of the form

$$u = \left(\frac{z^2 - hz}{2\mu}\right)\frac{\partial p}{\partial r} - \frac{\rho r}{12\mu h^2}(\omega\cos\lambda + \Omega)^2(z^4 - zh^3) \qquad (16)$$

The axial velocity distribution is not important to the analysis to follow so it is presented in Appendix A.

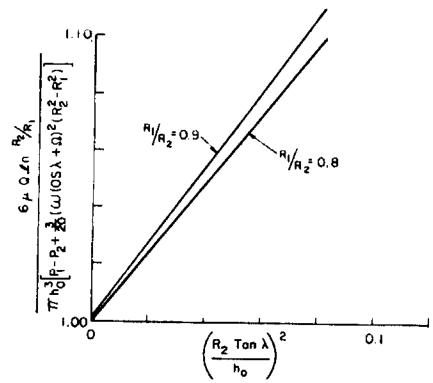

Fig. 2 Dimensionless leakage rate versus misalignment parameter

Pressure Distribution. The partial differential equation for the pressure distribution is obtained by integrating the continuity equation across the film thickness as follows:

$$\int_0^h \frac{1}{r}\frac{\partial}{\partial r}(ur)dz + \int_0^h \frac{1}{r}\frac{\partial v}{\partial \theta}dz + \int_0^h \frac{\partial w}{\partial z}dz = 0 \qquad (17)$$

With the aid of equations (13) and (16), equation (17) can be integrated to yield

$$\frac{\partial}{\partial r}\left(rh^3\frac{\partial p}{\partial r}\right) = \frac{3}{10}\rho(\omega\cos\lambda + \Omega)^2\frac{\partial(r^2h^3)}{\partial r}$$
$$+ 6\mu r(\omega\cos\lambda - \Omega)\frac{\partial h}{\partial \theta} \qquad (18)$$

This equation is essentially the short bearing equation of lubrication theory modified for inertial effects, since it allows for a tangential pressure gradient implicity, but not explicity. Nahavandi and Osterle [4] have shown the short-bearing approximation to be quite accurate in their investigation of nonpressurized thrust bearings. In their analysis they employ equation (18) using absolute angular velocities and neglecting the inertial term.

Integrating this equation twice with respect to r gives

$$p = \frac{3}{20}\rho(\omega\cos\lambda + \Omega)^2r^2 + 2\mu(\omega\cos\lambda - \Omega)\sin\theta$$
$$\cdot\tan\lambda\int\frac{r^2dr}{h^3} + \phi(\theta)\int\frac{dr}{rh^3} + \psi(\theta) \qquad (19)$$

Defining the geometry integrals by symbols

$$I_1(r, \theta) = \int\frac{r^2dr}{h^3} \qquad (20)$$

$$I_2(r, \theta) = \int\frac{dr}{rh^3} \qquad (21)$$

and employing the boundary conditions

$$p = P_1 \quad \text{at} \quad r = R_1$$
$$p = P_2 \quad \text{at} \quad r = R_2 \qquad (22)$$

the arbitrary functions of integration in equation (19) are found to be

$$\phi(\theta) = \frac{P_2 - P_1 - \left\{\frac{3}{20}\rho(\omega\cos\lambda + \Omega)^2(R_2^2 - R_1^2) + 2\mu(\omega\cos\lambda - \Omega)\tan\lambda\sin\theta[I_1(R_2, \theta) - I_1(R_1, \theta)]\right\}}{I_2(R_2, \theta) - I_2(R_1, \theta)} \qquad (23)$$

$$\psi(\theta) = P_1 - \frac{3}{20}\rho(\omega\cos\lambda + \Omega)^2R_1^2 \qquad (24)$$

Leakage Rate. The volumetric leakage rate through the seal is given by

$$Q = \int_0^{2\pi}\int_0^h urdzd\theta = -\frac{1}{12\mu}\int_0^{2\pi}\phi(\theta)d\theta \qquad (25)$$

which cannot in general be integrated in a closed form due to the complexity of $\phi(\theta)$. The tangential velocity given by equation (13) is obviously restricted to values of $\frac{r\tan\lambda}{h_o}$ which are small compared to unity. The flow rate integral equation (25) may be evaluated by expanding in powers of $\frac{r\tan\lambda}{h_o}$ with the result

$$Q = \left(\frac{\pi h_o^3}{6\mu\ln R_2/R_1}\right)\left[P_1 - P_2 + \frac{3}{20}\rho(\omega\cos\lambda + \Omega)^2(R_2^2 - R_1^2)\right]$$
$$\cdot\left\{1 + \frac{1}{2}\left\{\left[\frac{3(R_2 - R_1)}{\ln R_2/R_1}\right]^2 - \left[\frac{3(R_2^2 - R_1^2)}{\ln R_2/R_1}\right]\right\}\left(\frac{\tan\lambda}{h_o}\right)^2\right.$$
$$+ \frac{3}{8}\left\{\left[\frac{3(R_2 - R_1)}{\ln R_2/R_1}\right]^4 - 3\left[\frac{3(R_2 - R_1)}{\ln R_2/R_1}\right]^2\left[\frac{3(R_2^2 - R_1^2)}{\ln R_2/R_1}\right]\right.$$
$$+ 2\left[\frac{3(R_2 - R_1)}{\ln R_2/R_1}\right]\left[\frac{\frac{10}{3}(R_2^3 - R_1^3)}{\ln R_2/R_1}\right] + \left[\frac{3(R_2^2 - R_1^2)}{\ln R_2/R_1}\right]^2$$
$$\left.\left. - \left[\frac{\frac{15}{4}(R_2^4 - R_1^4)}{\ln R_2/R_1}\right]\right\}\left(\frac{\tan\lambda}{h_o}\right)^4 + \ldots\right\} \qquad (26)$$

Fig. 2 shows the effect of misalignment on the leakage rate. Since the expansion given in equation (26) can be rearranged to contain powers of the quantity $\left(\frac{R_2\tan\lambda}{h_o}\right)^2$ it is convenient to plot the expression in the braces as a function of this parameter. Geometric considerations require $\left(\frac{R_2\tan\lambda}{h_o}\right)$ to be less than unity if the two surfaces are to avoid contact with one another. When there is no misalignment the dimensionless leakage rate, plotted as the ordinate in Fig. 2, is unity. Fig. 2 shows that the increase in leakage rate with misalignment is proportional to $\left(\frac{R_2\tan\lambda}{h_o}\right)^2$.

Equation (25) computes only the net flow rate through the seal. Examination of equation (16), however, shows that due to inertial effects, the axial variation of the radial velocity at a given r and θ is determined by a fourth-order polynomial in z. Because of the boundary conditions imposed on u, equation (16) has roots at $z = 0$, and 1.0. Dividing out these known factors of the polynomial and setting $u = 0$ yields the following formula for the remaining roots

$$\left(\frac{z}{h}\right)^2 + \left(\frac{z}{h}\right) + 1 = \frac{6\frac{\partial p}{\partial r}}{\rho r(\omega\cos\lambda + \Omega)^2} \qquad (27)$$

Equation (27) has a positive root in the range of $0 < \frac{z}{h} < 1.0$ when

$$\frac{1}{6} < \frac{\dfrac{\partial p}{\partial r}}{\rho r(\omega \cos \lambda + \Omega)^2} < \frac{1}{2} \tag{28}$$

Regions of the flow field where relation (28) is satisfied by the local flow conditions are regions where there is an inflow along the nonrotating surface and outflow along the rotating surface. The existence of such regions is a direct consequence of inertial effects and points out the importance of including these effects in the analysis.

Torque. The equation for the torque is

$$T = \int_0^{2\pi}\int_{R_1}^{R_2} \tau_{z\theta} r^2 dr d\theta = \int_0^{2\pi}\int_{R_1}^{R_2} \mu\left[\frac{r}{h}(\omega \cos\lambda + \Omega) + \omega\cos\theta\sin\lambda\right] r^2 dr d\theta = \frac{\pi\mu(\omega\cos\lambda + \Omega)(R_2^4 - R_1^4)}{2h_0}$$
$$\times\left[1 + \frac{1}{3}\left(\frac{R_2^6 - R_1^6}{R_2^4 - R_1^4}\right)\left(\frac{\tan\lambda}{h_0}\right)^2 + \frac{3}{16}\left(\frac{R_2^8 - R_1^8}{R_2^4 - R_1^4}\right)\left(\frac{\tan\lambda}{h_0}\right)^4 + \ldots\right] \tag{29}$$

where an expansion in powers of $\frac{r \tan\lambda}{h_0}$ has been used. Once again it is convenient to rearrange the torque equation to contain powers of the parameter $\left(\frac{R_2 \tan\lambda}{h_0}\right)^2$ and plot the dimensionless torque as a function of that parameter. Fig. 3 shows that the deviation of the torque from its aligned value is, like the leakage rate, proportional to $\left(\frac{R_2\tan\lambda}{h_0}\right)^2$. It should also be noted that the torque is less sensitive to misalignment than is the leakage rate.

Load Capacity. The pressure force tending to separate the two faces of the seal is given by

$$F = \int_{R_1}^{R_2}\int_0^{2\pi} pr dr d\theta = \pi P_1(R_2^2 - R_1^2) + \frac{\pi}{2}\left(\frac{3\rho}{20}\right)(\omega\cos\lambda + \Omega)^2(R_2^2 - R_1^2)^2$$
$$+ 2\pi\left[\frac{P_2 - P_1 - \frac{3\rho}{20}(\omega\cos\lambda + \Omega)^2(R_2^2 - R_1^2)}{\ln R_2/R_1}\right]\times\left[\frac{R_2^2 \ln R_2/R_1}{2} - \left(\frac{R_2^2 - R_1^2}{4}\right)\right]$$
$$\times\left\{1 + \left[\frac{1}{R_2^2\ln R_2/R_1 - \left(\frac{R_2^2 - R_1^2}{2}\right)}\right]\left\{\frac{3}{4}(R_2^2 - R_1^2)^2 - \frac{3(R_2 - R_1)}{\ln R_2/R_1}\left[(R_2^2 - R_1^2) - \frac{3}{2}R_1(R_2^2 - R_1^2)\right]\right.\right.$$
$$\left.\left. + \left[\left(\frac{3(R_2 - R_1)}{\ln R_2/R_1}\right)^2 - 3\left(\frac{R_2^2 - R_1^2}{\ln R_2/R_1}\right)\right]\left[\frac{R_2^2\ln R_2/R_1}{2} - \left(\frac{R_2^2 - R_1^2}{4}\right)\right]\right\}\left(\frac{\tan\lambda}{h_0}\right)^2 + \ldots\right\} \tag{30}$$

having again expanded the integral in powers of $\frac{r\tan\lambda}{h_0}$. It should be noted that F is not the net force exerted on the seal disk. That force would be obtained by subtracting the externally applied forces.

A numerical evaluation of the effect of misalignment using

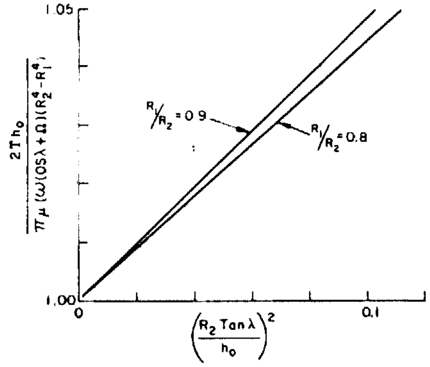

Fig. 3 Dimensionless torque versus misalignment parameter

$R_1/R_2 = 0.8$, 0.9, and $\left(\frac{R_2\tan\lambda}{h_0}\right)^2 < 0.1$ shows that misalignment decreases this force by less than one percent.

Conclusions. The effects of misalignment on leakage flow, torque, and separating force are in the same sequence of importance as the effects of a change in clearance in an aligned face seal. That is, the leakage is greatly affected by misalignment, the torque is only moderately affected by misalignment, and the separating force practically unaltered by misalignment.

Aligned Face Seal

Film Thickness Varying Radially. The analysis presented previously shows the relative importance of the small unavoidable misalignments which are very likely to occur in an actual face seal application.

For the aligned seal where the film thickness varies in the radial direction due to thermal or elastohydrodynamic distortion, the leakage rate is given by

$$\frac{6\mu Q}{\pi}\int_{R_1}^{R_2}\frac{dr}{rh^3} = P_1 - P_2 + \frac{3}{20}\rho(R_2\omega)^2\left[1 - \left(\frac{R_1}{R_2}\right)^2\right] \tag{31}$$

Snapp [2] has investigated the effects of various film shapes which depend only on the radius. To include the effects of inertia in such cases one need only replace the true pressure difference with the effective pressure difference which is the right-hand side of equation (31).

The load carrying capacity for such a seal is computed from the formula

$$\frac{F}{\pi(R_2^2 - R_1^2)} + \frac{12\mu Q}{\pi(R_2^2 - R_1^2)}\int_{R_1}^{R_2}\left[\int_{R_1}^{r}\frac{d\xi}{\xi h^3}\right] r dr = P_1 + \frac{3}{40}\rho(R_2\omega)^2\left[1 - \left(\frac{R_1}{R_2}\right)^2\right] \tag{32}$$

where it is apparent that the effect of inertia on the separating force can be introduced into an analysis which neglects them by considering the effective supply pressure to be the right-hand side of equation (32) and computing Q as suggested previously.

The torque is unaffected by inertia so that no correction is required for analyses which neglect it.

Regions of reverse flow can be expected in seals of this type when

$$-\frac{6}{5} < \frac{36\mu Q}{\rho(r\omega)^2 h^3 \pi} < \frac{4}{5} \tag{33}$$

which is in agreement with Soo's results [3] for h = constant.

The equation for the boundary separating the inflow and outflow regions is

$$\frac{z}{h} = \frac{1}{2}\left[\sqrt{1 + 4\left(\frac{4}{5} - \frac{36\mu Q}{\rho(r\omega)^2 h^3 \pi}\right)} - 1\right] \tag{34}$$

Film Thickness Varying Tangentially. There is good experimental evidence that face seals which are originally flat do not remain so after reasonably long periods of operation. Nau [5] observed a two-cycle tangential waviness whose amplitude was of the order of 50 microin. after only 6 hr of operation.

In this case, it will be assumed that the film thickness is given by

$$h = c(1 + a\cos k\theta) \tag{35}$$

and that the tangential velocity is given by

$$v = r\omega\left(\frac{z}{h}\right) \tag{36}$$

The partial differential equation for the pressure distribution is

$$h^3\frac{\partial}{\partial r}\left(r\frac{\partial p}{\partial r}\right) = \frac{3}{5}\rho r\omega^2 h^2 - 6\mu r\omega\frac{\partial h}{\partial\theta} \tag{37}$$

which when solved yields

$$p = \frac{3}{20}\rho r^2\omega^2 - \frac{3\mu r^2\omega}{2h^3}\frac{\partial h}{\partial\theta} + f(\theta)\ln r + g(\theta) \tag{38}$$

The boundary conditions of equations (22) are satisfied when the arbitrary functions of integration are chosen to be

$$f(\theta) = \frac{1}{\ln R_2/R_1}\left[P_1 - P_2 + \frac{3\mu\omega}{2h^3}\frac{\partial h}{\partial\theta}(R_1^2 - R_2^2) - \frac{3}{20}\rho\omega^2(R_1^2 - R_2^2)\right] \tag{39}$$

$$g(\theta) = P_2 - \frac{3}{20}\rho(R_2\omega)^2 + \frac{3\mu R_2^2\omega}{2h^3}\frac{\partial h}{\partial\theta} - f(\theta)\ln R_2 \tag{40}$$

The pressure distribution now becomes

$$p - P_2 = \frac{3}{20}\rho\omega^2(r^2 - R_2^2) - \frac{3\mu\omega}{2h^3}\frac{\partial h}{\partial\theta}(r^2 - R_2^2) + \frac{\ln r/R_2}{\ln R_1/R_2}\left[P_1 - P_2 + \frac{3\mu\omega}{2h^3}\frac{\partial h}{\partial\theta}(R_1^2 - R_2^2) - \frac{3}{20}\rho\omega^2(R_1^2 - R_2^2)\right] \tag{41}$$

Of primary interest, of course, is the leakage rate. Because of the periodicity of h the flow rate is given by

$$Q = -\frac{1}{12\mu}\int_0^{2\pi} h^3 f(\theta) d\theta = \frac{\pi c^3}{6\mu\ln R_2/R_1}\left[P_1 - P_2 + \frac{3}{20}\rho\omega^2(R_2^2 - R_1^2)\right]\left(1 + \frac{3}{2}a^2\right) \tag{42}$$

The term $1 + \frac{3}{2}a^2$ reflects the effect of the radial pumping of the waviness which is independent of the frequency k. To stay within the limitations of the theory, the dimensionless amplitude a should be restricted to value of the order of 0.3 or less.

The torque for this geometry is found to be

$$T = \frac{\pi\mu\omega(R_2^4 - R_1^4)}{2c}\left[1 + \frac{a^2}{2}\right] \tag{43}$$

where terms of order a^4 have been neglected. Here, as in the misaligned case, the variations in geometry affect the torque less severely than leakage flow.

As might be expected, due to the periodicity of the film thickness, the load capacity is the same as that for a perfectly flat, aligned face seal in the absence of cavitation.

The boundaries of subambient pressure regions within the fluid film, when they exist, are determined by setting $p = P_2$ in equation (41) and solving for r as a function of θ. The resulting equation is

$$\frac{1 - \left(\frac{r}{R_2}\right)^2}{\left(\frac{\ln r/R_2}{\ln R_1/R_2}\right)} = [1 - (R_1/R_2)^2] - \left\{\frac{P_1 - P_2}{\frac{3}{20}\rho(R_2\omega)^2\left[\left(\frac{10\nu}{\omega h^2}\right)\frac{\partial h}{\partial\theta} - 1\right]}\right\} \tag{44}$$

where it is required that $R_1 < r < R_2$.

Whether or not such regions exist within the film is obviously very much dependent on the term containing $P_1 - P_2$. The importance of this term can be easily demonstrated by observing that if $P_1 = P_2$, $r = R_1$ is a solution of the equation and the entire film is subambient. The significance of such regions will be discussed in the following section on cavitation.

Cavitation

Separation Cavitation. In the previous analyses, it was noted that under proper conditions a reverse flow along the rotating disk would occur. This is possible only when the outer perimeter of the seal is immersed in the sealing fluid. When the outer perimeter is exposed to a gas, such as air, it is quite possible that the film will separate from the stationary surface and cling to the rotating surface forming a free surface within the seal. Cavitation of this type, called "separation cavitation," has been investigated by Birkhoff and Hays [6] in journal bearings in the absence of inertia.

At the inception of a separated cavity, the leakage-flow rate would probably increase sharply, since the effective width of the seal would be reduced. This problem cannot be treated with any of the results so far presented since they do not satisfy the free-surface boundary conditions.

True Cavitation A second type of cavitation has been proposed which requires a region of subambient pressure, rather than a separation of the flow from the stationary disk. In the regions of subambient pressure, it is assumed that dissolved gases in the fluid (mostly air) come out of solution and eventually rupture the film. Unlike separation cavitation, "true cavitation" produces a cavity which extends completely across the clearance space. Nau [5] suggests that this cavity pressure should be very close to the pressure at which the liquid was saturated with the dissolved gases.

Since the appearance and coalescence of gas bubbles cannot cause the liquid to violate the conservation of mass, one would expect to see streamers of liquid spiraling outward through the constant pressure cavity, possibly rejoining to form a continuous film near the outer edge of the seal.

The formation of a true cavity region with its accompanying streamers is an extremely difficult problem in the case of the misaligned seal, but it can be analyzed in the case of the aligned seal because symmetry considerations permit one to assume that the cavity will be a circular annulus.

Nahavandi and Osterle [4] have investigated the misaligned case assuming that the inner and outer radii are at atmosphere pressure. In regions of the seal where subambient pressures were calculated they set the pressure equal to ambient with the result that the predicted load capacity was greatly improved. They made no attempt, however, to satisfy conservation of mass within

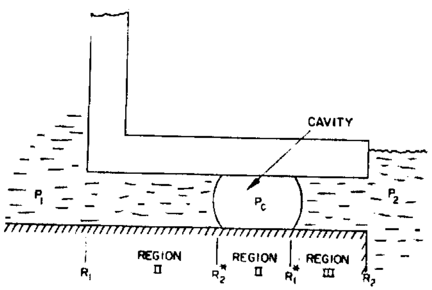

Fig. 4 True cavitation

the remaining liquid regions so that the degree of improvement may not be as large as predicted. Also, seals do not generally operate without an applied pressure differential across the boundaries. The application of a boundary pressure which is above ambient could easily eradicate potential subambient regions and eliminate cavitation, a possibility which seems to have been overlooked in some recent experimental investigations of seal performance.

Since the analysis of the aligned, constant clearance seal is somewhat more instructive it will be presented here, while the results for a seal with a radially varying film thickness seal will be given in Appendix B. Fig. 4 shows a constant clearance seal with inner and outer regions of liquid separated by a true cavity. In this model it will be assumed that surface tension effects at the liquid-cavity interface are negligible [7] while the liquid streamers are pumped through the constant pressure cavity by viscous and inertial forces. The leakage rate in either region I or III is given by the equation

$$Q = \frac{\pi r c^3}{6\mu}\left[\frac{3\rho r\omega^2}{10} - \frac{dp}{dr}\right]$$

The conditions on the boundary of a true cavity are not known at the present time. For enclosed cavities it is often assumed that the pressure gradient in the fluid normal to the cavity boundary vanishes on the leading edge of the cavity (the Swift-Stieber condition). That this assumption is an oversimplification has been demonstrated by Dowson [8] and others who have shown experimentally that the cavity forms somewhat downstream of the predicted boundary position following a subcavity pressure loop. If it is assumed, in the absence of a precise criteria, that the Swift-Stieber criteria is a reasonable approximation [7], this condition together with the constant cavity pressure condition give the inner radius of the cavity as

$$(R_2^*)^2 = \frac{20\mu Q}{\rho\omega^2\pi c^3} < (R_2)^2 \tag{45}$$

The flow rate arriving at the cavity is

$$Q = \frac{\pi c^3}{6\mu \ln R_2^*/R_1}\left[P_1 - P_c + \frac{3\rho\omega^2}{20}(R_2^{*2} - R_1^2)\right] \tag{46}$$

Eliminating Q between equations (45) and (46) yields

$$P_1 - P_c = \frac{3}{10}\rho(R_2^*\omega)^2\left\{\ln R_2^*/R_1 - \frac{1}{2}\left[1 - \left(\frac{R_1}{R_2^*}\right)^2\right]\right\} \tag{47}$$

The streamers will reunite after crossing region II when equation (44) is once again satisfied, i.e.,

$$Q = \frac{\rho(R_2^*\omega)^2\pi c^3}{20\mu} = \frac{\pi c^3}{6\mu \ln R_2^*/R_1}$$

$$\times\left[P_c - P_2 + \frac{3\rho\omega^2}{20}(R_2^2 - R_1^{*2})\right] \tag{48 (Cont.)}$$

Solving this equation for R_1^*/R_2 gives

$$\left(\frac{R_1^*}{R_2}\right)^2 = \frac{1}{2}\left[\frac{1-(R_1^*/R_2)^2}{\ln R_2/R_1^*}\right] - \frac{10}{3}\left(\frac{P_2 - P_c}{\rho(R_2\omega)^2}\right)\left[\frac{1}{\ln (R_2/R_1^*)}\right] \tag{49}$$

Equations (47) and (49) can be solved for R_2^* and R_1^* where, for a cavity to exist, it is required that $R_1 < R_2^* < R_1^* < R_2$. If it is assumed that $P_c = P_2$, equation (49) shows that R_2^*/R_1^* is always greater than unity, approaching unity only when R_1^*/R_2 approaches unity from values of R_1^*/R_2 less than unity. In this case, the film would not be expected to reform having once ruptured. If on the other hand, it is assumed that $P_c < P_2$, very small values of $\frac{10}{3}\left(\frac{P_2 - P_c}{\rho(R_2\omega)^2}\right)\left[\frac{1}{\ln R_2/R_1^*}\right]$ would rectify this situation for R_1^* less than but nearly equal to R_2.

Inspection of equation (44) shows that the leakage rate with a cavity is less than predicted by the full film solution. The critical quantity in the foregoing analysis is, of course, the cavity pressure, P_c, a quantity which can only be estimated at the present time.

Limitations of the Theory

All of the analysis presented so far is based on the presumption that

$$\frac{P_0}{\rho U_0^2} \sim \frac{\nu}{RU_0}\left(\frac{R}{h_0}\right)^2 \sim \left(\frac{R\Omega_0}{U_0}\right)^2 \gg 1 \tag{5}$$

The experiments of Dowson [9] show that for flat aligned seals (with $\Omega = 0$) equations (6), (8), and (9) are accurate when

$$\frac{(R\omega/U_0)^2}{P_0/\rho U_0^2} < 10$$

but these experiments give no indication of any Reynolds number limitation. The following investigation will be directed toward establishing more precisely this limitation for flat, aligned face seals.

The Stream Function. It was shown earlier in equation (16) that the radial component of velocity takes the form

$$u = f(\xi)\frac{h^2}{\mu}\frac{\partial p}{\partial r} + \frac{\rho r\omega^2 h^2}{\mu}g(\xi) \tag{50}$$

where $\xi = 1 - \frac{z}{h}$. Combining equations (44) and (50) and eliminating the pressure gradient for a constant clearance seal ($h = c$) gives

$$u = F(\xi)\frac{Q}{rc} + r\omega\,\mathrm{Re}\,G(\xi) \tag{51}$$

where $\mathrm{Re} = \frac{c^2\omega}{\nu}$ is the Reynolds number

Following a scheme used by Soo [3] for small leakage rates, a stream function will be defined by

$$\psi = r\omega\,\mathrm{Re}\,cW(\xi) + \frac{Q}{2\pi r}U(\xi) \tag{52}$$

so that

$$u = -\frac{\partial\psi}{\partial z} = -r\omega\,\mathrm{Re}\,W' - \frac{Q}{2\pi rc}U' \tag{53}$$

$$w = \frac{1}{r}\frac{\partial(r\psi)}{\partial r} = 2c\omega\,\mathrm{Re}\,W \tag{54}$$

where the prime denotes differentiation with respect to ξ.

In addition, it will be assumed that

$$v = r\omega V(\xi) \tag{55}$$

Reduction of Momentum Equations. Taking into account the smallness of the clearance compared to the nominal radius, the linear momentum equations reduce to

$$u\frac{\partial u}{\partial r} + w\frac{\partial u}{\partial z} - \frac{v^2}{r} = -\frac{1}{\rho}\frac{\partial p}{\partial r} + \nu\frac{\partial^2 u}{\partial z^2} \tag{56}$$

$$\frac{u}{r}\frac{\partial(rv)}{\partial r} + w\frac{\partial v}{\partial z} = \nu\frac{\partial^2 v}{\partial z^2} \tag{57}$$

$$\frac{\partial p}{\partial z} = 0 \tag{58}$$

Eliminating the pressure gradient by partially differentiating equation (56) with respect to z and introducing, u, v, and w from equations (53), (54), and (55) allows equation (20) to be rewritten in the form

$$2\,\mathrm{Re}\left(\frac{U_0}{r\omega}\right)^2 U'U'' + 2\,\mathrm{Re}^2\,WW''' + 2\,\mathrm{Re}^2\left(\frac{U_0}{r\omega}\right)(WU'')'$$

$$+ 2\,\mathrm{Re}\,VV' = \mathrm{Re}\,W^{\mathrm{IV}} + \left(\frac{U_0}{r\omega}\right)U^{\mathrm{IV}} \tag{59}$$

Neglecting the term of order $(U_0/r\omega)^2$ and separating the remaining terms into two groups, depending on whether or not they contain the coefficient $U_0/r\omega$, yields two equations

$$U^{\mathrm{IV}} - 2\,\mathrm{Re}^2(WU'')' = 0 \tag{60}$$

$$W^{\mathrm{IV}} - 2\,\mathrm{Re}^2\,WW''' - 2VV' = 0 \tag{61}$$

Using the same procedure equation (57) reduces to

$$V'' + 2\,\mathrm{Re}^2\,VW' - 2\,\mathrm{Re}^2\,WV' = 0 \tag{62}$$

neglecting a term of order $U_0/r\omega$.

Perturbation Solutions. The solutions of equations (60), (61), and (62) will yield equations for U, V, and W containing the parameter Re^2. These solutions represent a higher-order approximation for u, v, and w which will contain the effects of Reynolds number. The form of these equations suggests a perturbation solution of the type

$$U = U_1 + \mathrm{Re}^2 U_2 \tag{63}$$

$$V = V_1 + \mathrm{Re}^2 V_2 \tag{64}$$

$$W = W_1 + \mathrm{Re}^2 W_2 \tag{65}$$

The boundary conditions to be satisfied by U_1, V_1, and W_1 are

$$\begin{aligned} U_1(0) - U(1) &= 1\\ U_1'(0) &= U_1'(1) = 0\\ V_1(0) &= 1,\ V_1(1) = 0\\ W_1(0) &= W_1(1) = 0\\ W_1'(0) &= W_1'(1) = 0 \end{aligned} \tag{66}$$

which in turn guarantee that

$$\begin{aligned} u(0) &= u(c) = 0\\ v(0) &= r\omega,\ v(c) = 0\\ w(0) &= w(c) = 0 \end{aligned} \tag{67}$$

The solutions of equations (60), (61), and (62), (with $\mathrm{Re}^2 = 0$) for U_1, V_1, and W_1 are found to be

$$U_1 = 2\xi^3 - 3\xi^2$$

$$V_1 = 1 - \xi \tag{68}$$

Table 1

ξ	u_1'	u_2'	v_1	v_2	w_1	w_2	w_1'	w_2'
0	0	0	1.0	0	0	0	0	0
0.2	-0.96	-0.000204	0.8	-0.000680	-0.00114	-0.0000 00096	-0.00851	-0.0000 0021
0.4	-1.44	-0.000126	0.6	-0.000791	-0.00290	0.0000 00051	-0.00320	0.0000 0132
0.6	-1.44	+0.000111	0.4	-0.000554	+0.00230	0.0000 00321	0.00480	0.0000 0041
0.8	-0.96	+0.000196	0.2	-0.00076	[illegible]	0.0000 00234	0.00147	-0.0000 0140
1.0	0	0	0	0	0	0	0	0

$$W_1 = -\frac{1}{20}\xi^2 + \frac{7}{60}\xi^3 - \frac{1}{12}\xi^4 + \frac{1}{60}\xi^5 \tag{68 (Cont.)}$$

It is easily verified, with the aid of the transformation $\xi = 1 - \frac{z}{h}$, that these solutions are consistant with the results of Dowson [1].

The perturbation functions are determined from the equations

$$\begin{aligned} U_2^{\mathrm{IV}} &= 2(W_1U_1''' + W_1'U_1'')\\ V_2'' &= 2(W_1V_1' - W_1'V_1)\\ W_2^{\mathrm{IV}} &= 2(V_1V_1' + W_1W_1''') \end{aligned} \tag{69}$$

The solution to these equations must satisfy the conditions of equation (66) with the exception that

$$\begin{aligned} U_2(0) - U_2(1) &= 0\\ U_2(0) &= 0 \end{aligned} \tag{70}$$

The solutions satisfying these conditions are given by

$$U_2' = \frac{1}{20}\xi^4 - \frac{39}{300}\xi^5 + \frac{76}{600}\xi^6 - \frac{11}{210}\xi^7 + \frac{4}{560}\xi^8$$

$$-2.285(10^{-4})\xi^2 - 1.200(10^{-4})\xi \tag{71}$$

$$V_2 = \frac{1}{30}\xi^3 - \frac{1}{15}\xi^4 + \frac{34}{600}\xi^5 - \frac{2}{90}\xi^6 + \frac{2}{630}\xi^7 - \frac{9}{2100}\xi \tag{72}$$

Because of their complexity W_2 and W_2' are given in Appendix C.

Table 1 shows the importance of the perturbation terms as they affect u, v, and w in equations (53), (54), and (55). From these comparisons it is apparent that $\mathrm{Re} \leq 10$ will not cause any significant change in the velocities computed from the earlier simplified analysis.

Pressure Distribution. The equation for the pressure gradient which includes the perturbation terms takes the form

$$\frac{dp}{dr} = \rho r\omega^2\left[\frac{3}{10} + (2W_1W_1'' - W_1'W_1')\mathrm{Re}^2\right]$$

$$- \frac{Q\mu}{\pi rc^3}[6 - \mathrm{Re}^2(U_1''W_1)] \tag{73}$$

Table 2 shows the numerical values of the coefficients of Re^2. Clearly the assumption that the pressure is independent of z is invalidated in the higher-order approximation but this assumption is still reasonably good so long as $\mathrm{Re} \leq 10$.

Torque. Since the literature contains analytical and experimental data on torque it would be instructive to compare the results of the perturbation analysis with them.

The shear stresses acting on the rotating and stationary surfaces are given by

$$\tau_{z\theta}(0) = -\mu\left(\frac{r\omega}{c}\right)[1 + 0.004286\,\mathrm{Re}^2] \tag{74}$$

$$\tau_{z\theta}(1) = -\mu\left(\frac{r\omega}{c}\right)[1 - 0.001269\,\mathrm{Re}^2] \tag{75}$$

Table 2

ξ	$2v_1v_1'' - v_1'v_1'$	$v_1''v_1$
0	0	0
0.2	-0.000008	0.004320
0.4	-0.000213	-0.003005
0.6	-0.000153	-0.002674
0.8	-0.000018	-0.004379
1.0	0	0

Calculating the moment coefficient due to twice the moment on a rotating disk with no central hole yields

$$C_M = \frac{2T}{\rho\omega^2 R_2^5/2} = 2\pi\left(\frac{\nu}{R_2^2\omega}\right)\left(\frac{R_2}{c}\right)[1 + 0.004286\ \mathrm{Re}^2]$$

$$= \frac{2\pi}{\left(\frac{R_2^2\omega}{\nu}\right)\left(\frac{c}{R_2}\right)} + 0.01346\left[2\left(\frac{R_2^2\omega}{\nu}\right)\left(\frac{c}{R_2}\right)^2\right] \quad (76)$$

This equation compares favorably with the analytical results of Schultz-Grunow [10],

$$C_M = \frac{2\pi}{\left(\frac{R_2^2\omega}{\nu}\right)\left(\frac{c}{R_2}\right)} + 2\left(\frac{R_2^2\omega}{\nu}\right)\left(\frac{c}{R}\right)^2\left[0.01461 + 0.1256\left(\frac{c}{R}\right)^2\right] \quad (77)$$

when c/R is small.

Equation (76) also compares favorably with the experimental results for small c/R given by Schlichting [11] up to Reynolds numbers of the order 10.

Thus there is good evidence that the perturbation technique is on a firm physical foundation.

Conclusion. In addition to the qualitative order-of-magnitude relationships between the significant dimensionless parameters of equation (5), we are now on a position to assert quantitative restrictions on these numbers for which the analysis will be valid.

Defining the Euler number as $\mathrm{Eu} = \frac{P_0}{\rho(R\omega)^2}$, we have the limitation that

$$\mathrm{Re} \sim \frac{1}{\mathrm{Eu}} < 10 \quad (78)$$

It should be noted that $\mathrm{Re} < 10$ corresponds to $\frac{R^2\omega}{\nu} \sim 10^4$ for seals with small c/R. Since $\frac{R^2\omega}{\nu} \sim 10^4$ is very near to the threshold of turbulence, it is expected that the simplified analysis given earlier will be sufficiently accurate to cover almost the entire laminar range of operation.

Conclusions

1 Misalignment and surface waviness are found to affect face seal performance in the same order that clearance changes affect the performance of flat surface, aligned seals. Variations in surface flatness of the magnitude which are likely to occur during manufacture or as a result of thermal distortion can result in increases in leakage rate of the order of 10 percent with accompanying increases in the torque of about five percent. Misalignments of the order which would be difficult to detect in an actual seal application could result in similar increases in the leakage and torque. Waviness and misalignment are found to have a negligible effect on the separating pressure force. These conclusions are, of course, only applicable in the absence of any cavitation.

2 Regions of subambient pressure and reversed flow are found to occur in the clearance space under certain conditions. These regions, which are a direct consequence of inertial effects, are potentially regions of true or separation cavitation. Since very small leakage rates and hence small pressure differences are required to satisfy these conditions, cavitation might well be a relatively rare occurrence in practice.

3 The basic theory used in the analysis seems capable of covering the entire range laminar face seal operation with reasonable accuracy, provided the clearance to radius ratio is small.

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APPENDIX A

If instead of integrating the continuity equation all the way across the fluid film as in equation (17), the integration is carried to a point z within the film, then the equation for w becomes

$$w = \left\{\frac{\rho}{6\mu}(\omega\cos\lambda + \Omega)^2\left[\frac{1}{r}\frac{\partial}{dr}\left(\frac{r}{h}\right)^2\right]\right\}z^4 - \left\{\frac{1}{6\mu}\left[\frac{1}{r}\frac{\partial}{dr}\left(r\frac{\partial p}{dr}\right)\right]\right\}z^3 + \left\{\frac{1}{4\mu}\left[\frac{h}{r}\frac{\partial}{\partial r}\left(r\frac{\partial p}{\partial r}\right) + \frac{\partial p}{\partial r}\frac{\partial h}{\partial r}\right] + \frac{1}{2}(\omega\cos\lambda + \Omega)\left[\frac{1}{h^2}\frac{\partial h}{\partial \theta}\right] - \frac{\rho}{24\mu}(\omega\cos\lambda + \Omega)^2 \times \left[\frac{1}{r}\frac{\partial}{\partial r}(r^2 h)\right]\right\}z^2$$

Using equation (1), this equation becomes

$$w = \left\{\frac{\rho}{30\mu}(\omega\cos\lambda + \Omega)^2\left(\frac{1}{h^2}\right)\left(1 + \frac{r}{h}\cos\theta\tan\lambda\right)\right\}z^4$$
$$- \left\{\frac{\rho}{10\mu}(\omega\cos\lambda + \Omega)^2 + (\omega\cos\lambda - \Omega)\sin\theta\tan\lambda \times \left(\frac{r}{h^2}\right)\left(1 + \frac{r}{h}\cos\theta\tan\lambda\right) + \frac{\phi(\theta)}{2\mu}\left(\frac{1}{rh^4}\right)\cos\theta\tan\lambda\right\}z^3$$
$$+ \left\{\frac{3}{20}\left(\frac{\rho}{\mu}\right)(\omega\cos\lambda + \Omega)^2 h + \frac{3}{2}(\omega\cos\lambda - \Omega)\sin\theta\tan\lambda \times \left(\frac{r}{h^2}\right)\left(1 + \frac{r}{h}\cos\theta\tan\lambda\right) + \frac{3}{4}\frac{\phi(\theta)}{\mu}\cos\theta\tan\lambda\left(\frac{1}{rh^3}\right)\right.$$
$$- \frac{3}{40}\left(\frac{\rho}{\mu}\right)(\omega\cos\lambda + \Omega)^2 r\cos\theta\tan\lambda$$
$$- \frac{1}{2}(\omega\cos\lambda - \Omega)\sin\theta\cos\theta\tan^2\lambda\left(\frac{r^2}{h^2}\right)$$
$$- \frac{\phi(\theta)}{4\mu}\cos\theta\tan\lambda\left(\frac{1}{rh^2}\right) + \frac{1}{2}(\omega\cos\lambda + \Omega)\sin\theta\tan\lambda\left(\frac{r}{h^2}\right)$$
$$\left. - \frac{1}{24}\left(\frac{\rho}{\mu}\right)(\omega\cos\lambda + \Omega)^2(2h - r\cos\tan\lambda)\right\}z^2$$

APPENDIX B

Subambient pressures in a full-film aligned seal will occur when $\frac{dp}{dr} = 0$ within the film. Applying this condition to equation (44) yields the criteria

$$\frac{36\,\mu Q}{\rho\pi(r\omega)^2h^3} = \frac{9}{5}$$

where h can be a function of radius, in which case Q is computed from equation (31). If this condition is satisfied at some radius within the seal, the eventual formation of a true cavity should be expected and the calculations amended appropriately.

When the film thickness varies in the radial direction, R_2^* must be chosen so as to satisfy the equation

$$P_1 - P_a = \frac{9}{5}\rho(R_2^*\omega)^2\left\{h^2(R_2^*)\int_{R_1}^{R_2^*}\frac{dr}{rh^3} - \frac{1}{2}\left[1 - \left(\frac{R_1}{R_2^*}\right)^2\right]\right\}$$

where $h\,(R_2^*)$ is the film thickness at the radial distance R_2^* and the integral is the same as the one used in equation (31).

Having satisfied this equation with the proper choice of R_2^*, $R_1^* > R_2^*$ must now be chosen to satisfy the equation

$$\left(\frac{R_1^*}{R_2}\right)^2 = \frac{\frac{1}{2}\left[1 - \left(\frac{R_1^*}{R_2}\right)^2\right] - \frac{10}{3}\left(\frac{P_2 - P_a}{\rho(R_2\omega)^2}\right)}{h^2(R_2^*)\int_{R_1^*}^{R_2}\frac{dr}{rh^3}}$$

The leakage rate is then given by

$$Q = \frac{\pi\rho(R_2^*\omega)^2h^3(R_2^*)}{20\mu}$$

APPENDIX C

The perturbation function W_1 is given by

$$\frac{W_1}{2} = 0.66712980(10^{-4})\frac{\xi^3}{6} - 0.6662400(10^{-4})\frac{\xi^2}{2}$$
$$+ 0.15306122(10^{-4})\xi^5 - 0.97222222(10^{-4})\xi^6$$
$$+ 0.21558957(10^{-4})\xi^7 - 0.20252267(10^{-4})\xi^8$$
$$+ 0.98173343(10^{-4})\xi^9 - 0.26102294(10^{-4})\xi^{10}$$
$$+ 0.62432717(10^{-4})\xi^{11} - 0.34231201(10^{-4})\xi^{12} \quad (65)$$
$$+ 0.18606393(10^{-4})\xi^{13} - 0.67348876(10^{-4})\xi^{14}$$
$$+ 0.15633984(10^{-4})\xi^{15} - 0.20996052(10^{-7})\xi^{16}$$
$$+ 0.12350619(10^{-4})\xi^{17} \quad (65\ Cont.)$$

The perturbation function W_1' is given by

$$\frac{W_1'}{2} = 0.66712980(10^{-4})\frac{\xi^2}{2} - 0.6662400(10^{-4})\xi$$
$$+ 0.76530610(10^{-4})\xi^4 - 0.58333333(10^{-4})\xi^5$$
$$+ 0.15091300(10^{-4})\xi^6 - 0.16201814(10^{-4})\xi^7$$
$$+ 0.88356009(10^{-4})\xi^8 - 0.26102294(10^{-4})\xi^9$$
$$+ 0.68675988(10^{-4})\xi^{10} - 0.41077441(10^{-4})\xi^{11}$$
$$+ 0.24188311(10^{-4})\xi^{12} - 0.94288426(10^{-4})\xi^{13}$$
$$+ 0.23450976(10^{-4})\xi^{14} - 0.33593683(10^{-4})\xi^{15}$$
$$+ 0.20996052(10^{-7})\xi^{16}$$

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III. H

The Effects of Geometry and Inertia on Face Seal Performance—Turbulent Flow

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The "short bearing" equation of lubrication theory, modified to include the inertial effects, is used to study the influence of geometric deviations from the ideal. The turbulent nature of the flow is described by an isotropic apparent viscosity and a power-law velocity distribution. It is found that geometric deviations from the ideal are less influential than in laminar flow.

Introduction

In a previous paper [1][1] it was shown that inertial effects influence the performance characteristics of face seals. The inclusion of these effects in the analysis was found necessary if some of the observed anomalous behavior of seals is to be explained theoretically. That investigation was, however, restricted to the laminar range of operation. This paper extends the theoretical approach into the fully turbulent range, reexamining the effects of geometry on face seal performance characteristics at high rotational Reynolds numbers.

Analysis of Misaligned Face Seal

Film Thickness Equation and Seal Surface Velocities. The geometry of the problem is shown in Fig. 1(*a*). It will be convenient to adopt a coordinate system rotating with the precessional angular velocity $\bar{\Omega}$, since in this system the time-dependence at a fixed coordinate position is eliminated.

In Fig. 1(*b*) the x axis is chosen so that it always passes through the point of minimum thickness. With this axis as a reference line for the cylindrical coordinate system, the equation for the film thickness is given by

$$h = h_0 - r \cos\theta \tan\lambda \tag{1}$$

The surface velocity of a point on the lower disk is given by

$$V_\Omega = -r\Omega \tag{2}$$

while the velocity of a point on the upper disk is given by

$$V_\omega = \bar{\omega} \times \bar{p} \cong (r\omega \cos\lambda)\bar{e}_\theta + (r\omega \sin\theta \sin\lambda)\bar{e}_z \tag{3}$$

neglecting terms of the order λ^2.

Equations of Motion. In the previous analysis of misaligned face seal with laminar flow [1] it was shown that the momentum equations in the rotating coordinate system reduce to

$$-\rho\left(\frac{v^2}{r} + 2\Omega v + r\Omega^2\right) = -\frac{\partial p}{\partial r} + \frac{\partial \tau_{rz}}{\partial z} \tag{4}$$

$$\frac{\partial \tau_{z\theta}}{\partial z} = 0 \tag{5}$$

$$\frac{\partial p}{\partial z} = 0 \tag{6}$$

under the conditions

$$\frac{P_0}{\rho U_0^2} \sim \frac{\nu}{RU_0}\left(\frac{R}{h_0}\right)^2 \sim \left(\frac{R\Omega_0}{U_0}\right) \gg 1 \tag{7}$$

While it is true that the inertia effects are considerably larger in turbulent flow than in laminar flow as indicated by the relatively large Reynolds numbers associated with turbulent flow, it also is known that the shear stresses are several orders of magnitude larger in turbulent flow than in laminar. In the light of the compensating effects it does not appear to be necessary to include those higher-order inertial terms which were included in the investigation of high Reynolds number laminar flow [1].

[1] Numbers in brackets designate References at end of paper.

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Nomenclature

a = ratio of the waviness amplitude to the nominal clearance
c = nominal clearance
C_f = coefficient of friction for shear flow
C_M = torque coefficient = $\dfrac{2T}{\rho\omega^2 R_2^5/2}$
$\bar{e}_\theta, \bar{e}_z$ = unit vectors
f = leakage coefficient of friction
F = total force in clearance space tending to separate the seal surfaces
h = local film thickness
h_0 = disk clearance along the center line
$I_1(r,\theta) = \displaystyle\int \frac{r^{11/4}dr}{(h/2)^{9/4}}$
$I_2(r,\theta) = \displaystyle\int \frac{dr}{r^{1/4}(h/2)^{9/4}}$
k = an integer
l = the mixing length
L = axial length of shaft seal
n = power law index
$N = \left(\dfrac{1}{1+\frac{1}{n}} - 1\right)\left(\dfrac{1}{2+\frac{1}{n}}\right) + \left(\dfrac{1}{1+\frac{1}{n}}\right)$
p = pressure in clearance space
P_0 = nominal pressure
P_1 = pressure at inside radius of the seal
P_2 = pressure at outside radius of the seal
P_c = cavity pressure
Q = volumetric leakage rate
r = radial coordinate
R = nominal radius
R_1 = inside radius of seal
R_2 = outside radius of seal
R_1^* = outside radius of cavity
R_2^* = inside radius of cavity
$R_e = \dfrac{U_0 c}{\nu}$
$R_\omega = \dfrac{Vc}{\nu}$
T = total torque on seal face
u = temporal mean radial velocity component
u_r = u relative to seal surface
U_0 = nominal face seal leakage velocity = $\dfrac{Q}{2\pi Rc}$
v = temporal mean tangential velocity component

(Continued on next page)

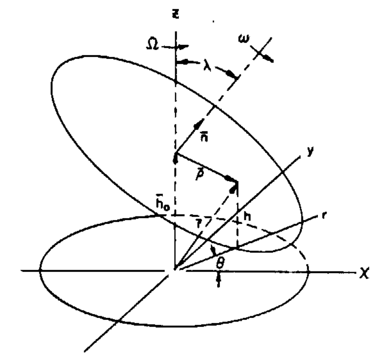

Fig. 1(a) Face seal, inertial coordinate system

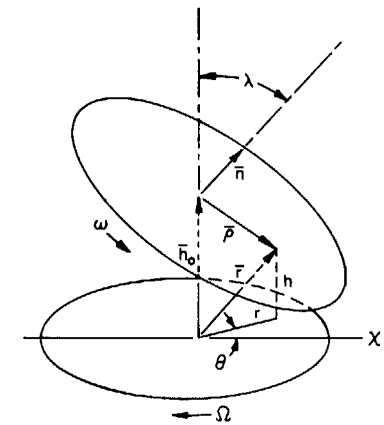

Fig. 1(b) Face seal, noninertial coordinate system

Turbulent Shear Stress at the Seal Surface. It was demonstrated that [1] the foregoing equations would accurately predict face seal performance throughout the range of laminar operation if the Stokesian constitutive equation was employed. No such general constitutive equation for turbulent flow is presently available. One of the most successful phenomenological formulations for the turbulent shear stress has been the "mixing length" theory proposed by L. Prandtl. For three-dimensional flows this formulation has been generalized [2] as follows

$$\tau_{rz} = (\rho l^2\Phi + \mu)\left(\frac{\partial w}{\partial r} + \frac{\partial u}{\partial z}\right) \tag{8}$$

$$\tau_{z\theta} = (\rho l^2\Phi + \mu)\left(\frac{\partial v}{\partial z} + \frac{1}{r}\frac{\partial w}{\partial \theta}\right) \tag{9}$$

where l is the mixing length and Φ is defined by

$$\Phi^2 = 2\left[\left(\frac{\partial u}{\partial r}\right)^2 + \left(\frac{1}{r}\frac{\partial v}{\partial \theta} + \frac{u}{r}\right)^2 + \left(\frac{\partial w}{\partial z}\right)^2\right] + \left[r\frac{\partial}{\partial r}\left(\frac{v}{r}\right) + \frac{1}{r}\frac{\partial u}{\partial \theta}\right]^2 + \left[\frac{1}{r}\frac{\partial w}{\partial \theta} + \frac{\partial v}{\partial z}\right]^2 + \left[\frac{\partial u}{\partial z} + \frac{\partial w}{\partial r}\right]^2 \tag{10}$$

An order of magnitude investigation of these equations, assuming that the nominal leakage velocity is small compared to the nominal tangential velocity, shows that to a good approximation for incompressible flow

$$\tau_{rz} \cong \left[\rho l^2\left|\frac{\partial v}{\partial z}\right| + \mu\right]\frac{\partial u}{\partial z} \tag{11}$$

$$\tau_{z\theta} \cong \left[\rho l^2\left|\frac{\partial v}{\partial z}\right| + \mu\right]\frac{\partial v}{\partial z} \tag{12}$$

These equations indicate that the apparent viscosities $\rho l^2\Phi + \mu$ for both radial and tangential flow are isotropic as in laminar flow. Contrary to laminar flow the apparent viscosity is not constant throughout the flow field, but varies from point to point with changes in tangential velocity gradient.

A study of the turbulent velocity profiles in both shear and pressure flows by Burton [2] has shown that except very near the wall the velocity distribution is quite accurately represented by the "1/7 power-law" approximation to the universal velocity profile

$$\frac{v}{V^*} = 8.74\left(\frac{V^*z}{\nu}\right)^{1/7} \tag{13}$$

The wall shear stress computed from the 1/7 power law, generalized for three-dimensional flows, is given by

$$\tau_w = 0.0225\rho\left(\frac{\nu}{z}\right)^{1/4}(u_r^2 + v_r^2)^{3/8} \tag{14}$$

where u_r and v_r are the velocity components relative to the wall. The radial and tangential shear components are given by

$$\tau_{z\theta_w} = \left[\frac{v_r}{(u_r^2 + v_r^2)^{1/2}}\right]\tau_w \tag{15}$$

$$\tau_{rz_w} = \left[\frac{u_r}{(u_r^2 + v_r^2)^{1/2}}\right]\tau_w \tag{16}$$

Assuming that the tangential velocity, which is the result of almost pure shear flow follows the 1/7 power law to the middle of

Nomenclature

v_r = v relative to seal surface
V = surface velocity of shaft
$V^* = \sqrt{\dfrac{\tau_w}{\rho}}$
w = temporal mean axial velocity component
w_r = w relative to seal surface
W_0 = nominal shaft seal leakage velocity = $\dfrac{Q}{2\pi Rc}$
x = tangential coordinate in shaft seal
z = axial coordinate
θ = tangential coordinate in face seal
λ = angle of axis misalignment
μ = viscosity of fluid
ν = μ/ρ
ρ = density of fluid
$\bar{p}$ = vector imbedded in the rotating disk
τ = shear stress
τ_w = shear stress at seal surface
$\bar{\omega}$ = rotational angular velocity of upper disk
$\bar{\Omega}$ = precessional angular velocity of upper disk
Ω_0 = characteristic angular velocity of fluid

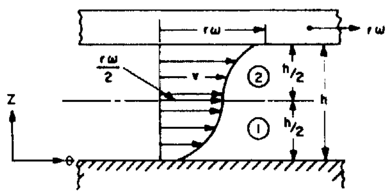

Fig. 2 Tangential velocity profile

the clearance space leads to an equation for tangential shear stress at the wall of the form

$$\tau_{s\theta_w} = 0.0225\rho(\nu)^{1/4}\left[\frac{\left(\frac{r\omega\cos\lambda + r\Omega}{2}\right)^7}{h/2}\right]^{1/4}\left[1+\left(\frac{u_r}{v_r}\right)^2\right]^{1/2} \quad (17)$$

where it is also assumed that the mid-clearance velocity of the fluid relative to the surfaces is $\frac{r}{2}$ $(\omega\cos\lambda + \Omega)$.

The radial velocity would also be expected to follow, at least approximately, the $^1/_7$ power law so that the relative velocity ratio could be estimated from

$$u_r/v_r \sim U_0/R\Omega_0$$

Since the analysis is based on the assumption that $\frac{U_0}{R\Omega_0} \ll 1$, equation (17) may be simplified, with small sacrifice in accuracy, to read

$$\tau_{s\theta_w} = 0.0225\rho(\nu)^{1/4}\left[\frac{\left(\frac{r\omega\cos\lambda + r\Omega}{2}\right)^7}{h/2}\right]^{1/4} \quad (18)$$

Tangential Velocity Distribution. The $^1/_7$ power law and the wall shear stress equation (14) are consistent with the premise that the tangential shear stress is very nearly independent of the coordinate z as indicated by equation (5). With this in mind, the clearance space will be divided into two regions as shown in Fig. 2 and a $^1/_7$ power law tangential velocity distribution will be assumed in each region. When these two distributions are joined at mid-clearance with the common velocity $\frac{r}{2}$ $(\omega\cos\lambda + r\Omega)$, they become

$$v_1 = \frac{r}{2}(\omega\cos\lambda + \Omega)\left(\frac{z}{h/2}\right)^{1/7} - r\Omega \quad (19)$$

for $0 \leq z \leq h/2$, and

$$v_2 = r\omega\cos\lambda - \frac{r}{2}(\omega\cos\lambda + \Omega)\left(\frac{h-z}{h/2}\right)^{1/7} \quad (20)$$

for $h/2 \leq z \leq h$.

The Radial Velocity Distribution. Equation (6) indicates that the pressure is independent of z so that equation (4) may be partially integrated with respect to z to give

$$\tau_{sr} = f(r,\theta) + \frac{\partial p}{\partial r}z - \rho\int\left(\frac{v^2}{r} + r\Omega^2 + 2\Omega v\right)dz \quad (21)$$

The isotropy of the apparent viscosity in equations (11) and (12) allows them to be written as

$$\tau_{sr} = \tau_{s\theta}\begin{bmatrix}\frac{\partial u}{\partial z}\\ \frac{\partial v}{\partial z}\end{bmatrix} \quad (22)$$

Table 1

$\left(\frac{z}{h/2}\right)$	$F\left(\frac{z}{h/2}\right)$	$G\left(\frac{z}{h/2}\right)$
0.0	0.000	0.000
0.1	0.711	0.697
0.2	0.776	0.765
0.4	0.833	0.830
0.6	0.860	0.866
0.8	0.873	0.886
1.0	0.875	0.894
1.2	0.873	0.896
1.4	0.860	0.893
1.6	0.833	0.873
1.8	0.776	0.823
1.9	0.711	0.762
2.0	0.000	0.000

Recalling that $\tau_{s\theta} = \tau_{s\theta_w}$ the momentum equation may now be rewritten as

$$\tau_{s\theta_w}\frac{\partial u}{\partial z} = \frac{\partial v}{\partial z}\left[f(r,\theta) + \frac{\partial p}{\partial r}z - \rho\int\left(\frac{v^2}{r} + r\Omega^2 + 2\Omega v\right)dz\right] \quad (23)$$

This equation may now be integrated with respect to z to give the radial velocity distribution

$$u\tau_{s\theta_w} = v\left[f(r,\theta) + \frac{\partial p}{\partial r}z - \rho\int\left(\frac{v^2}{r} + r\Omega^2 + 2v\Omega\right)dz\right] - \int v\left[\frac{\partial p}{\partial r} - \rho\left(\frac{v^2}{r} + r\Omega^2 + 2\Omega v\right)\right]dz + g(r,\theta) \quad (24)$$

where the radial velocity must satisfy the conditions $u(r, \theta, 0) = u(r, \theta, h) = 0$.

Equation (24) is the general formulation for the radial velocity distribution in a face seal whether the flow is laminar or turbulent. All that is required is the proper selection of $\tau_{s\theta_w}$ and $v(r, \theta, z)$. For turbulent flow equations (18), (19), and (20) will be chosen to fulfill these requirements. The arbitrary functions of integration can be adjusted so that the boundary conditions are satisfied, as well as continuity of u and $\frac{\partial u}{\partial z}$ at $z = \frac{h}{2}$. The radial velocity then takes the form

$$u\tau_{s\theta_w} = -\left(\frac{r\omega\cos\lambda + r\Omega}{2}\right)\left(\frac{h}{2}\right)\frac{\partial p}{\partial r}F\left(\frac{z}{h/2}\right) + \frac{\rho}{r}\left(\frac{r\omega\cos\lambda + r\Omega}{2}\right)^2\left(\frac{h}{2}\right)G\left(\frac{z}{h/2}\right) \quad (25)$$

where the functions F and G are given by

$$F\left(\frac{z}{h/2}\right) = \left(\frac{z}{h/2}\right)^{1/7}\left[1 - \frac{1}{8}\left(\frac{z}{h/2}\right)\right], 0 \leq z \leq h/2$$
$$= \left(\frac{h-z}{h/2}\right)^{1/7}\left[\frac{3}{4} + \frac{1}{8}\left(\frac{z}{h/2}\right)\right], h/2 \leq z \leq h \quad (26)$$

$$G\left(\frac{z}{h/2}\right) = \left(\frac{z}{h/2}\right)^{1/7}\left[\frac{35}{36} - \frac{7}{90}\left(\frac{z}{h/2}\right)^{9/7}\right], 0 \leq z \leq h/2$$
$$= \left(\frac{h-z}{h/2}\right)^{1/7}\left[\frac{1}{12} + \frac{1}{2}\left(\frac{z}{h/2}\right) + \frac{7}{18}\left(\frac{h-z}{h/2}\right)^{8/7} - \frac{7}{90}\left(\frac{h-z}{h/2}\right)^{9/7}\right], h/2 \leq z \leq h$$

Table 1 illustrates the axial variation in the pressure gradient and centrifugal force contributions to the radial velocity distribution.

The pressure gradient coefficient is, as expected, symmetric with respect to the mid-clearance following closely the $^1/_7$ power law from the seal surfaces outward. The centrifugal force coefficient is slightly unsymmetric, as in the laminar case, following roughly the $^1/_7$ power law outward from the seal surfaces. It will be recalled that the simplification of equation (17) was made on the presumption that the axial variation of the radial velocity profile would closely approximate a $^1/_7$ power distribution. Turbulence introduced by the tangential velocity through the rotation of the seal surfaces has produced a typically turbulent axial variation in the radial velocity distribution.

Pressure Distribution. The pressure distribution is obtained from the integrated form of the continuity equation

$$\int_0^h\frac{1}{r}\frac{\partial}{\partial r}(ur)dz + \int_0^h\frac{1}{r}\frac{\partial v}{\partial\theta}dz + \int_0^h\frac{\partial w}{\partial z}dz = 0 \quad (27)$$

which becomes

$$\frac{1}{r}\frac{\partial}{\partial r}\int_0^h(ur)dz + \frac{1}{r}\int_0^h\frac{\partial v}{\partial\theta}dz = 0 \quad (28)$$

Carrying out the indicated integrations using equations (19), (20), and (26) gives

$$\frac{\partial}{\partial r}\left[r^{1/4}\left(\frac{h}{2}\right)^{5/4}\frac{\partial p}{\partial r}\right] = \frac{835}{816}\rho\left(\frac{\omega\cos\lambda + \Omega}{2}\right)^2 \times\frac{\partial}{\partial r}\left[r^{5/4}\left(\frac{h}{2}\right)^{5/4}\right] + \frac{30}{49}\left[0.0225(\nu)^{1/4}\rho r\left(\frac{\omega\cos\lambda - \Omega}{2}\right)\times\left(\frac{\omega\cos\lambda + \Omega}{2}\right)^{3/4}\right]r\sin\theta\tan\lambda \quad (29)$$

This equation is analogous to the short bearing equation of lubrication theory modified to include the effects of inertia.

Integrating twice with respect to the r, the equation for the radial distribution of pressure is

$$p = \frac{835}{816}\left(\frac{\rho}{2}\right)\left(\frac{\omega\cos\lambda + \Omega}{2}\right)^2 r^2 + \frac{30}{49}\left[0.0225(\nu)^{1/4}\rho \times\left(\frac{\omega\cos\lambda - \Omega}{2}\right)\left(\frac{\omega\cos + \Omega}{2}\right)^{3/4}\right]\left(\frac{\sin\theta\tan\lambda}{3}\right) \times\int\frac{r^{11/4}dr}{(h/2)^{5/4}} + \phi(\theta)\int\frac{dr}{r^{1/4}(h/2)^{5/4}} + \psi(\theta) \quad (30)$$

The boundary conditions to be satisfied by the pressure are

$$p = P_1 \quad \text{at} \quad r = R_1$$
$$p = P_2 \quad \text{at} \quad r = R_2 \quad (31)$$

Defining the integrals

$$I_1(r,\theta) = \int\frac{r^{11/4}dr}{(h/2)^{5/4}} \quad (32)$$

$$I_2(r,\theta) = \int\frac{dr}{r^{1/4}(h/2)^{5/4}} \quad (33)$$

the pressure boundary conditions are satisfied when the functions of integration are chosen to be

$$\phi(\theta) = \left[\frac{1}{I_2(R_2,\theta) - I_2(R_1,\theta)}\right]\left\{P_2 - P_1 - \frac{835}{816}\left(\frac{\rho}{2}\right)\times\left(\frac{\omega\cos\lambda + \Omega}{2}\right)^2(R_2^2 - R_1^2) + \frac{10}{49}(0.0225\nu^{1/4}\rho)\times\left(\frac{\omega\cos\lambda - \Omega}{2}\right)\left(\frac{\omega\cos\lambda + \Omega}{2}\right)^{3/4}\tan\lambda\sin\theta \times[I_1(R_2,\theta) - I_1(R_1,\theta)]\right\} \quad (34)$$

$$\psi(\theta) = P_1 - \frac{835}{816}\left(\frac{\rho}{2}\right)\left(\frac{\omega\cos\lambda + \Omega}{2}\right)^2 R_1^2 \quad (35)$$

Solving for the zeros of equation (26), having first eliminated the zeros at $z = 0$ and $z = h$ reveals that the radial velocity will vanish within the clearance space when

$$\frac{35}{144} \leq \frac{\frac{\partial p}{\partial r}}{\rho r(\omega\cos\lambda + \Omega)^2} \leq \frac{13}{48} \quad (36)$$

When this condition is satisfied a region of outward flow will exist adjacent to the rotating disk and a region of inward flow will exist adjacent to the stationary disk. This region of reversed flow is directly attributable to inertial effects and not predicted by analyses, such as used in lubrication theory, which neglect inertia.

Leakage Rate. The volumetric leakage rate is calculated from the equation

$$Q = \int_0^{2\pi}\int_0^h urdzd\theta = -\frac{\frac{49}{30}\int_0^{2\pi}\phi(\theta)d\theta}{0.0225\rho(\nu)^{1/4}\left(\frac{\omega\cos\lambda + \Omega}{2}\right)^{3/4}} \quad (37)$$

The integral in this formula cannot be represented by a closed-form algebraic expression due to the complexity of $\phi(\theta)$.

Since the governing equation (29) for the pressure distribution does not explicitly contain a tangential pressure gradient term, the theory is inherently restricted to small misalignments. A measure of the relative misalignment is the parameter $\frac{r\tan\lambda}{h_0}$, which must be restricted in this theory to values which are small compared to unity. If the integrand in equation (37) is expanded in powers of this parameter, the flow rate may be expressed as

$$Q = \left(\frac{\pi}{0.0195}\right)\left[\frac{h_0^5}{\rho^3\mu(\omega\cos\lambda + \Omega)^3}\right]^{1/4} \times\left[\frac{P_1 - P_2 + \left(\frac{835}{816}\right)\left(\frac{\rho}{2}\right)\left(\frac{\omega\cos\lambda + \Omega}{2}\right)^2(R_2^2 - R_1^2)}{4/3(R_2^{3/4} - R_1^{3/4})}\right]$$
$$\times\left(1 + \frac{1}{2}\left\{\left[\frac{27}{28}\left(\frac{1-\left(\frac{R_1}{R_2}\right)^{7/4}}{1-\left(\frac{R_1}{R_2}\right)^{3/4}}\right)\right]^2 - \frac{27}{32}\left(\frac{13}{11}\right)\times\left[\frac{1-\left(\frac{R_1}{R_2}\right)^{11/4}}{1-\left(\frac{R_1}{R_2}\right)^{3/4}}\right]\right\}\left(\frac{R_2\tan\lambda}{h_0}\right)^2 + \frac{3}{8}\right.$$
$$\times\left\{\left[\frac{27}{28}\left(\frac{1-\left(\frac{R_1}{R_2}\right)^{7/4}}{1-\left(\frac{R_1}{R_2}\right)^{3/4}}\right)\right]^4 - 3\left(\frac{27}{28}\right)\frac{13}{11}\left[\frac{1-\left(\frac{R_1}{R_2}\right)^{11/4}}{1-\left(\frac{R_1}{R_2}\right)^{3/4}}\right]\times\left[\frac{27}{28}\left(\frac{1-\left(\frac{R_1}{R_2}\right)^{7/4}}{1-\left(\frac{R_1}{R_2}\right)^{3/4}}\right)\right]^2 + 2\left(\frac{27}{28}\right)\frac{39}{40}\left(\frac{17}{16}\right)\right.$$
$$\times\left[\frac{1-\left(\frac{R_1}{R_2}\right)^{7/4}}{1-\left(\frac{R_1}{R_2}\right)^{3/4}}\right]\left[\frac{1-\left(\frac{R_1}{R_2}\right)^{15/4}}{1-\left(\frac{R_1}{R_2}\right)^{3/4}}\right] + \left[\frac{27}{28}\left(\frac{13}{11}\right)\times\left(\frac{1-\left(\frac{R_1}{R_2}\right)^{11/4}}{1-\left(\frac{R_1}{R_2}\right)^{3/4}}\right)\right]^2 - \left(\frac{9}{19}\right)\left(\frac{13}{16}\right)\left(\frac{17}{32}\right)\left(\frac{21}{4}\right) \quad (38)$$

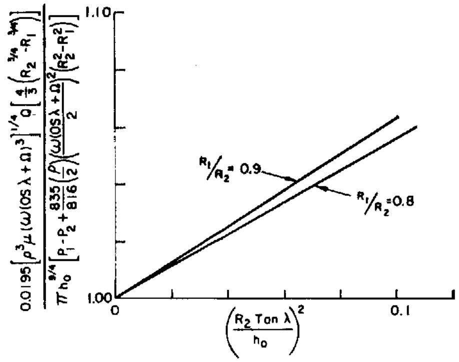

Fig. 3 Dimensionless leakage rate versus misalignment parameter

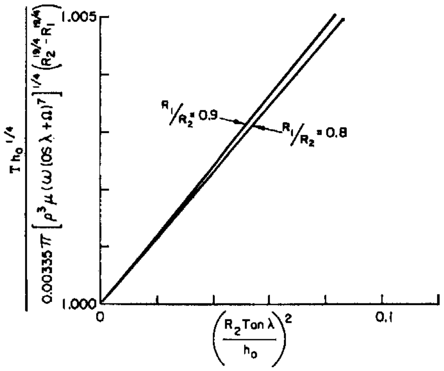

Fig. 4 Dimensionless torque versus misalignment parameter

$$\times\left[\frac{1-\left(\frac{R_1}{R_2}\right)^{19/4}}{1-\left(\frac{R_1}{R_2}\right)^{3/4}}\right]\left(\frac{R_2\tan\lambda}{h_0}\right)^4+\ldots\Bigg) \quad (38)\ (Cont.)$$

Fig. 3 illustrates the effect of misalignment on the net flow rate, where the deviation from the perfectly aligned case is proportional to $\left(\frac{R_2\tan\lambda}{h_0}\right)^2$.

Torque. The equation for the torque is

$$T = \int_0^{2\pi}\int_{R_1}^{R_2} \tau_{r\theta} r^2 dr d\theta = 0.0225\rho(\nu)^{1/4}$$
$$\times\left(\frac{\omega\cos\lambda+\Omega}{2}\right)^{7/4}\int_0^{2\pi}\int_{R_1}^{R_2}\frac{r^{11/4}}{(h/2)^{1/4}}drd\theta \quad (39)$$

Expanding the integrand in powers of $\frac{r\tan\lambda}{h_0}$ and integrating allows the torque to be expressed as

$$T = 0.00335\pi\left[\frac{\rho^3\mu(\omega\cos\lambda+\Omega)^7R_2^{19}}{h_0}\right]^{1/4}\left[1-\left(\frac{R_1}{R_2}\right)^{19/4}\right]$$
$$\times\left\{1+\frac{19}{27}\left(\frac{5}{64}\right)\left[\frac{1-\left(\frac{R_1}{R_2}\right)^{27/4}}{1-\left(\frac{R_1}{R_2}\right)^{19/4}}\right]\left(\frac{R_2\tan\lambda}{h_0}\right)^2+\ldots\right\} \quad (40)$$

Fig. 4 shows the effect of misalignment on torque, where again the deviation from the aligned case is proportional to $\left(\frac{R_2\tan\lambda}{h_0}\right)^2$.

Load Capacity. The pressure force acting to separate the seal surfaces is given by

$$F = \int_{R_1}^{R_2}\int_0^{2\pi} prdrd\theta = \pi P_1(R_2^2-R_1^2)$$
$$+\frac{\pi}{2}\left(\frac{835}{816}\right)\left(\frac{\rho}{2}\right)\left(\frac{\omega\cos\lambda+\Omega}{2}\right)^2(R_2^2-R_1^2)^2$$
$$+2\pi\left[\frac{P_2-P_1-\frac{835}{816}\left(\frac{\rho}{2}\right)\left(\frac{\omega\cos\lambda+\Omega}{2}\right)^2(R_2^2-R_1^2)}{(R_2^{3/4}-R_1^{3/4})}\right.$$
$$\times\left[\frac{4}{11}(R_2^{11/4}-R_1^{11/4})-\frac{R_1^{3/4}}{2}(R_2^2-R_1^2)\right] \quad (41)$$

$$\times\left(1+\frac{3}{8}\left[\frac{1}{\frac{4}{11}\left[1-\left(\frac{R_1}{R_2}\right)^{11/4}\right]-\frac{1}{2}\left(\frac{R_1}{R_2}\right)^{3/4}\left[1-\left(\frac{R_1}{R_2}\right)^2\right]}\right]\right.$$
$$\times\left\{\frac{9}{11}\left(\frac{13}{8}\right)\left\{\frac{4}{19}\left(1-\left(\frac{R_1}{R_2}\right)^{19/4}\right)-\frac{1}{2}\left(\frac{R_1}{R_2}\right)^{11/4}\left[1-\left(\frac{R_1}{R_2}\right)^2\right]\right\}\right.$$
$$-\frac{27}{28}\left(\frac{9}{7}\right)\left[\frac{1-\left(\frac{R_1}{R_2}\right)^{7/4}}{1-\left(\frac{R_1}{R_2}\right)^{3/4}}\right]\times\left\{\frac{4}{15}\left[1-\left(\frac{R_1}{R_2}\right)^{15/4}\right]\right.$$
$$-\frac{1}{2}\left(\frac{R_1}{R_2}\right)^{7/4}\left[1-\left(\frac{R_1}{R_2}\right)^2\right]\right\}+\frac{4}{3}\left\{\frac{4}{11}\left[1-\left(\frac{R_1}{R_2}\right)^{11/4}\right]\right.$$
$$-\frac{1}{2}\left(\frac{R_1}{R_2}\right)^{3/4}\left[1-\left(\frac{R_1}{R_2}\right)^2\right]\right\}\left[\left(\frac{27}{28}\right)^2\left(\frac{1-\left(\frac{R_1}{R_2}\right)^{7/4}}{1-\left(\frac{R_1}{R_2}\right)^{3/4}}\right)^2\right.$$
$$\left.-\frac{27}{32}\left(\frac{13}{11}\right)\left(\frac{1-\left(\frac{R_1}{R_2}\right)^{11/4}}{1-\left(\frac{R_1}{R_2}\right)^{3/4}}\right)\right]\right\}\left(\frac{R_2\tan\lambda}{h_0}\right)^2+\ldots\Bigg) \quad (41)\ (Cont.)$$

having once again employed an expansion in powers of $\frac{r\tan\lambda}{h_0}$. The force F is not the net force exerted on the seal. That force is computed by taking into account the external pressure and mechanical forces exerted on the seal.

A numerical investigation of the effects of misalignment for $R_1/R_2 = 0.8$ and 0.9 with $\left(\frac{R_2\tan\lambda}{h_0}\right)^2 < 0.1$ shows a negligible decrease in this force due to misalignment.

Conclusion. As with laminar flow, the effects of misalignment on the performance characteristics can be ranked in the same order as changes in the nominal clearance affect each of these characteristics; i.e., the leakage is most greatly affected, the torque is mildly affected, and separating force practically unaffected.

Aligned Face Seal

Film Thickness Varying Radially. In face seals where misalignment effects are negligible, variations in film thickness from other causes, such as thermal or elastohydrodynamic distortions, could influence the performance characteristics. If these distortions result in film thickness variations which are purely radial, then the leakage rate is given by

$$\frac{0.0195(\rho^3\mu\omega^3)^{1/4}Q}{\pi}\int_{R_1}^{R_2}\frac{dr}{r^{7/4}h^{9/4}} = P_1 - P_2$$
$$+\frac{835}{816}\left(\frac{\rho}{2}\right)\left(\frac{R_2\omega}{2}\right)^2\left[1-\left(\frac{R_1}{R_2}\right)^2\right] \quad (42)$$

The right-hand side of this equation could be interpreted as the effective pressure causing leakage, which is seen to be slightly less than that expected for laminar flow under the same operating conditions [1]. The major difference between laminar and turbulent flow occurs on the left-hand side where the effects of speed, geometry, and fluid properties are seen to be quite different.

The load carrying capacity for such a seal is calculated from the equation

$$\frac{F}{\pi(R_2^2-R_1^2)}+\frac{0.039(\rho^3\mu\omega^3)^{1/4}Q}{\pi(R_2^2-R_1^2)}\int_{R_1}^{R_2}\left[\int_{R_1}^{r}\frac{d\xi}{\xi^{7/4}h^{9/4}}\right]rdr$$
$$= P_1+\frac{1}{2}\left(\frac{835}{816}\right)\left(\frac{\rho}{2}\right)\left(\frac{R_2\omega}{2}\right)^2\left[1-\left(\frac{R_1}{R_2}\right)^2\right] \quad (43)$$

Here again the right-hand side may be interpreted as the effective internal pressure which differs only slightly from that found in the laminar flow. The term on the left-hand side containing the effects of geometry once again differs significantly from the laminar result of reference [1].

The torque, which is unaffected by inertia, is obtained by the straightforward integration of equation (39) treating the film thickness as a function of the radius only.

In the aligned case the pressure gradient is related to the centrifugal force effect and the leakage rate through the equation

$$\frac{dp}{dr} = \frac{835}{816}\rho r\left(\frac{\omega}{2}\right)^2 - \frac{0.0195}{\pi}\left(\frac{\rho^3\mu\omega^3}{rh^9}\right)^{1/4}Q \quad (44)$$

so that the condition for reversed flow given by equation (36) may be expressed somewhat more explicitly by

$$-27.7 \le \left(\frac{36\mu Q}{\rho(r\omega)^2h^2\pi}\right)\left(\frac{\rho r\omega h}{\mu}\right)^{1/4} \le 23.6 \quad (45)$$

Film Thickness Varying Tangentially. Should the film thickness vary tangentially (despite precautions to the contrary) as has been observed by Nau [3], the film thickness could be approximated by the expression

$$h = c(1 + a\cos k\theta) \quad (46)$$

Following a procedure similar to the one used earlier, the pressure distribution is found to take the general form

$$p = \left(\frac{835}{816}\right)\rho\left(\frac{r^2}{2}\right)\left(\frac{\omega}{2}\right)^2 - 0.0225\left(\frac{30}{49}\right)\left(\frac{4}{11}\right)\frac{\rho\nu^{1/4}}{2}$$
$$\times\left[\frac{(\omega/2)^{7/4}r^{11/4}}{(h/2)^{5/4}}\right]\frac{\partial h}{\partial\theta}+\frac{4}{3}f(\theta)r^{3/4}+g(\theta) \quad (47)$$

Using the arbitrary functions of integration to satisfy the boundary conditions of equation (31) yields

$$p - P_2 = \left(\frac{835}{816}\right)\left(\frac{\rho}{2}\right)\left(\frac{\omega}{2}\right)^2[r^2-R_2^2] - 0.0225$$
$$\times\left(\frac{30}{49}\right)\left(\frac{4}{11}\right)\frac{\rho\nu^{1/4}}{2}\left[\frac{(\omega/2)^{7/4}}{(h/2)^{5/4}}\right](r^{11/4}-R_2^{11/4})\frac{\partial h}{\partial\theta}$$
$$+\left[\frac{r^{3/4}-R_2^{3/4}}{R_1^{3/4}-R_2^{3/4}}\right]\left\{P_1-P_2+\left(\frac{835}{816}\right)\frac{\rho}{2}\left(\frac{\omega}{2}\right)^2\right.$$
$$\times(R_2^2-R_1^2)+0.0225\left(\frac{30}{49}\right)\left(\frac{4}{11}\right)\frac{\rho\nu^{1/4}}{2}$$
$$\left.\times\left[\frac{(\omega/2)^{7/4}}{(h/2)^{5/4}}\right](R_2^{11/4}-R_1^{11/4})\frac{\partial h}{\partial\theta}\right\} \quad (48)$$

The leakage rate is calculated from the formula

$$Q = \frac{1}{0.0225\left(\frac{30}{49}\right)\rho\nu^{1/4}\left(\frac{\omega}{2}\right)^{7/4}}\int_0^{2\pi}f(\theta)\left(\frac{h}{2}\right)^{5/4}d\theta \quad (49)$$

Expanding the film thickness function in the integrand using the binomial theorem and integrating gives

$$\frac{0.0195}{\pi}\left(\frac{\rho^3\mu\omega^3}{c^9}\right)^{1/4}\frac{4}{3}(R_2^{3/4}-R_1^{3/4})Q = \left\{P_1-P_2+\frac{835}{816}\right.$$
$$\left.\times\left(\frac{\rho}{2}\right)\left(\frac{R_2\omega}{2}\right)^2\left[1-\left(\frac{R_1}{R_2}\right)^2\right]\right\}\left[1+\frac{45}{64}a^2+\ldots\right] \quad (50)$$

where the limitations of the theory restrict a^2 to the order of 0.1 or less. The radial pumping effect of the surface waviness appears on the right-hand side of this equation, and its influence is of the same order of magnitude as the misalignment investigated earlier.

The torque may be obtained using the same technique with the result that

$$\frac{T}{0.00335\pi}\left(\frac{c}{\rho^3\mu\omega^7R_2^{19}}\right)^{1/4} = \left[1-\left(\frac{R_1}{R_2}\right)^{19/4}\right]$$
$$\times\left[1+\frac{5}{64}a^2+\ldots\right] \quad (51)$$

where the increase in torque due to waviness is comparable to that associated with misalignment.

Since the film thickness equation chosen is periodic the load carrying capacity, or separating force, is the same as would be calculated for a flat seal operating at a constant clearance equal to c.

Cavitation

It was observed in the previous analysis of the laminar face seals that the tendency to form regions of reversed flow [equations (36), (45)] could cause the ingestion of gases from the outer perimeter of the seal and lead ultimately to the separation of the liquid from the stationary surface forming a free surface within the clearance space. Cavities formed in this way have been described by Birkhoff and Hays [4] by the term "separation cavitation." A thorough investigation of this phenomena is beyond the scope of this paper and will be the subject of a subsequent analysis.

"True cavitation," in contrast to separation cavitation, results in the complete rupture of the fluid film from surface to surface. The formation of such a cavity is attributed by Nau [5] to dissolved gases in the liquid coming out of solution in regions where the film pressure is less than atmospheric.

The analysis of true cavitation follows the same procedure as in laminar flow [1]. Fig. 5 shows the notation used in the investigation of the parallel surface face seal (see Appendix for the case where the film thickness varies in the radial direction).

Using the rupture criteria that $\frac{dp}{dr} = 0$ at the inner boundary of the cavity and assuming that surface tension effects are negligible

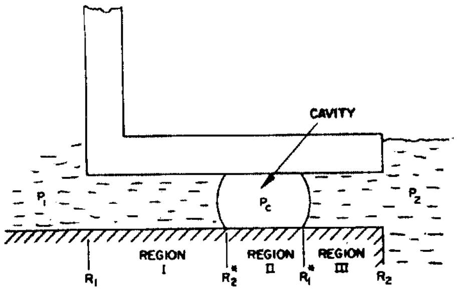

Fig. 5 True cavitation

and assuming that surface tension effects are negligible [5], equation (44) shows that the inner radius is given by

$$(R_2^*)^{5/4} = \frac{0.0762}{\pi}\left(\frac{\mu}{\rho\omega^3c^3}\right)^{1/4} Q \qquad (52)$$

The pressure drop across region I is

$$p - P_c = \frac{1}{3}\left(\frac{835}{816}\right)\rho(R_2^*\omega)^2\left[\frac{5}{8} - \left(\frac{R_1}{R_2^*}\right)^{5/4} + \frac{3}{8}\left(\frac{R_1}{R_2^*}\right)^2\right] \qquad (53)$$

From region III, the equation for the outer boundary of the cavity is found to be

$$\left(\frac{R_1^*}{R_2}\right)^{1/4} = \frac{3}{8}\left[\frac{1-\left(\dfrac{R_1^*}{R_2}\right)^2}{1-\left(\dfrac{R_1^*}{R_2}\right)^{7/4}}\right] - 3\left(\frac{816}{835}\right)\frac{P_2 - P_c}{\rho(R_2\omega)^2}\left[\frac{1}{1-\left(\dfrac{R_1^*}{R_2}\right)^{7/4}}\right] \qquad (54)$$

In the last equation it is evident that P_2 must be greater than P_c for $R_2^*/R_2 < 1.0$ and $R_2^*/R_1^* < 1.0$. It is quite apparent that the cavity pressure is the critical quantity in true cavitation. The magnitude of P_c can be controlled by many factors, among them, the concentration of the dissolved gases, the equilibrium rate of solution and dissolution, and the temperature to name a few. It would not be surprising, therefore, to find fluids reacting quite differently to the same stimuli, some cavitating quite easily, others not at all. It is also clear from equation (44) that since the pressure gradient vanishes at R_2^* instead of having negative value as it would be in a cavitationless pressurized seal, the leakage rate with cavitation will be less than without it.

In conclusion it should be noted that the two types of cavitation just described are associated with extremely low leakage rates, i.e., small pressure differences, and hence may be a relatively rare occurrence under ordinary seal operating conditions.

Limitations of the Theory

It is not possible at this time to compare the theory with actual seal performance, since experimental data for turbulent seal operation is presently unavailable. It is possible, however, to compare the results of the theory with the experimental behavior of other turbulent flows which are related to the flow in a face seal, and thereby obtain indirect evidence concerning the validity of the theory.

Torque. The laminar flow would be expected to fill the clearance space when

$$\frac{V^*c}{2\nu} < 5.0 \qquad (55)$$

In laminar flow

$$\tau_w = \mu\frac{r\omega}{c} \qquad (56)$$

so that laminar flow would be expected when

$$\frac{R\omega^2}{\nu} < 100\frac{R}{c} \qquad (57)$$

A fully turbulent core flow exists when

$$\frac{V^*c}{2\nu} > 30 \qquad (58)$$

Using equation (18), with $\lambda = \Omega = 0$ and $h = c$ shows that a predominantly turbulent film would be expected when

$$\frac{R^2\omega}{\nu} > 1700\frac{R}{c} \qquad (59)$$

The $^1/_7$ power law approximation to the universal velocity profile should be valid when

$$1700\frac{R}{c} < \frac{R^2\omega}{\nu} < 40{,}000\frac{R}{c} \qquad (60)$$

The transitions predicted by equations (57) and (59) are in good agreement with the experimental results presented by Schlichting [6], for a disk with $R_1 = 0$.

If the torque coefficient, defined by the equation

$$C_M = \frac{2T}{\rho\omega^2R_2^5/2} \qquad (61)$$

is computed from equation (51) with $a = \dfrac{R_1}{R_2} = 0$, it is found that

$$C_M = \frac{0.042}{\left[\dfrac{R_2^2\omega}{\nu}\left(\dfrac{c}{R_2}\right)\right]^{1/4}} \qquad (62)$$

Comparing this equation with those in [6] shows that this equation predicts a C_M which is somewhat less than that actually measured. Daily and Nece [7] noted the same discrepancy for disks where $R_1 \ll R_2$, and proposed that the coefficient in equation (14) be chosen to be 0.03955 rather than 0.0225. This suggestion is based on the results of Soo [8], which are logically incorrect, so that this correction must be considered to be purely empirical.

From the available torque data, one would conclude that the theory is qualitatively correct while leaving something to be desired quantitatively. Since data cited is for disks with no central hole, and the theory is based on the assumption that the clearance is small compared to the local radius, the overall comparison could be considered encouraging.

Coupled Couette-Poiseuille Flow. The assumption that the tangential velocity profile and wall shear stress may be represented by the $^1/_7$ power law is not a crucial point in this analysis. Any other velocity profile and associated shear stress may be used, e.g., those used in the laminar analysis [1]. The $^1/_7$ power law was chosen because of its simplicity and generally widespread acceptance. The crucial point of the analysis is hypothesis that turbulence is generated by seal rotation, and that the apparent viscosity distribution caused by this rotation controls the leakage rate through the seal.

A simple calculation using typical fluid properties and seal dimensions will show that in the absence of rotation turbulent radial flow requires the application of extremely large pressure

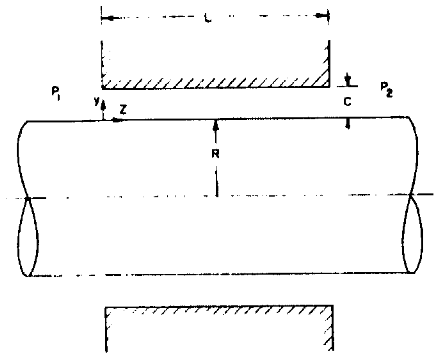

Fig. 6 Shaft seal

differences. So it seems certain that if turbulence occurs it is most likely the result of rotation.

The question of the mechanism whereby the rotational generated turbulence influences the flow is not so easily settled. The best that can be done at the present time is to show that when the concept of coupled shear and pressure flows used in this analysis is applied to another configuration which has been investigated experimentally, it predicts with reasonable accuracy the observed performance. Such a configuration is the rotating shaft seal investigated analytically and experimentally by Yamada [9] and illustrated in Fig. 6. The equations of motion for concentric operation of the seal are given by

$$\frac{\partial p}{\partial x} = \frac{\partial \tau_{yx}}{\partial y} \qquad (63)$$

$$\frac{\partial p}{\partial z} = \frac{\partial \tau_{yz}}{\partial y} \qquad (64)$$

Combining these equations by eliminating the apparent viscosity and integrating twice with respect to y, yields an equation analogous to equation (24)

$$w\tau_{yxw} = vf(x, z) + \frac{\partial p}{\partial z}\left(vy - \int_0^y v\,dy\right) + g(x, z) \qquad (65)$$

where w and v are the axial and tangential velocities, respectively.

Splitting the clearance space in half and assuming that a power-law profile can be used in each region, the tangential velocity distributions due to tangential shear are given by

$$\frac{v}{V/2} = \left(\frac{y}{c/2}\right)^{1/n}, \quad 0 \le y \le c/2$$
$$\frac{v}{V/2} = 2 - \left(\frac{c-y}{c/2}\right)^{1/n}, \quad c/2 \le y \le c \qquad (66)$$

where continuity of v and $\dfrac{\partial v}{\partial y}$ are satisfied at $y = c/2$.

Using the arbitrary functions of integration to satisfy the boundary conditions $w = 0$ at $y = 0$ and c, the axial velocity distribution becomes

$$w\tau_{yxw} = -\frac{V}{2}\left(\frac{c}{2}\right)\left\{\left(1 - \frac{y}{c/2}\right)\left(\frac{y}{c/2}\right)^{1/n} + \left(\frac{1}{1+\frac{1}{n}}\right)\left(\frac{y}{c/2}\right)^{1+\frac{1}{n}}\right\}\frac{\partial p}{\partial z} \qquad (67)$$

when $0 \le y \le c/2$, and

$$w\tau_{yxw} = -\frac{V}{2}\left(\frac{c}{2}\right)\left\{\left(\frac{y}{c/2} - 1\right)\left(\frac{c-y}{c/2}\right)^{1/n} + \left(\frac{1}{1+\frac{1}{n}}\right)\left(\frac{c-y}{c/2}\right)^{1+\frac{1}{n}}\right\}\frac{\partial p}{\partial z}$$

when $c/2 \le y \le c$.

The axial leakage rate is given by

$$Q = 2\pi R\int_0^h w\,dy = \frac{\pi NRVc^2}{2\tau_{yxw}}\left(\frac{P_1 - P_2}{L}\right) \qquad (68)$$

where

$$N = \left(\frac{1}{1+n} - 1\right)\left(\frac{1}{2+\frac{1}{n}}\right) + \left(\frac{1}{1+\frac{1}{n}}\right) \qquad (69)$$

In this equation $\dfrac{\partial p}{\partial z}$ has been replaced by $\dfrac{P_2 - P_1}{L}$ since continuity considerations required $\dfrac{dQ}{dz} = 0$.

All that remains is to supply the appropriate formulation for τ_{yxw}. If the $^1/_7$ power law is chosen for the case where $P_1 - P_2 = 0$, then

$$\tau_{yxw} = 0.0225\frac{\rho\nu^{1/4}(V/2)^{7/4}}{(c/2)^{1/4}} \qquad (70)$$

and the coefficient of friction for shear flow becomes

$$C_f = \frac{2\tau_{yxw}}{\rho\left(\dfrac{V}{2}\right)^2} = \frac{0.045}{R_s^{1/4}} \qquad (71)$$

where

$$R_s = \frac{(V/2)(c/2)}{\nu} \qquad (72)$$

The accuracy of this equation for turbulent journal bearings has been confirmed empirically by Burton [2] who compared it with the experimental results of Smith and Fuller [10].

If equation (70) is now corrected to allow for pressure-induced axial leakage, it takes the form

$$\tau_{xrw} = 0.0225\rho\left(\frac{\nu}{c/2}\right)^{1/4} v_r^{7/4}\left[1 + \left(\frac{w_r}{v_r}\right)^2\right]^{7/8} \qquad (73)$$

where the midchannel relative velocities are given by

$$v_r = V/2$$
$$w_r = \left[\frac{1}{N\left(1+\frac{1}{n}\right)}\right]W_a \qquad (74)$$
$$W_a = \frac{Q}{2\pi Rc}$$

Defining the leakage coefficient of friction f by the equation

$$\frac{P_1 - P_2}{\rho} = f\frac{L}{2c}\frac{W_a^2}{2} \qquad (75)$$

and setting $n = 7$, the leakage coefficient can be determined combining equations (68), (69), (73), (74), and (75). The result is

$$f = \frac{0.337}{R_a^{1/4}}\left[1 + \left(0.357\frac{R_\omega}{R_a}\right)^2\right]^{3/8} \qquad (76)$$

where

$$R_\omega = \frac{Vc}{\nu}$$
$$R_e = \frac{W_0 c}{\nu} \tag{77}$$

Yamada, after an extensive experimental investigation of turbulent shaft seals, suggests the semiempirical formula for small R_e and $R\omega > 10^4$

$$f = \frac{0.26}{R_e^{0.24}}\left[1 + \left(\frac{7}{16}\,\frac{R_\omega}{R_e}\right)^2\right]^{0.38} \tag{78}$$

Yamada's restrictions on this equation are exactly the type imposed on the face seal analysis by last condition of equation (7). For $\frac{R\omega}{R_e} \ll 1.0$ equations (76) and (77) reduce to

$$f \rightarrow 0.1555\left(\frac{R_\omega^{1/4}}{R_e}\right) \tag{79}$$

and

$$f \rightarrow 0.1435\left(\frac{R_\omega^{0.76}}{R_e}\right) \tag{80}$$

It is evident from this comparison that the rotationally generated turbulence of the tangential Couette flow is exercising an influence on the turbulent axial Poiseuille flow as demonstrated by equations (76) and (78). Furthermore, where the axial flow is small compared to the rotationally induced tangential flow, this coupling is essentially unidirectional from the Couette to the Poiseiulle flow and consistant with the basic hypothesis of this paper.

Conclusions

1 Misalignment and surface waviness were found to be somewhat less influential with turbulent flow than with laminar flow. In order of importance, these effects were shown to affect seal performance in the same order as changes in clearance affect the aligned, flat surface seal characteristics, i.e., leakage rate the most, separating force imperceptibly, with the torque in between.

2 Regions of subambient pressure and reversed flow resulting from centrifugal force effects were also found. As in laminar flow, the presence of these regions indicates that under the proper conditions either true or separation cavitation could be initiated which would be accompanied by substantial changes in the seal performance characteristics.

3 Comparison of the results of the theory with available experimental evidence indicates good qualitative agreement and encouraging quantitative agreement. Final verification of the theory awaits comparison with definitive experimental evidence.

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APPENDIX

The equation for region I of the cavitating seal which must be satisfied by the proper selection of Re_2^* when the film thickness varies radial direction is given by

$$P_1 - P_c = \frac{1}{4}\left(\frac{835}{816}\right)\rho(R_2^*\omega)^2 \times \left\{\frac{h^{5/4}(R_2^*)}{(R_2^*)^{5/4}}\int_{R_1}^{R_1^*}\frac{dr}{r^{1/4}h^{5/4}} - \frac{1}{2}\left[1 - \left(\frac{R_1}{R_1^*}\right)^2\right]\right\}$$

where $h(R_2^*)$ is the film thickness at R_2^*.

Given R_2^*, $R_1^* < R_2^*$ must then satisfy the equation

$$\left(\frac{R_2^*}{R_2}\right)^{5/4} = 4\left(\frac{816}{835}\right)\left\{\frac{\frac{P_2 - P_c}{\rho(R_2\omega)^2} + \frac{1}{8}\left(\frac{835}{816}\right)\left[1 - \left(\frac{R_1^*}{R_2}\right)^2\right]}{\frac{h^{5/4}(R_2^*)}{(R_2)^{5/4}}\int_{R_1^*}^{R_2}\frac{dr}{r^{1/4}h^{5/4}}}\right\}$$

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III. I

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Reversed Flow in Face Seals

The stream surfaces for laminar and turbulent flow in parallel surface seals are analytically determined for the cases where the fluid separates from one of the solid boundaries. These stream surfaces are used to determine when there is an exchange of external fluid across the seal land as well as the rate of these exchanges.

Introduction

In an earlier investigation [1][1] it was noted that when a face seal was designed to have zero or nearly zero net leakage rate, the radial velocity would be directed radially inward along the stationary disk and outward along the rotating disk. The existence of these oppositely directed radial flow fields is attributable to centrifugal force effects within the fluid film. The hydrodynamic features of this multidirectional flow were not explored in the earlier study other than to note that zero net leakage rate did not mean that there was no exchange of the inner and outer fluids across the seal land.

The analysis to follow is a more detailed investigation of the nature of these reversed flow regimes. The objective of the investigation is to develop an analytical method for: (1) determining when there is an exchange of the inner and outer fluid, and (2) determining the rates at which the inner and outer fluid are exchanged when they are.

Analysis of Laminar Flow

The aligned, flat face seal possesses the kind of symmetry which lends itself to the analysis of the problem at hand. It was shown in reference [1] that reversed flow regimes would be present for the assumed configuration when

$$\frac{1}{6} < \frac{\frac{dp}{dr}}{\rho r\omega^2} < \frac{1}{2} \tag{1}$$

where

$$\frac{dp}{dr} = \frac{3}{10}\rho r\omega^2 - \frac{6\mu Q}{\pi r c^3} \tag{2}$$

In terms of the net leakage rate the criterion of equation (1) can also be expressed as

$$-\frac{1}{60} < \beta(r) < \frac{1}{90} \tag{3}$$

where

$$\beta(r) \equiv \frac{\mu Q}{2\rho(r\omega)^2 c^3\pi} \tag{4}$$

It should be noted at the outset that reversed flow is possible only when pressure increases radial outward as indicated by equation (1). For net radial outflows, which are positive, reversed flow fields are possible only when the centrifugal pressure rise of equation (2) more than offsets the viscous pressure drop.

The investigation of the laminar flow field is most conveniently carried out with the aid of a suitably defined stream function. Since the flow is axisymmetric this stream function will be defined by

$$u = -\frac{\partial\psi}{\partial z}, \quad w = \frac{1}{r}\frac{\partial}{\partial r}(r\psi) \tag{5}$$

Stream surfaces tangent to the velocity vector are given by the equation

$$r\psi = \text{constant} \tag{6}$$

It was shown in references [1] and [2] that the stream function satisfying the continuity and momentum equations was

$$\psi(r, \xi) = cr\omega R_e W(\xi) + \frac{Q}{2\pi r}U(\xi) \tag{7}$$

where to a first-order approximation

$$V(\xi) \cong 2\xi^3 - 3\xi^2 \tag{8}$$

$$W(\xi) \cong -\frac{1}{20}\xi^3 + \frac{7}{60}\xi^2 - \frac{1}{12}\xi^4 + \frac{1}{60}\xi^5 \tag{9}$$

and the Reynold's number is defined by

$$R_e = \frac{c^2\omega}{\nu} \tag{10}$$

In these equations $\xi = [1 - (z/c)]$ where z is measured from the nonrotating surface toward the rotating one (Fig. 1). The values of the stream function at the bounding surfaces are

$$\psi(r, 0) = 0 \tag{11a}$$

$$\psi(r, 1) = -\frac{Q}{2\pi r} \tag{11b}$$

Recently a more comprehensive analysis of the flow between

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[1] Numbers in brackets designate References at end of paper.

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Nomenclature

$a = c/2$
c = axial clearance
q = exchange leakage rate
Q = net leakage rate
r = radial coordinate
R_1 = inner radius
R_2 = outer radius
$R_e = \frac{c^2\omega}{\nu}$
R_s = separation radius
$u = \bar{u}$ = radial velocity component
$\bar{v}$ = tangential velocity component
z = axial coordinate measured from stationary disk
$\alpha_2 = \frac{a^2\omega}{\nu}$
$\beta(r) = \frac{\mu Q}{2\rho(r\omega)^2 c^3\pi}$
μ = viscosity
$\nu = \mu/\rho$
$\xi = 1 - z/c$
ξ_t = coordinate of streamline turning point at seal radial boundary
ρ = density
$\tau_{\theta\omega}$ = tangential wall shear stress
ψ = stream function
ω = angular velocity of rotating disk

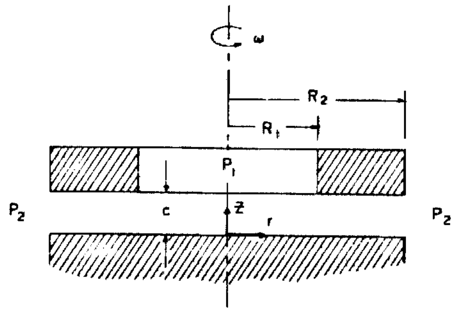

Fig. 1 Face seal configuration

rotating disks has been carried out by Kreith and Viviand [3]. A comparison of their results with the formulations given in reference [1] shows close agreement for seal-like configurations (see Appendix).

The volumetric flow-rate passing between any stream surface and the rotating disk is given by

$$q = 2\pi \int_0^z r u dz$$

$$= 2\pi r[\psi(r, 0) - \psi(r, \xi)]$$

$$= -2\pi r\psi(r, \xi) \qquad (12)$$

Laminar Net Radial Outflow ($Q > 0$)

In this case the internal and external fluid flows are separated from each other by the stream surface which branches from the stationary surface, $r\psi = -(Q/2\pi)$ (Fig. 2). The coordinates of this surface are determined by solving the equation

$$W(1) - W(\xi) + \beta(r)[U(1) - U(\xi)] = 0 \qquad (13)$$

for ξ, given certain values of r. Actually, it is more convenient to select values of $\beta(r)$ rather than r, since it is known that

$$0 < \beta(r) < \frac{1}{90}, \qquad Q > 0 \qquad (14)$$

Fig. 3 shows the results of such a calculation from which the coordinates of the stream surface can be obtained. In a practical sense after having designed a seal and determined its net leakage rate using the analysis of reference (1), one would next calculate $\beta(R_1)$ from equation (4). If $\beta(R_1) \leq 1/90$, then for $R_1 > r >$

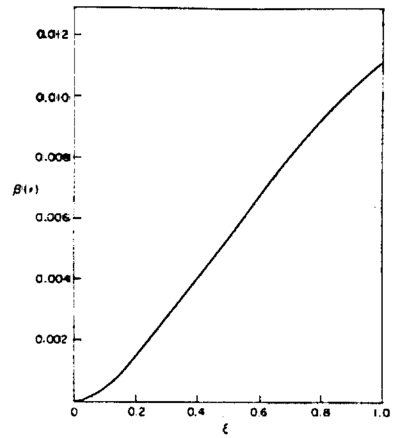

Fig. 3 Laminar flow β(r) versus ξ for Q > 0

$R_2\beta(r)$ will be in the range given by equation (14) and a reversed flow will be present.

What is important from the standpoint of exchange flows is not so much what the stream surface looks like, but rather at what radius it branches from the nonrotating surface. If it branches as shown in Fig. 2, then there will be no fluid exchange. If, however, the branching occurs at a radius R_s less than R_1, then an exchange will occur. The separation radius is calculated from

$$R_s^2 = \frac{45\mu Q}{\rho\omega^2 c^3 \pi} \qquad (15)$$

Laminar Net Radial Inflow ($Q < 0$)

In this case the separating stream surface branches from the rotating surface (Fig. 4) and the coordinates of this surface are determined by the roots of

$$W(\xi) + \beta(r)U(\xi) = 0 \qquad (16)$$

for $\beta(r)$ given in the range

$$-\frac{1}{60} < \beta(r) < 0, \qquad Q < 0 \qquad (17)$$

Fig. 5 shows the results of this calculation. There will be an

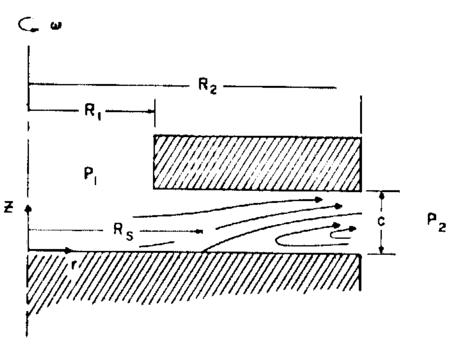

Fig. 2 Reversed flow field with net outflow (Q > 0)

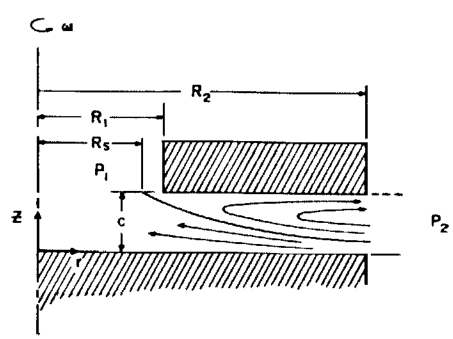

Fig. 4 Reversed flow field with net inflow (Q < 0)

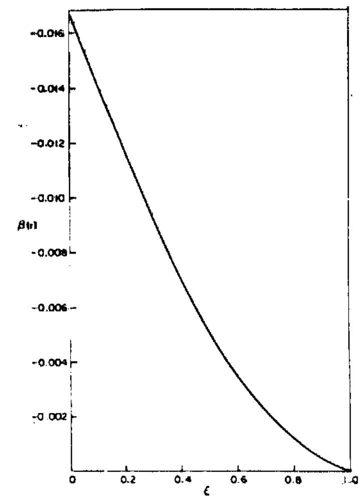

Fig. 5 Laminar flow β(r) versus ξ for Q < 0

exchange of inner and outer fluid if $R_s < R_1$ as shown in Fig. 4. In this case

$$R_s^2 = -\frac{30\mu Q}{\rho\omega^2 c^3 \pi} \qquad (18)$$

Laminar Net Radial Flow Zero ($Q = 0$)

The case of no net leakage rate cannot be treated in the same way as the two previous cases. In this case

$$\psi(r, \xi) = c\omega R_s W(\xi) \qquad (19)$$

From equations (11) it is clear that the z axis, as well as the stationary and the rotating disks, lie in the stream surface $r\psi = 0$. The result is that ψ is multivalued in the range $0 \leq \xi \leq 1.0$. It is not difficult, however, to show that the radial velocity vanishes on the surface $\xi = 0.475$ for all radii. With this information the stream surfaces can be sketched as shown in Fig. 6.

The solution given by equation (19) is actually valid for disks with $R_1 = 0$ as shown in Fig. 6 which is the subject of [3]. For seals, however, the stream surfaces are only meaningful for $R_1 < r < R_2$.

It is interesting to note that if the inner and outer radii of a face seal are exposed to open reservoirs of liquid, the heights of the fluid in the reservoirs will eventually adjust themselves so that the hydrostatic pressure difference cancels the centrifugally induced pressure rise, yielding $Q = 0$. This condition is analogous to the shutoff condition in a centrifugal pump, but can be attained here only when there is an exchange flow across the seal.

Exchange Flow Rates in Laminar Flow

The exchange flow rates are most conveniently evaluated by determining the number value of the stream function corresponding to the stream surface which reverses its direction at the seal inner or outer radius.

For example, when $Q > 0$ the distance ξ_t shown in Fig. 7(*a*) locates the appropriate stream surface. This point is easily determined since it lies on the surface $u = 0$. The formula for the surface $u = 0$ was derived in [1] and shown to be

$$\xi = 1 - \frac{1}{2}\left[\sqrt{1 + 4\left(\frac{4}{5} - 72\beta(r)\right)} - 1\right] \qquad (20)$$

When $Q > 0$, ξ_t is given by

$$\xi_t = 1 - \frac{1}{2}\left[\sqrt{1 + 4\left(\frac{4}{5} - 72\beta(R_1)\right)} - 1\right] \qquad (21)$$

so that

$$q_{out} = -2\pi R_1 \psi(R_1, \xi_t) \qquad (22)$$

and

$$q_{in} = Q - q_{out} \qquad (23)$$

Similarly for $Q < 0$ as shown in Fig. 7(*b*)

$$\xi_t = 1 - \frac{1}{2}\left[\sqrt{1 + 4\left(\frac{4}{5} - 72\beta(R_1)\right)} - 1\right] \qquad (24)$$

so that

$$q_{out} = -2\pi R_1 \psi(R_1, \xi_t) \qquad (25)$$

and

$$q_{in} = Q - q_{out} \qquad (26)$$

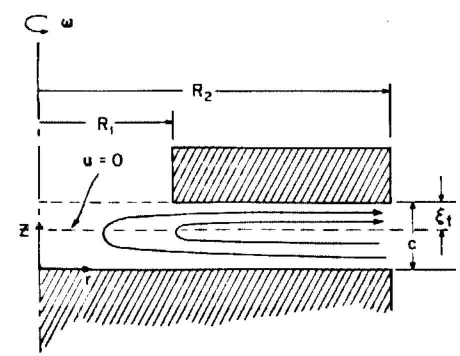

Fig. 6 Reversed flow field with zero net flow (Q = 0)

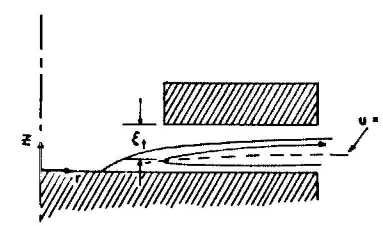

Fig. 7a Exchange flow for Q > 0

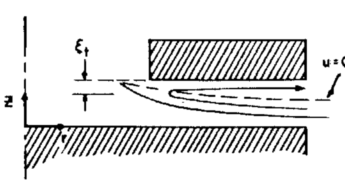

Fig. 7b Exchange flow for Q < 0

When $Q = 0$, the stream surface separating in inward and outward flows is coincident with the $u = 0$ surface (Fig. 6), given by $\xi_i = 0.475$, so that

$$\psi(R_1, 0.475) = cR_1\omega R_s W(0.475) = -0.00262cR_1\omega R_s \quad (27)$$

The inflow and outflow rates are equal, of course, in magnitude and given by

$$q_{out} = -q_{in} = 0.00262(2\pi R_1)cR_1\omega R_s \quad (28)$$

Analysis of Turbulent Flow

Reversed flow under turbulent conditions can be studied using basically the same techniques as were used for laminar flow.

It was determined analytically in reference [4] that reversed flow would occur when

$$-27.7 \le \left(\frac{36\mu Q}{\rho\pi c^3(r\omega)^2}\right)\left|\frac{\rho r\omega c}{\mu}\right|^{1/4} \le 23.6 \quad (29)$$

It was also shown in reference [4] that the radial velocity profile could be represented by the relation

$$u\tau_{r\theta_w} = -\left(\frac{r\omega}{2}\right)\left(\frac{c}{2}\right)F\left(\frac{z}{c/2}\right)\frac{dp}{dr} + \frac{\rho}{r}\left(\frac{r\omega}{2}\right)^2\left(\frac{c}{2}\right)G\left(\frac{z}{c/2}\right) \quad (30)$$

where

$$\frac{dp}{dr} = \frac{835}{816}\rho r\left(\frac{\omega}{2}\right)^2 - \frac{0.0195}{\pi}\left|\frac{\rho^3\mu\omega^3}{rc^9}\right|^{1/4} Q \quad (31)$$

and

$$F\left(\frac{z}{c/2}\right) = \left(\frac{z}{c/2}\right)^{1/7}\left[1 - \frac{1}{8}\left(\frac{z}{c/2}\right)\right], \quad 0 \le z \le c/2.$$

$$= \left(\frac{c-z}{c/2}\right)^{1/7}\left[\frac{3}{4} + \frac{1}{8}\left(\frac{z}{c/2}\right)\right], \quad c/2 \le z \le c \quad (32)$$

$$G\left(\frac{z}{c/2}\right) = \left(\frac{z}{c/2}\right)^{1/7}\left[\frac{35}{36} - \frac{7}{90}\left(\frac{z}{c/2}\right)^{8/7}\right], \quad 0 \le z \le c/2$$

$$= \left(\frac{c-z}{c/2}\right)^{1/7}\left[\frac{1}{12} + \frac{1}{2}\left(\frac{z}{c/2}\right) + \frac{7}{18}\left(\frac{c-z}{c/2}\right)^{8/7} - \frac{7}{90}\left(\frac{c-z}{c/2}\right)^{9/7}\right], \quad c/2 \le z \le c \quad (33)$$

It should be noted that in these equations z is measured from the stationary surface, and also that $\tau_{r\theta_w}$ is the tangential wall shear stress and is independent of z. The functions F and G are tabulated in Table 1.

Table 1 Functions $F\left(\frac{z}{c/2}\right)$ and $G\left(\frac{z}{c/2}\right)$

$\left(\frac{z}{c/2}\right)$	$F\left(\frac{z}{c/2}\right)$	$G\left(\frac{z}{c/2}\right)$
0.0	0.000	0.000
0.1	0.711	0.697
0.2	0.776	0.765
0.4	0.833	0.830
0.6	0.860	0.866
0.8	0.873	0.886
1.0	0.875	0.894
1.2	0.873	0.896
1.4	0.860	0.893
1.6	0.833	0.873
1.8	0.776	0.823
1.9	0.711	0.762
2.0	0.000	0.000

Turbulent Net Radial Outflow ($Q > 0$)

Combining equations (30) and (31) and using the definition of the stream function given in equation (5), the coordinates of the stream surface branching from the stationary seal surface are given by the zeros of the equation

$$\int_0^z G\left(\frac{z}{c/2}\right)dz + \left[\left(\frac{0.0195}{9}\right)\left(\frac{36\mu Q}{\pi\rho c^3(r\omega)^2}\right)\left|\frac{\rho r\omega c}{\mu}\right|^{1/4} - \frac{835}{816}\right]\int_0^z F\left(\frac{z}{c/2}\right)dz = 0 \quad (34)$$

for given values of r.

For this purpose the following integrals are required

$0 \le z \le c/2$

$$\int_0^z F\left(\frac{z}{c/2}\right)dz = \frac{7}{8}\left(\frac{c}{2}\right)\left(\frac{z}{c/2}\right)^{8/7}\left[1 - \frac{1}{15}\left(\frac{z}{c/2}\right)\right] \quad (35)$$

$$\int_0^z G\left(\frac{z}{c/2}\right)dz = \frac{c}{2}\left[\frac{35}{36}\left(\frac{7}{8}\right)\left(\frac{z}{c/2}\right)^{8/7} - \frac{7}{90}\left(\frac{7}{17}\right)\left(\frac{z}{c/2}\right)^{17/7}\right] \quad (36)$$

$c/2 \le z \le c$

$$\int_0^z F\left(\frac{z}{c/2}\right)dz = \frac{7}{8}\left(\frac{c}{2}\right)\left\{\frac{28}{15} - \left[\left(\frac{c-z}{c/2}\right)^{8/7} - \frac{1}{15}\left(\frac{c-z}{c/2}\right)^{15/7}\right]\right\} \quad (37)$$

$$\int_0^z G\left(\frac{z}{c/2}\right)dz = \frac{c}{2}\left\{\frac{13}{12}\left(\frac{7}{8}\right) - \frac{1}{2}\left(\frac{7}{15}\right) + \frac{7}{18}\left(\frac{7}{16}\right) - \frac{7}{90}\left(\frac{7}{17}\right) + \left(\frac{35}{36}\right)\left(\frac{7}{8}\right) - \frac{7}{90}\left(\frac{7}{17}\right) - \left[\frac{13}{12}\left(\frac{7}{8}\right)\left(\frac{c-z}{c/2}\right)^{8/7} - \left(\frac{1}{2}\right)\left(\frac{7}{15}\right)\left(\frac{c-z}{c/2}\right)^{15/7} + \frac{7}{18}\left(\frac{7}{16}\right)\left(\frac{c-z}{c/2}\right)^{16/7} - \left(\frac{7}{90}\right)\left(\frac{7}{17}\right)\left(\frac{c-z}{c/2}\right)^{17/7}\right]\right\} \quad (38)$$

These integrals have been calculated and are tabulated in Table 2.

Table 2

$\left(\frac{z}{c/2}\right)$	$\int_0^{\frac{z}{c/2}} F\left(\frac{z}{c/2}\right)d\left(\frac{z}{c/2}\right)$	$\int_0^{\frac{z}{c/2}} G\left(\frac{z}{c/2}\right)d\left(\frac{z}{c/2}\right)$
0.0	0.0000	0.0000
0.1	0.0626	0.0611
0.2	0.1372	0.1345
0.4	0.2989	0.2951
0.6	0.4685	0.4652
0.8	0.6419	0.6406
1.0	0.8167	0.8187
1.2	0.9915	0.9979
1.4	1.1648	1.1771
1.6	1.3345	1.3540
1.8	1.4961	1.5245
1.9	1.5708	1.6041
2.0	1.6333	1.6714

Fig. 8 shows the computed results for the positive portion of the range given by equation (29). In order to be consistent with the laminar results, the range given by equation (29) is rewritten in the form

$$-0.385 \le \beta(r)\left|\frac{\rho r\omega c}{\mu}\right|^{1/4} \le 0.328 \quad (39)$$

The product

$$\beta(r)\left|\frac{\rho r\omega c}{\mu}\right|^{1/4}$$

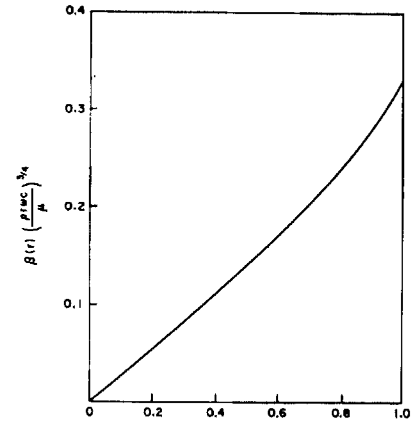

Fig. 8 Turbulent flow $\beta(r)\left(\frac{\rho r\omega c}{\mu}\right)^{1/4}$ versus ξ for $Q > 0$

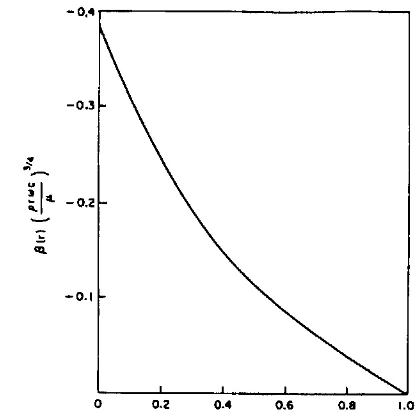

Fig. 9 Turbulent flow $\beta(r)\left(\frac{\rho r\omega c}{\mu}\right)^{1/4}$ versus ξ for $Q < 0$

is plotted in Fig. 9 as a function of $\xi = 1 - \frac{z}{c}$.

Turbulent Net Radial Inflow ($Q < 0$)

The separating stream surface branching from the rotating surface is described by the roots of the equation

$$\int_0^z G\left(\frac{z}{c/2}\right)dz + \left[\left(\frac{0.0195}{9}\right)\left(\frac{36\mu Q}{\pi\rho c^3(r\omega)^2}\right)\left|\frac{\rho r\omega c}{\mu}\right|^{1/4} - \frac{835}{816}\right]\int_0^z F\left(\frac{z}{c/2}\right)dz = 0 \quad (40)$$

The results of this computation are shown in Fig. 9 for the negative portion of the range given by (39).

Turbulent Net Radial Flow Zero ($Q = 0$)

As with laminar flow, the zero net leakage case requires special treatment. The surface $u = 0$ is parallel to the seal surfaces and is located by combining equations (30) and (31) when $Q = u = 0$. The result is

$$0 = G\left(\frac{z}{c/2}\right) - \frac{835}{816}F\left(\frac{z}{c/2}\right) \quad (41)$$

Using the tabulated values of F and G from Table 1 it is easily demonstrated that $\xi_i = 0.45$, very nearly the same location as in the laminar case.

Exchange Flow Rates in Turbulent Flow

The procedure for finding the exchange rates is straightforward. For inflow the distance ξ_i shown in Fig. 7(a) must be found. Since this point also lies on the $u = 0$ surface, it is determined by choosing a value of $\left(\frac{z}{c/2}\right)_i$ which satisfies the expression

$$G\left(\frac{z}{c/2}\right) + \left[\left(\frac{0.0195}{9}\right)\left(\frac{36\mu Q}{\pi\rho c^3(R_1\omega)^2}\right)\left|\frac{\rho R_1\omega c}{\mu}\right|^{1/4} - \frac{835}{816}\right]F\left(\frac{z}{c/2}\right) = 0 \quad (42)$$

using either equations (32) or (33), or Table 1.

As before, the flow rate through a cylindrical area z in height is given by

$$q(r, z) = \int_0^z 2\pi r u dz = \left[\frac{\pi\rho c\left(\frac{r\omega}{2}\right)^2}{\tau_{r\theta_w}}\right] \times \left\{\int_0^z G\left(\frac{z}{c/2}\right)dz + \left[\left(\frac{0.0195}{9}\right)\left(\frac{36\mu Q}{\pi\rho c^3(r\omega)^2}\right)\left|\frac{\rho r\omega c}{\mu}\right|^{1/4} - \frac{835}{816}\right]\int_0^z F\left(\frac{z}{c/2}\right)dz\right\} \quad (43)$$

where

$$\tau_{r\theta_w} = 0.0225\rho(\nu)^{1/4}\left(\frac{r\omega}{2}\right)\left|\frac{\left(\frac{r\omega}{2}\right)^3}{c/2}\right|^{1/4} \quad (44)$$

When $Q > 0$, the exchange rates are computed from equations (43) and (44) with the aid of equations (35–38) or Table 2 setting $r = R_1$ and $\left(\frac{z}{c/2}\right) = \left(\frac{z}{c/2}\right)_i$, i.e.,

$$q_{in} = q(R_1, z_i) \quad (45)$$

$$q_{out} = Q - q_{in}(R_1, Z_i) \quad (46)$$

For $Q > 0$, $\left(\frac{z}{c/2}\right)_i$ is determined by the value of $\frac{z}{c/2}$ which satisfies equation (42). Then the exchange rates are computed with the help of equations (36–38) and (43–44), i.e.,

$$q_{in} = q(R_1, z_i) \quad (47)$$

$$q_{out} = Q - q_{in}(R_1, z_i) \quad (48)$$

When the net leakage rate is zero, the exchange flow rates are equal and they can be described analytically because the separating stream surface is a plane. The inflow is calculated from

$$q_{in} = 2\pi R_1 \int_0^{z_i = 0.55c} u(R_1, z)dz \quad (49)$$

where

$$u(R_1, z) = \left[\frac{\frac{c}{2}\left(\frac{\rho}{R_1}\right)\left(\frac{R_1\omega}{2}\right)^2}{\tau_{r\theta_w}(R_1)}\right]\left[G\left(\frac{z}{c/2}\right) - \frac{835}{816}F\left(\frac{z}{c/2}\right)\right] \quad (50)$$

and

$$\tau_{x\theta_w}(R_1) = 0.0225\rho(\nu)^{1/4}\left(\frac{R_1\omega}{2}\right)\left|\frac{\left(\frac{R_1\omega}{2}\right)^2}{c/2}\right|^{1/4} \quad (51)$$

Carrying out the indicated integration yields the exchange rates

$$q_{in} = -q_{out} = 0.09422(2\pi R_1)c^2\omega\left|\frac{\frac{c}{2}\left(\frac{R_1\omega}{2}\right)}{\nu}\right|^{1/4} \quad (52)$$

Sample Calculation

To illustrate the use of the theory, the following typical seal data will be used

$$P_1 - P_2 = 0.75 \text{ psi}$$
$$c = 0.0005 \text{ in}$$
$$R_1 = 0.75 \text{ in}$$
$$R_2 = 1.00 \text{ in}$$
$$\text{rpm} = 3600$$
$$\mu = 6.24(10^{-6})\ \frac{\text{lb}_m}{\text{ft sec}}$$
$$\rho = 62.4\ \frac{\text{lb}_m}{\text{cu ft}}$$

The computed volumetric leakage rate for laminar flow is given by

$$Q = \frac{\pi c^3}{6\mu \ln\left(\frac{R_2}{R_1}\right)}\left\{P_1 - P_2 + \frac{3}{20}\rho(R_2\omega)^2\left[1 - \left(\frac{R_1}{R_2}\right)^2\right]\right\}$$
$$= 1.255(10^{-7}) \text{ cu ft/sec}$$

The separation radius computed from equation (15) is

$$R_s = \sqrt[4]{\frac{45\mu Q}{\rho\omega^2 c^3\pi}} = 0.5 \text{ in}$$

so that there is an exchange of inner and outer fluid. At $r = R_1$

$$\beta(R_1) = \frac{\mu Q}{2\rho(R_1\omega)^2 c^3\pi} = 4.95(10^{-3})$$

From equation (21)

$$\xi_i = 1 - \frac{1}{2}\left[\sqrt{1 + 4\left(\frac{4}{5} - 72\beta(R_1)\right)} - 1\right]$$
$$= 0.665$$

so that

$$U(0.665) = 2\xi^3 - 3\xi^2 = -0.738$$

$$W(0.665) = \frac{1}{60}(-3\xi^5 + 7\xi^4 - 5\xi^3 + \xi^2) = -1.9(10^{-3})$$

Using equations (7) and (22), the outflow exchange rate is

$$q_{out} = -2\pi R_1\psi(R_1, \xi_i)$$
$$= -2\pi R_1(cR_1\omega)R_eW(\xi_i) - QU(\xi_i)$$
$$= 1.41(10^{-7}) \text{ cu ft/sec}$$

The inflow exchange rate is then

$$q_{in} = Q - q_{out} = -0.155(10^{-7}) \text{ cu ft/sec}$$

Conclusion

The means of determining whether reversed flow conditions result in an exchange of inner and outer fluids have been developed. When there is an exchange, a technique for dividing the net flow into individual exchange rates is presented. Analyses are given which are applicable to both laminar and turbulent flow when both the inner and outer boundaries are immersed in the same fluid.

Because the special case of zero net leakage ($Q = 0$) results in a stream surface parallel to the seal surfaces, the equations for the exchange rates can be deduced without reference to the other specifics of the seal operation. These results are offered as one possible explanation of the so-called "inward pumping effect" observed in laboratory tests on faces whose boundaries are pressurized hydrostatically. Such behavior has been observed when the inner and outer fluids are different [5] so the analysis given here does not strictly apply. The criterion given by equations (3) and (39) can be used, however, to design a face seal so as to avoid any tendency toward reversed flow and insure unidirectional leakage in the two fluid case.

Acknowledgment

The work reported here was carried out at the General Electric Research and Development Center under NASA Contract NAS 7-434. This work was performed for the Jet Propulsion Laboratory. R. S. Weiner was technical manager for JPL and Frank Compitello was project manager for NASA headquarters. Numerical computations were carried out by Miss Judy Reilly of the Research and Development Center.

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APPENDIX

An equation for the radial velocity distribution is given in [3] which contains the zeroth-order perturbation terms and uses the midplane as the origin for the axial coordinate (Z). When that equation ((29) of reference [3]) is compared with the radial velocity equation of reference [1] they are found to be identical, taking the form in the notation of [3]

$$\bar{u} = \left(\frac{\bar{r}\omega^2 a^2}{240\nu}\right)[-1 + 20Z + 6Z^2 - 20Z^3 - 5Z^4]$$
$$+ \frac{3}{8}\left(\frac{Q}{\pi\bar{r}a}\right)(1 - Z^2)$$

This result was obtained by setting $K = 0$, which in the notation of [3] means that the lower surface is not rotating.

The transformation of the tangential velocity of reference [1] into the notation of [3] yields

$$\frac{\bar{v}}{\bar{r}\omega} = \frac{Z + 1}{2}$$

which is the form taken in the limit by equation (30) of (3) as the radius to clearance ratio becomes larger and $K = 0$. As

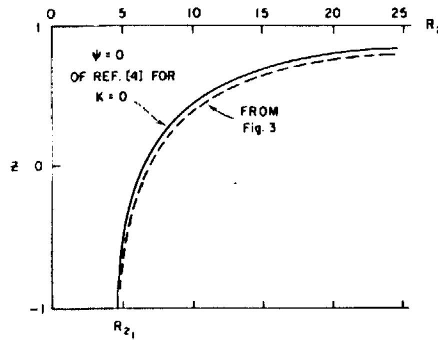

Fig. 10 Comparison with separation streamline of reference [4] ($Q > 0$)

pointed out in [3], "the angular velocity profile will differ very little from a linear profile and the difference vanishes as $\bar{r}/a$ increases."

The pressure distribution given in [3] for $K = 0$ can be written in the form

$$p = \frac{3}{20}\rho(\bar{r}\omega)^2\left[1 + \frac{1}{18}\left(\frac{a}{\bar{r}}\right)^2(1 - Z^2)(1 - 5Z^2 - 20Z)\right]$$

$$-\frac{3Q\mu}{4\pi a^3}\left[1 + \frac{1}{3}\left(\frac{263}{525}\right)\left(\frac{\omega_0 a^2}{\nu}\right)^2\right]\ln\left(\frac{\bar{r}}{a}\right) + \text{constant}$$

In seal-like configurations where $\bar{r}/a$ is extremely large, the coefficient of the first term is very nearly unity.

In the notation of this paper an [1]

$$\left(\frac{\omega_0 a^2}{\nu}\right)^2 = \frac{1}{16}\left(\frac{c^2\omega}{\nu}\right)^2$$

It was estimated in reference [1] that $\frac{c^2\omega}{\nu}$ could be as large as 10 without seriously altering the coefficient of the second term. If instead $\left(\frac{c^2\omega}{\nu}\right)^2$ is restricted to less than 10 for seal-like configurations, then the two analyses yield pressure distributions which are in close agreement.

One final comparison is possible. The parameter R_2 (not a radius) defined in [3] is related to the analysis of this paper by the formula

$$R_2^2 = \frac{1}{4\beta(r)}$$

The separation point radius for $K = 0$ was shown in [3] to be equivalent to $R_{2_1} = R_2 = 3\sqrt{5}/2$ or $\beta(r) = 1/90$ for radial net outflow. A plot of the separating stream surface taken from reference [3] [Fig. 5(a)] is shown in Fig. 10 as a solid line. The same surface, as predicted from Fig. 3 of this paper, is drawn as a dashed line in Fig. 11 showing obviously good agreement.

III. J

Paper No.
68—Lub-14

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Thermal Effects in Face Seals

The laminar and turbulent flow of an incompressible fluid between the rotating parallel surfaces of a face seal is investigated analytically to determine the influence of conduction, convection, and dissipation on the temperature distribution. A method of estimating the general temperature level within the seal is suggested.

Introduction

The analyses of face seals and thrust bearings are often based on the assumption that the flow is isothermal within the seal clearance. The validity of this approximation is usually argued on the basis that the seal faces are often very good conductors and so will not permit large radial temperature gradients.

Even when the seal operates approximately isothermally, the temperature within the seal clearance need not necessarily be the same as the ingested sealant temperature. An accurate prediction of a seals performance requires an accurate evaluation of the fluid viscosity within the clearance space. The key to this evaluation is the prediction of the clearance temperature level which is the subject of the investigation to follow.

Analysis of Laminar Flow

Because of its simplicity, the flat, aligned face seal will be analyzed. It has been shown [1][1] that the radial and tangential velocity profiles for such a seal are given by the formulas

$$u = \left(\frac{z^2 - cz}{2\mu}\right)\frac{\partial p}{\partial r} - \frac{\rho r\omega^2}{12\mu c^2}(z^4 - zc^3) \tag{1}$$

$$v = r\omega\left(\frac{z}{c}\right) \tag{2}$$

where, because of the thinness of the fluid film, it has been assumed that the temperature variation across the clearance does not cause a significant change in the viscosity or density in the axial direction.

The volumetric leakage rate is given by

$$Q = 2\pi r\int_0^c u\,dz \tag{3}$$

Carrying out the indicated integration with the aid of equation (1) yields for the pressure gradient

$$\frac{dp}{dr} = \frac{3}{10}\rho r\omega^2 - \frac{6\mu Q}{\pi rc^3} \tag{4}$$

[1] Numbers in brackets designate References at end of paper.

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The appropriate form of the energy equation for laminar flow through a face seal is

$$\frac{\partial}{\partial z}\left(-k\frac{\partial T}{\partial z}\right) + \rho C_p u\frac{\partial T}{dr} = u\frac{dp}{dr} + \mu\left[\left(\frac{\partial u}{\partial z}\right)^2 + \left(\frac{\partial v}{\partial z}\right)^2\right] \tag{5}$$

The integration of this equation across the clearance can be performed under the same restrictions as imposed on equation (4), i.e., ρ and μ are not functions of z. The various terms in equation (5) will be taken one by one.

The first term in equation (5) when integrated across the clearance represents the local energy flux transferred out of the fluid to the seal walls. Symbolically this flux can be represented by

$$q_w'' = \int_0^c \frac{\partial}{\partial z}\left(-k\frac{\partial T}{\partial z}\right)dz = \left(-k\frac{\partial T}{\partial z}\right)_c - \left(-k\frac{\partial T}{\partial z}\right)_0 \tag{6}$$

The second and third term associated with the radial transport of energy, when integrated, take the form

$$\left(\rho C_p\frac{dT}{dr} - \frac{dp}{dr}\right)\int_0^c u\,dz \tag{7}$$

Here it is once again assumed that the temperature does not vary significantly in the axial direction. This does conflict with the assumption that there is a significant temperature gradient in the axial direction which results from small temperature changes over extremely small axial distances (equation (6)). Equations (3) and (4) yield

$$\int_0^c u\,dz = \frac{Q}{2\pi r} = \frac{c^3}{12\mu}\left[\frac{3}{10}\rho r\omega^2 - \frac{dp}{dr}\right] \tag{8}$$

for the integral coefficient in equation (7).

The last term in equation represents the energy converted by viscous dissipation within the liquid film. The radial velocity contribution to the dissipation is

$$\mu\int_0^c\left(\frac{\partial u}{\partial z}\right)^2 dz = \frac{c^3}{12\mu}\left(\frac{dp}{dr}\right)^2 - \frac{c^2}{20\mu}(\rho r\omega^2)\frac{dp}{dr} + \frac{1}{7}\left(\frac{c^3}{\mu}\right)\left(\frac{\rho r\omega^2}{4}\right)^2 \tag{9}$$

Nomenclature

- c = seal clearance
- C_p = fluid specific heat
- k = fluid thermal conductivity
- p = fluid pressure
- Q = volumetric leakage rate
- q_w'' = total local heat transfer flux from seal
- q_w = total heat transfer from seal
- r = radial coordinate
- R_1 = inner seal radius
- R_2 = outer seal radius
- T = temperature
- T_1 = temperature at inner radius
- T_2 = temperature at outer radius
- u = radial velocity component
- v = tangential velocity component
- z = axial coordinate
- μ = viscosity
- ν = μ/ρ
- ρ = density
- ω = angular velocity of seal surface
- τ_{zr} = radial shear stress
- $\tau_{z\theta_w}$ = tangential shear stress at the wall

The tangential velocity contribution to the dissipation is given by

$$\mu\int_0^c\left(\frac{\partial v}{\partial z}\right)^2 dz = \frac{\mu}{c}(r\omega)^2 \tag{10}$$

Integrating equation (5) across the clearance and making the substitutions indicated in the foregoing reduces the energy equation to the following form

$$q_w'' + \rho C_p\left(\frac{Q}{2\pi r}\right)\frac{dT}{dr} = -\frac{c^2}{40\mu}(\rho r\omega^2)\frac{dp}{dr} + \frac{c^3}{7\mu}\left(\frac{\rho r\omega^2}{4}\right)^2 + \frac{\mu}{c}(r\omega)^2 \tag{11}$$

In order to integrate this equation in the radial direction it will be necessary to assume that ρ and μ are independent of the radial coordinate. Since neither of these properties is strongly pressure dependent and only viscosity is significantly influenced by temperature variations, this assumption is consistent with the notion that radial temperature change across the seal is not large.

When integrated in the radial direction, equation (11) becomes

$$\frac{1}{Q}\int_{R_1}^{R_2} 2\pi r q_w'' dr + \rho C_p(T_2 - T_1) = \frac{3}{20}\rho\omega^2(R_2^3 - R_1^3) + \left[\frac{2\pi\mu\omega^2}{Qc} + \left(\frac{9}{100}\right)\frac{\pi c^3(\rho\omega^2)^2}{\mu Q}\right]\left(\frac{R_2^4 - R_1^4}{4}\right) \tag{12}$$

The numerator of the first term is the total energy exchange

$$q_w = \int_{R_1}^{R_2} 2\pi r q_w'' dr \tag{13}$$

Using this definition the energy equation can be written in the following form

$$q_w/Q + \rho C_p(T_2 - T_1) = \frac{3}{5}\rho\left(\frac{R_2\omega}{2}\right)^2\left[1 - \left(\frac{R_1}{R_2}\right)^2\right] \times \left\{1 + \frac{3}{20}\left[\frac{\rho R_2\omega^2}{\left(\frac{\mu Q}{\pi c^3 R_2}\right)}\right]\left[1 + \frac{200}{9}\left(\frac{\mu}{\rho\omega c^2}\right)^2\right] \times \left[1 + \left(\frac{R_1}{R_2}\right)^2\right]\right\} \tag{14}$$

where the enthalpic and kinetic energy transport terms are easily identified.

Analysis of Turbulent Flow

The basic difference between the laminar and turbulent flow analyses is the way in which the viscous dissipation term in the energy equation is formulated. The general form of this term in dyadic notation is

$$\tau:\nabla V \tag{15}$$

where τ is the stress dyadic and V is the velocity vector. For face seal flow this quantity may be approximated by

$$\tau:\nabla V \cong \tau_{z\theta}\left(\frac{\partial u}{\partial z}\right) + \tau_{zr}\left(\frac{\partial v}{\partial z}\right) \tag{16}$$

When the flow is laminar

$$\tau:\nabla V \cong \mu\left[\left(\frac{\partial u}{\partial z}\right)^2 + \left(\frac{\partial v}{\partial z}\right)^2\right] \tag{17}$$

which appears in equation (5). For isotropic turbulent flow it has been shown [2] that

$$\tau_{zr} = \tau_{z\theta}\left(\frac{\partial u}{dz}\Big/\frac{\partial v}{\partial z}\right) \tag{18}$$

which means that

$$\tau:\nabla V = \tau_{z\theta}\left[\frac{\partial v}{\partial z} + \left(\frac{\partial u}{\partial z}\right)^2\Big/\frac{\partial v}{\partial z}\right] \tag{19}$$

where

$$\tau_{z\theta} = \tau_{z\theta_w} = 0.0225\rho(\nu)^{1/4}\left(\frac{r\omega}{2}\right)\left|\frac{\left(\frac{r\omega}{2}\right)^3}{c/2}\right|^{1/4} \tag{20}$$

Following the same procedure used in the laminar analysis, the integrated form of the energy equation for turbulent flow is

$$q_w'' + \rho C_p\frac{\partial T}{\partial r}\int_0^c u\,dz = r\omega\tau_{z\theta_w} + \frac{\rho}{r}\int_0^c u(v)^2 dz \tag{21}$$

With the aid of equation (3) the working form of the energy equation can be written as

$$q_w'' + \frac{\rho C_p Q}{2\pi r}\frac{\partial T}{dr} = r\omega\tau_{z\theta_w} + \frac{\rho}{r}\int_0^c u(v)^2 dz \tag{22}$$

The velocity distributions to be used when evaluating the last term of this equation will be the same as used in [2], i.e.,

$$u\tau_{z\theta_w} = -\left(\frac{r\omega}{2}\right)\left(\frac{c}{2}\right)\frac{dp}{dr}F\left(\frac{z}{c/2}\right) + \frac{\rho}{r}\left(\frac{r\omega}{2}\right)^3\left(\frac{c}{2}\right)G\left(\frac{z}{c/2}\right) \tag{23}$$

$$v_1 = \left(\frac{rw}{2}\right)\left(\frac{z}{c/2}\right)^{1/7}, \quad 0 \le z \le c/2$$

$$v_2 = rw\left[1 - \frac{1}{2}\left(\frac{c - z}{c/2}\right)^{1/7}\right], \quad c/2 \le z \le c \tag{24}$$

The pressure gradient from the same reference is

$$\frac{dp}{dr} = \frac{835}{816}\rho r\left(\frac{\omega}{2}\right)^2 - \frac{0.0195}{\pi}\left|\frac{\rho^3\mu\omega^3}{rc^9}\right|^{1/4} Q \tag{25}$$

The evaluation of the last term in equation (22) leads to

$$\tau_{z\theta_w}\left(\frac{\rho}{r}\right)\int_0^c u(v)^2 dz = -1.2090\frac{\rho}{r}\left(\frac{r\omega}{2}\right)^3\left(\frac{c}{2}\right)^2 \times \left[\frac{835}{816}\rho r\left(\frac{\omega}{2}\right)^2 - \frac{0.0195}{\pi}\left|\frac{\rho^3\mu\omega^3}{rc^9}\right|^{1/4} Q\right] + 1.7228\left(\frac{\rho}{r}\right)^2\left(\frac{r\omega}{2}\right)^6\left(\frac{c}{2}\right) \tag{26}$$

or

$$\frac{\rho}{r}\int_0^c u(v)^2 dz = \left\{\frac{1}{0.0225\rho(\nu)^{1/4}\frac{\omega}{2}\left|\frac{\omega^3}{c/2}\right|^{1/4}}\right\} \times \left\{0.48568\left(\frac{c\rho}{2}\right)^2\left(\frac{\omega}{2}\right)^5 r^{5/4} + 1.2090\left(\frac{0.0195}{\pi}\right)\left|\frac{\rho^3\mu\omega^3}{c^9}\right|^{1/4}\rho Q\left(\frac{\omega}{2}\right)^2\left(\frac{c}{2}\right)^2\right\} \tag{27}$$

It should be observed that this rather complicated expression is really a simple function of the radius of the form

$$\frac{\rho}{r}\int_0^c u(v)^2 dz = Ar^{5/4} + B \tag{28}$$

Returning to equation (22) and integrating with respect to r yields

$$\int_{R_1}^{R_2} 2\pi r q_w'' dr + \rho Q C_p \int_{R_1}^{R_2} \frac{\partial T}{\partial r} dr$$

$$= 0.0225\rho\,(\nu)^{1/4} \frac{\omega^2}{2} \left| \frac{\left(\frac{\omega}{2}\right)^2}{c/2} \right|^{1/4} \int_{R_1}^{R_2} 2\pi r^{15/4} dr$$

$$+ 2\pi \int_{R_1}^{R_2} A r^{9/4} dr + 2\pi \int_{R_1}^{R_2} B r dr \quad (29)$$

Carrying out the integration, assuming that A and B (i.e., ρ and μ) are independent of the radius, yields

$$q_w + \rho Q C_p (T_2 - T_1) = 0.0225\rho(\nu)^{1/4}(\pi\omega^2) \times \left| \frac{\left(\frac{\omega}{2}\right)^2}{c/2} \right|^{1/4} \left(\frac{4}{19}\right)(R_2^{19/4} - R_1^{19/4})$$

$$+ 2\pi A \left(\frac{4}{3}\right)(R_2^{13/4} - R_1^{13/4}) + 2\pi B \left(\frac{1}{2}\right)(R_2^2 - R_1^2) \quad (30)$$

Rearranging and replacing A and B with their earlier representations gives the final form

$$\frac{q}{Q_w} + \rho C_p (T_2 - T_1) = \frac{2}{19}\left(\frac{2\pi}{Q}\right) R_2^2 (R_2\omega)^2 \times \left[1 - \left(\frac{R_1}{R_2}\right)^{19/4}\right] \left\{ 0.0225\rho(\nu)^{1/4} \left| \frac{\left(\frac{R_2\omega}{2}\right)^2}{c/2} \right|^{1/4} \right\}$$

$$+ \frac{2\pi}{Q}\left(\frac{4}{13}\right)\left[1 - \left(\frac{R_1}{R_2}\right)^{13/4}\right] \left\{ \frac{0.48568\left(\frac{\rho c}{2}\right)^2 \left(\frac{R_2\omega}{2}\right)^4}{0.0225\rho(\nu)^{1/4} \left| \frac{\left(\frac{R_2\omega}{2}\right)^2}{c/2} \right|^{1/4}} \right\}$$

$$+ \left[1 - \left(\frac{R_1}{R_2}\right)^2\right] \left\{ \frac{0.0195(1.2090) \left| \frac{\rho^3\mu\omega^3}{R_2 c^5} \right|^{1/4} \rho R_2 \left(\frac{R_2\omega}{2}\right)^2 \left(\frac{c}{2}\right)^2}{0.0225\rho(\nu)^{1/4} \left| \frac{\left(\frac{R_2\omega}{2}\right)^2}{c/2} \right|^{1/4}} \right\} \quad (31)$$

Discussion of Results

If the flow-rate Q is set equal to zero equations (14) and (31) indicate that all the heat energy is carried away through the seal surfaces. Actually the situation is somewhat more complicated than that since it has been established from hydrodynamic dynamic considerations that zero net flow results in equal but opposite counterflows within the seal [1, 2]. In the general case this counterflow should cause the radial temperature distribution to be very flat except near the inner and outer boundaries. There the opposing stream's temperatures will undergo almost a step change from the seal temperature to the local ambient temperature and conversely. Because these streams are very thin and slow moving, equations (14) and (31) predict that energy exchanged between the opposing streams is conserved.

When the inner and outer reservoirs of sealant are finite and there is no replenishment of the fluid, both reservoirs will ultimately attain the same temperature in steady state and the energy exchange between opposing streams vanishes.

The temperature level within the clearance space for $Q = 0$ is determined by the equations (14) and (31) where the left hand side is evaluated by performing an analysis of the heat transfer capabilities of the seal itself, i.e., determining the ability of the seal to carry heat energy away from the clearance space by conduction through the seal face.

When there is no reversed flow-heat, energy can be transported from the clearance space by convection. Because this case is both complex and typical, the following numerical examples are offered to illustrate the use of the analysis presented above.

Sample Calculation

Using water as a sealant, let

$$P_1 - P_2 = 100 \text{ psi}$$
$$c = 0.0005 \text{ in.}$$
$$R_1 = 0.75 \text{ in.}$$
$$R_2 = 1.00 \text{ in.}$$
$$\text{rpm} = 3600$$
$$\mu = 6.24(10^{-4}) \frac{\text{lbm}}{\text{ft-sec}}$$
$$\rho = 62.4 \frac{\text{lb}_m}{\text{ft}^3}$$
$$C_p = 1.0 \frac{\text{Btu}}{\text{lb}_m \text{deg F}}$$

The volumetric leakage rate for laminar flow is obtained from the radial integration of (4) so that

$$Q = \frac{\pi c^3}{6\mu \ln R_2/R_1}\left\{P_1 - P_2 + \frac{3}{20}\rho(R_2\omega)^2\left[1 - \left(\frac{R_1}{R_2}\right)^2\right]\right\} = 0.977(10^{-4}) \text{ ft}^3/\text{sec}$$

An evaluation of the right-hand side of equation (14) yields

$$q_w + \rho C_p Q(T_2 - T_1) = 3.78 \frac{\text{ft lb}_f}{\text{sec}}$$

If it is assumed that the seal walls are perfect insulators ($q_w = 0$) then the temperature rise required to convect away the dissipated energy is

$$T_2 - T_1 = \frac{3.78}{\rho Q C_p} = 0.796 \text{ deg F}$$

Seal faces are rarely made from good insulators so that the radial temperature rise computed above is well in excess of the expected value. At the other extreme it could be assumed that $T_1 = T_2$. Taking the seal face to be made of graphite (thermal conductivity $= 80 \frac{\text{Btu-ft}}{\text{ft}^2 \text{ deg F hr}}$) which is $^1/_4$ in. thick, the axial temperature drop ΔT required across the seal material to conduct away the dissipated heat energy is 0.238 deg F.

The obvious conclusion here is that this seal operates almost isothermally and very near to ambient temperature surrounding the seal. If instead of water an oil with a viscosity 100 times larger than water is used as a sealant, the right-hand side of equation is approximately 100 times larger than the value for water. The adiabatic temperature rise $T_2 - T_1$ is 10,000 times the rise for water, which is of course a very unrealistic result. On the other hand, the axial temperature drop across the seal ΔT required to conduct away the dissipated energy when $T_1 = T_2$ is only about 24 deg F, which is not at all an unreasonable value. This is not to say that there is no radial temperature rise across the seal, only that no significant amount of energy is carried away by convection.

If $T_2 - T_1$ is chosen to be the order of magnitude of ΔT and

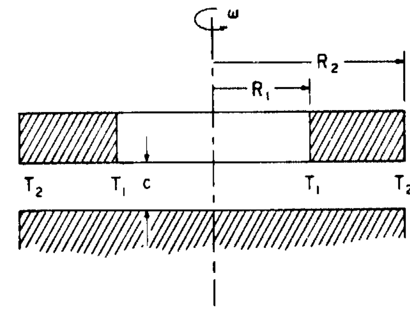

Fig. 1

the initial isothermal approximation is relaxed, it will have no effect on the previous conclusions concerning the water and oil seal operation, i.e., the water seal operates at about ambient temperature, and the dissipated energy in the oil seal is carried away almost exclusively by conduction.

In a practical sense the designer might well follow the procedures outlined above for $T_1 = T_2$ in order to establish an average temperature level within the seal clearance. This temperature could then be used to adjust initial assumptions for ρ and μ and at the same time determine the magnitude of $Qc_p(T_2 - T_1)$. An iterative calculation could then be carried forward to yield more refined predictions. It is clear from the analysis and the sample calculation that no general thermal analysis of the sealant can be made which does not involve the seal itself.

Conclusion

A general thermal analysis of the sealant flow has been presented which is restricted only by the assumption that the radial variation of temperature is not so large as to cause a significant viscosity change within the seal. Since seals do not ordinarily have a large radial span this restriction is not a severe one.

If the specifics of the seal configuration are known, as well as the ambient temperatures, the analysis can be used to estimate the general operating temperature level and energy budget for the seal. These quantities can then be used to obtain more accurate predictions of the expected seal performance.

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The Eccentric Face Seal with A Tangentially Varying Film Thickness

III. K

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It is shown analytically that face seal eccentricity combined with surface waviness can contribute an inward or outward leakage component to the total leakage flow depending on the phase angle between the eccentricity and the waviness.

INTRODUCTION

Previous investigations of the effects of geometry on the performance of face seals have considered such geometric aberrations as misalignment, waviness in the tangential direction, and film thickness variations in the radial direction (*1*) (*2*) (*3*). In each case, however, it was assumed that the face seals themselves were centered on a common centerline. It has been shown by Findlay (*4*) that the anomolous pumping effects occasionally observed in face seals could be attributed to the eccentricity of the seal faces which occurs when their centers are not on a common centerline. Preliminary analytical and experimental studies of the effects of such eccentricities indicate that the proposed pumping effect is indeed initiated by the introduction of an eccentricity (*4*). In the investigation to follow the effects of tangential waviness combined with eccentricity will be investigated analytically in order to predict their influence on face seal performance.

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ANALYSIS OF AN ALIGNED WAVY SEAL WITH ECCENTRICITY—LAMINAR FLOW

Momentum Equations

For seal-like configuration it was shown in (*1*) that the momentum equations can be reduced to

$$\frac{\partial p}{\partial r} = \rho \frac{v^2}{r} + \frac{\partial \tau_{zr}}{\partial z} \quad [1]$$

$$\frac{\partial \tau_{z\theta}}{\partial z} = 0 \quad [2]$$

$$\frac{\partial p}{\partial z} = 0 \quad [3]$$

If the flow is laminar these equations take the form

$$\frac{\partial p}{\partial r} = \rho \frac{v^2}{r} + \mu \frac{\partial^2 u}{\partial z^2} \quad [4]$$

$$\frac{\partial p}{\partial z} = \frac{\partial^2 v}{\partial z^2} = 0 \quad [5]$$

Velocity Distribution

For the geometry shown in Fig. 1 the vectorial velocity of a point on the flat surface which rotates about 0′ is given by

$$\overline{V} \cong \Omega\{[\epsilon \sin(\phi - \theta)]\bar{e}_r + r\bar{e}_\theta\} \quad [6]$$

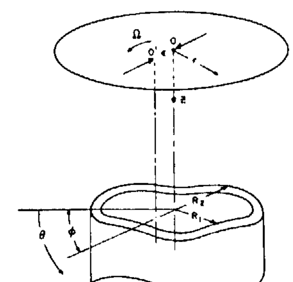

Fig. 1—Seal geometry for a wavy surface and radial eccentricity.

From Eqs. [5] and [6] the tangential velocity can be shown to be

$$v = r\Omega\left(1 - \frac{z}{h}\right) \quad [7]$$

where h is a function of θ only. When this equation is combined with Eq. [4] and integrated the result is

$$u = \frac{z^2}{2\mu}\frac{\partial p}{\partial r} - \frac{\rho r \Omega^2}{\mu}\left[\frac{z^2}{2} - \frac{z^3}{3h} + \frac{1}{12}\frac{z^4}{h^2}\right] + F(r,\theta)z + G(r,\theta) \quad [8]$$

The boundary conditions to be applied to this equation are

$$u = \Omega\epsilon \sin(\phi - \theta),\ z = 0$$
$$u = 0,\ z = h$$

so that the radial velocity distribution finally takes the form

$$u = \frac{1}{2\mu}(z^2 - hz)\frac{\partial p}{\partial r} - \frac{\rho r\Omega^2}{\mu}\left[-\frac{zh}{4} + \frac{z^2}{2} - \frac{z^3}{3h} + \frac{1}{12}\frac{z^4}{h^2}\right] + \Omega\epsilon\sin(\phi - \theta)\left[1 - \frac{z}{h}\right] \quad [10]$$

Pressure Distribution

The integrated form of the continuity equation for this case is

$$\int_0^h \frac{1}{r}\frac{\partial}{\partial r}(ur)\,dz + \int_0^h \frac{1}{r}\frac{\partial v}{\partial \theta}\,dz + \int_0^h \frac{\partial w}{\partial z}\,dz = 0 \quad [11]$$

Using the boundary conditions the $w(0) = w(h) = 0$, this equation reduces to

$$\frac{\partial}{\partial r}\left[r\int_0^h u\,dz\right] + \int_0^h \frac{\partial v}{\partial \theta}\,dz = 0 \quad [12]$$

The second term in Eq. [12] can be simplied as follows

$$\frac{\partial}{\partial \theta}\int_0^h v\,dz = \int_0^h \frac{\partial v}{\partial \theta}\,dz + v(h)\frac{\partial h}{\partial \theta} \quad [13]$$

with the aid of Leibniz Rule. Since $v(h) = 0$

$$\int_0^h \frac{\partial v}{\partial \theta}\,dz = \frac{\partial}{\partial \theta}\int_0^h v\,dz \quad [14]$$

This simplification, along with the velocity distributions given by Eqs. [7] and [8], result in a partial differential equation for the pressure distribution

$$\frac{\partial}{\partial r}\left(\frac{rh^3}{12\mu}\frac{\partial p}{\partial r}\right) = \frac{3}{5}\left(\frac{\rho\Omega^2 h^3 r}{12\mu}\right) + \frac{1}{2}\epsilon\Omega h\sin(\phi - \theta) + \frac{r\Omega}{2}\frac{\partial h}{\partial \theta} \quad [15]$$

Recalling that h is a function of θ only, Eq. [15] may be easily integrated to yield

$$p = \frac{3}{20}\rho r^2\Omega^2 + \left(\frac{6\mu\Omega\epsilon r}{h^2}\right)\sin(\phi - \theta) + \left(\frac{3\mu r^2\Omega}{2h^3}\right)\frac{\partial h}{\partial \theta} - \frac{12\mu}{h^3}f(\theta)\ln r + g(\theta) \quad [16]$$

The arbitrary functions of integration are determined using the pressure boundary conditions

$$p = P_1,\ r = R_1$$
$$p = P_2,\ r = R_2 \quad [17]$$

The pressure distribution satisfying these conditions is given by

$$p - P_2 = \frac{3}{20}\rho\Omega^2(r^2 - R_2^2) + \left(\frac{6\mu\Omega\epsilon}{h^2}\right)\sin(\phi - \theta)(r - R_2)$$

NOMENCLATURE

- a = ratio of the waviness amplitude to the nominal clearance
- c = nominal clearance
- $\bar{e}_r, \bar{e}_\theta$ = unit vectors
- F = total force in clearance space tending to separate the seal surfaces
- h = local film thickness
- k = an integer
- p = pressure in clearance space
- P_1 = pressure at the inner radius of the seal
- P_2 = pressure at the outer radius of the seal
- Q = volumetric leakage rate
- r = radial coordinate
- R_1 = inside radius of seal
- R_2 = outside radius of seal
- T = total torque on seal face
- u = temporal mean radial velocity component
- v = temporal mean tangential velocity component
- $\overline{V}$ = seal surface velocity vector
- w = temporal mean axial velocity component
- z = axial coordinate
- ϵ = seal eccentricity
- θ = tangential coordinate
- μ = fluid viscosity
- $\nu = \mu/\rho$
- ρ = fluid density
- τ = shear stress
- ϕ = phase angle between eccentricity line of centers and waviness reference line
- Ω = seal surface angular velocity

$$+\frac{3\mu\Omega}{2h^3}(r^2 - R_2^2)\frac{\partial h}{\partial \theta}$$

$$+\frac{\ln\left(\frac{r}{R_2}\right)}{\ln\left(\frac{R_1}{R_2}\right)}\left\{P_1 - P_2 - \frac{3}{20}\rho\Omega^2(R_1^2 - R_2^2)\right.$$

$$-\frac{3\mu\Omega}{2h^3}(R_1^2 - R_2^2)\frac{\partial h}{\partial \theta}$$

$$\left. - \left(\frac{6\mu\epsilon\Omega}{h^2}\right)\sin(\phi - \theta)(R_2 - R_1)\right\} \quad [18]$$

Leakage Flow

The net leakage-rate is computed from the equation

$$Q = \int_0^{2\pi}\int_0^h u\,dz\,r\,d\theta \quad [19]$$

Performing the first integration with the aid of Eq. [10] gives

$$Q = \int_0^{2\pi}\left[-\frac{r\Omega}{4}\frac{\partial h}{\partial \theta} + \frac{f(\theta)}{r}\right]r\,d\theta = \int_0^{2\pi} f(\theta)\,d\theta$$

$$= \frac{1}{12\,\mu\ln(R_2/R_1)}\int_0^{2\pi}\left\{\left[P_1 - P_2 + \frac{3}{20}\rho\Omega^2(R_2^2 - R_1^2)\right]h^3 + 6\,\mu\Omega\epsilon h\sin(\phi - \theta)(R_2 - R_1)\right\}d\theta \quad [20]$$

If it is now assumed that the surface waviness is approximated by the film thickness equation

$$h = c(1 + a\cos k\theta) \quad [21]$$

the formula for the leakage rate is

$$Q = \frac{\pi}{6\,\mu\ln(R_2/R_1)}\left\{c^3\left[P_1 - P_2 + \frac{3}{20}\rho\Omega^2(R_2^2 - R_1^2)\right]\left(1 + \frac{3}{2}a^2\right) + \frac{6\,\mu\Omega\epsilon\,ca(R_2 - R_1)\sin\phi}{2\pi}\int_0^{2\pi}\cos k\theta\cos\theta\,d\theta\right\} \quad [22]$$

The total leakage rate consists of two parts, one due to the waviness alone and one due to eccentricity and waviness together. The waviness contribution is same as found earlier (*1*). The eccentricity contribution is seen to exist only if there is a "once-per-revolution" ($k = 1$) distortion of the type observed Nau (*3*). Then Eq. [22] becomes

$$Q = \frac{\pi}{6\,\mu\ln(R_2/R_1)}\left\{c^3\left[P_1 - P_2 + \frac{3}{20}\rho\Omega^2(R_2^2 - R_1^2)\right]\left(1 + \frac{3}{2}a^2\right)\right.$$

$$\left. + 3\,\mu\Omega\epsilon(R_2 - R_1)\,ca\sin\phi\right\};\ k = 1 \quad [23]$$

Whether the eccentricity causes inward or outward pumping depends on

i) the direction of seal rotation
ii) the phase angle of the waviness relative to the line of centers.

Since it is quite difficult to determine the phase angle ϕ, it is not surprising to find reports of both inward and outward pumping effects exhibited by the same seal. A very logical explanation is a shift in ϕ between disassembly and reassembly, or a change in rotational direction.

Torque

Since the tangential velocity distribution is the same as the one used earlier (*1*) the torque is uneffected by eccentricity, being given by the formula

$$T = \frac{\pi\mu\Omega}{2c}(R_2^4 - R_1^4)\left[1 + \frac{a^2}{2}\right] \quad [24]$$

neglecting terms of order a^4.

Separating Force

The separating force is computed from the equation

$$F = \int_0^{2\pi}\int_{R_1}^{R_2} pr\,dr\,d\theta \quad [25]$$

using the pressure distribution given by Eq. [18]. It will be noted that integration with respect to θ will eliminate terms containing $\frac{1}{h^3}\frac{\partial h}{\partial \theta}$ if the film thickness is described by Eq. [21]. The remaining terms fall into two categories, those which contain ϵ and those which do not. The terms which do not contain ϵ do not contain h either, so their contribution to the separating force is the same as a concentric, aligned, flat-surface seal.

The eccentricity effects the separating force through the terms containing $\frac{1}{h^2}\sin(\phi - \theta)$. This expression can be expanded using Eq. (21) as follows

$$\frac{1}{h^2}\sin(\phi - \theta) = \frac{1}{c^2}(\sin\phi\cos\theta - \sin\theta\cos\phi)$$

$$(1 - 2a\cos k\theta + 3a^2\cos^2 k\theta - \ldots) \quad [26]$$

If $k = 1.0$

$$\int_0^{2\pi}\frac{\sin(\phi - \theta)}{h^2}\,d\theta \cong -\frac{2a\sin\phi}{c^2}\int_0^{2\pi}\cos^2\theta\,d\theta$$

$$= -\frac{2a\pi\sin\phi}{c^2} \quad [27]$$

The total separating force using this approximation (for $k = 1$) is

$$F = F_0 + F_\epsilon \cong \pi P_1(R_2^2 - R_1^2) + \frac{\pi}{2}\left(\frac{3\rho\Omega^2}{20}\right)(R_2^2 - R_1^2)^2$$

$$+\frac{2\pi}{\ln(R_2/R_1)}\left[P_2 - P_1 - \left(\frac{3\rho\Omega^2}{20}\right)(R_2^2 - R_1^2)\right]$$

$$\times\left[\frac{R_2^2}{2}\ln(R_2/R_1) - \frac{1}{4}(R_2^2 - R_1^2)\right]$$

$$+\left(\frac{12\,\mu a\Omega\epsilon R_2^3\pi\sin\phi}{c^2}\right)$$

$$\times\left\{\frac{1}{2}\left[1 - \left(\frac{R_1}{R_2}\right)^2\right] - \frac{1}{3}\left[1 - \left(\frac{R_1}{R_2}\right)^3\right]\right.$$

$$+\left[\frac{1 - (R_1/R_2)}{\ln(R_2/R_1)}\right]$$

$$\left.\times\left[\frac{1}{2}\left(\frac{R_1}{R_2}\right)^2\ln\left(\frac{R_2}{R_1}\right) - \frac{1}{4}\left(1 - \left(\frac{R_1}{R_2}\right)^2\right)\right]\right\};$$

$$k = 1 \quad [28]$$

The geometric factor contained in the last term is shown in Figure 2.

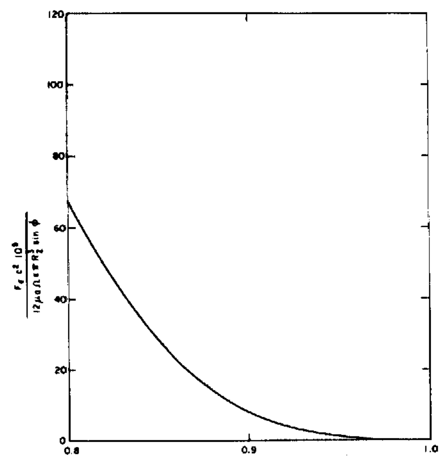

Fig. 2—Geometry factor for laminar flow (equation 28).

Examination of Eqs. [23] and [28] shows that ϵ is always multiplied by the dimension-less waviness amplitude. This means the eccentricity has no effect on seal performance unless it is accompanied by waviness. Unless very special precautions are taken one would expect that most seals would possess some degree of waviness after a period of operation, so that effects of eccentricity would manifest themselves sooner or later.

Numerical Example

The following data will be assumed for the purposes of a sample calculation:

$R_1 = 0.9$ inches $\qquad \epsilon = 3(10^{-3})$ inches

$R_2 = 1.0$ inches $\qquad \Omega = \frac{2\pi}{60}(10^3)$ rad/sec

$\mu = 10^{-7}\frac{\text{lb}_f\text{-sec}}{\text{in.}^2} \qquad \phi = 90°$

$\rho = 10^{-4}\frac{\text{lb}_f\text{-sec}^2}{\text{in.}^4} \qquad a = 0.5$

$c = 10^{-3}$ inches $\qquad P_1 = P_2$

The Reynolds number for this flow is

$$\frac{R_2^2\omega}{\nu} = 1.046(10^5)$$

The total leakage rate as given by Eq. [32] is

$$Q = \frac{\pi c^3}{6\,\mu\ln(R_2/R_1)}\left[P_1 - P_2 + \frac{3}{20}\rho\Omega^2(R_2^2 - R_1^2)\right]$$

$$\left(1 + \frac{3}{2}a^2\right) + \left[\frac{\epsilon\Omega\pi(R_2 - R_1)ca\sin\phi}{2\ln(R_2/R_1)}\right]$$

$$= 2.14(10^{-3}) + 0.24(10^{-3})$$

$$= 2.38(10^{-3})\ \text{in.}^3/\text{sec}$$

In this case about 10% of the total leakage is due to eccentricity.

The total separating force given by Eq. [28] (for $k = 1$) is

$$F = F_0 + F_\epsilon \cong \pi P_1(R_2^2 - R_1^2) + \frac{\pi}{2}\left(\frac{3\rho\Omega^2}{20}\right)(R_2^2 - R_1^2)$$

$$+\frac{2\pi}{\ln\left(\frac{R_2}{R_1}\right)}\left[P_2 - P_1 - \frac{3\rho\Omega^2}{20}(R_2^2 - R_1^2)\right]$$

$$\left[\frac{R_2^2}{2}\ln\left(\frac{R_2}{R_1}\right) - \frac{1}{4}(R_2^2 - R_1^2)\right]$$

$$+\left(\frac{12\,\mu a\Omega\epsilon R_2^3\pi\sin\phi}{c^3}\right)\left\{\frac{1}{2}\left[1 - \left(\frac{R_1}{R_2}\right)^2\right] - \frac{1}{3}\left[1 - \left(\frac{R_1}{R_2}\right)^3\right]\right.$$

$$\left. + \left[\frac{1 - \left(\frac{R_1}{R_2}\right)}{\ln\left(\frac{R_2}{R_1}\right)}\right]\left[\frac{1}{2}\left(\frac{R_1}{R_2}\right)^2\ln\left(\frac{R_2}{R_1}\right) - \left(\frac{1 - \left(\frac{R_1}{R_2}\right)^2}{4}\right)\right]\right\}$$

$$= 0 + 0.93276(10^{-2}) - 0.96889(10^{-2}) + 4.74(10^{-5})$$

$$= -3.14(10^{-4})\ \text{lb}_f$$

Here F is about 15% of F, with the net separating force negative as a result of centrifugal effects.

ANALYSIS OF AN ALIGNED WAVY SEAL WITH ECCENTRICITY-TURBULENT FLOW

Velocity Distribution

For turbulent flow the tangential velocity profile will be taken to the 1/7 power law measured from the surfaces. Using the velocity matching conditions that the mid-channel velocity is one-half the rotating surface velocity, the tangential profile is then given by

$$v_1 = r\Omega\left[1 - \frac{1}{2}\left(\frac{z}{h/2}\right)^{1/7}\right];\ 0 \le z \le h/2 \qquad [29]$$

$$v_2 = \frac{r\Omega}{2}\left(\frac{h-z}{h/2}\right)^{1/7};\ h/2 \le z \le h \qquad [30]$$

It will be recalled that these profiles are consistent with the concept that the tangential shear stress is constant across the clearance space, assuming that the shear flow follows the "law of the wall", (2). It is easily verified that Eqs. [29] and [30] also have the same slope at mid-channel with the result that the velocity profile is "smooth" all across the clearance space.

The radial velocity profile can now be computed from the integrated form of the equations of motion

$$u\tau_{z\theta_w} = v\left[f(r,\theta) + \frac{\partial p}{\partial r}z - \rho_r \int v^2\,dz\right] + \int \rho_r v^3\,dz - \int v\frac{\partial p}{\partial r}\,dz + g(r,\theta) \qquad [31]$$

which was derived in (2) with the aid of the concept of isotropic viscosity.

This radial velocity profile must satisfy the boundary conditions given in Eq. [9]. Evaluating the arbitrary functions $f(r,\theta)$ and $g(r,\theta)$

$$u\tau_{z\theta_w} = \left[\frac{v\epsilon\tau_{z\theta_w}}{r}\right]\sin(\phi-\theta) + \frac{\rho}{r}\left[\frac{v}{r\Omega}\int_0^h v^3\,dz - v\int_0^z v^2\,dz + \int_0^z v^3\,dz - \int_0^h v^3\,dz\right] + \left[vz - \frac{v}{r\Omega}\int_0^h v\,dz - \int_0^z v\,dz + \int_0^h v\,dz\right]\frac{\partial p}{\partial r} \qquad [32]$$

When the velocities given by Eqs. [29] and [30] are inserted into Eq. [35] the radial velocity distribution takes the form

$$u\tau_{z\theta_w} = \left(\frac{r\Omega}{2}\right)\frac{h}{2}\frac{\partial p}{\partial r}F\left(\frac{z}{h/2}\right) - \frac{\rho}{r}\left(\frac{r\Omega}{2}\right)^3\frac{h}{2}G\left(\frac{z}{h/2}\right) + \frac{v\epsilon\tau_{z\theta_w}}{r}\sin(\phi-\theta) \qquad [33]$$

where

$$F\left(\frac{z}{h/2}\right) = \left(\frac{z}{h/2}\right)^{1/7}\left[1 - \frac{1}{8}\left(\frac{z}{h/2}\right)\right];\ 0 \le z \le \frac{h}{2}$$
$$= \left(\frac{h-z}{h/2}\right)^{1/7}\left[\frac{3}{4} + \frac{1}{8}\left(\frac{z}{h/2}\right)\right];\ h/2 \le z \le h \qquad [34]$$

and

$$G\left(\frac{z}{h/2}\right) = \left(\frac{z}{h/2}\right)^{1/7}\left[\frac{13}{12} - \frac{1}{2}\left(\frac{z}{h/2}\right)^{8/7} + \frac{7}{18}\left(\frac{z}{h/2}\right)^{9/7} - \frac{7}{90}\left(\frac{z}{h/2}\right)^{10/7}\right];\ 0 \le z \le \frac{h}{2}$$
$$= \left(\frac{z}{h/2}\right)^{1/7}\left[\frac{35}{36} - \frac{7}{90}\left(\frac{h-z}{h/2}\right)^{9/7}\right];\ \frac{h}{2} \le z \le h \qquad [35]$$

With the exception of a sign change due to a change in direction rotation, Eq. [33] is identical to one reported earlier for turbulent flow (2). It should also be noted that z is measured from the rotating surface in this analysis, whereas it was previously measured from the stationary surface.

Pressure Distribution

The radial pressure gradient is determined from the continuity Eq. [12] with the aid of Eqs. [29], [30], [33], [34] and [35]. The result is

$$\frac{\partial p}{\partial r} = \frac{835}{816}\rho\left(\frac{\Omega}{2}\right)^2 r + \frac{f(\theta)}{r^{1/4}} + 0.0225\frac{\rho}{2}(\nu)^{1/4}\left(\frac{30}{49}\right)\left[\frac{\left|\frac{r\Omega}{2}\right|^{3/4}\left(\frac{r\Omega}{2}\right)}{\left(\frac{h}{2}\right)^{9/4}}\right] + 0.0225\,\rho(\nu)^{1/4}\epsilon\Omega\left(\frac{30}{49}\right)\left[\frac{\left|\frac{r\Omega}{2}\right|^{3/4}}{\left(\frac{h}{2}\right)^{5/4}}\right]\sin(\phi-\theta) \qquad [36]$$

Integrating this equation with respect to r yields the general solution

$$p = \frac{835}{816}\left(\frac{\rho}{2}\right)\left(\frac{r\Omega}{2}\right)^2 + 0.0225\left(\frac{30}{49}\right)\left(\frac{4}{11}\right)\frac{\rho}{2}(\nu)^{1/4}r^{11/4}\left[\frac{\left|\frac{\Omega}{2}\right|^{3/4}\left(\frac{\Omega}{2}\right)}{\left(\frac{h}{2}\right)^{9/4}}\right]\frac{\partial h}{\partial\theta} + 0.0225\,\rho(\nu)^{1/4}\epsilon\Omega\sin(\phi-\theta)\left(\frac{4}{7}\right)\left(\frac{30}{49}\right)\left[\frac{\left|\frac{\Omega}{2}\right|^{3/4}}{\left(\frac{h}{2}\right)^{5/4}}\right]r^{7/4} + \frac{4}{3}r^{3/4}f(\theta) + g(\theta) \qquad [37]$$

The arbitrary functions $f(r,\theta)$ and $g(r,\theta)$ may now be evaluated using the pressure boundary conditions

$$p = P_1 \qquad r = R_1$$
$$p = P_2 \qquad r = R_2$$

The pressure distribution which results is given by the formula

$$p - P_2 = \frac{835}{816}\left(\frac{\rho}{2}\right)\left(\frac{\Omega}{2}\right)^2(r^2 - R_2^2) + 0.0225\left(\frac{30}{49}\right)\left(\frac{4}{11}\right)\frac{\rho}{2}(\nu)^{1/4}\left[\frac{\left|\frac{\Omega}{2}\right|^{3/4}\left(\frac{\Omega}{2}\right)}{\left(\frac{h}{2}\right)^{9/4}}\right](r^{11/4} - R_2^{11/4})\frac{\partial h}{\partial\theta} + \left[\frac{r^{3/4} - R_2^{3/4}}{R_1^{3/4} - R_2^{3/4}}\right]\Bigg\{P_1 - P_2 + \frac{835}{816}\frac{\rho}{2}\left(\frac{\Omega}{2}\right)^2(R_2^2 - R_1^2) - 0.0225\left(\frac{30}{49}\right)\left(\frac{4}{11}\right)\frac{\rho}{2}(\nu)^{1/4}\left[\frac{\left|\frac{\Omega}{2}\right|^{3/4}\left(\frac{\Omega}{2}\right)}{\left(\frac{h}{2}\right)^{9/4}}\right](R_1^{11/4} - R_2^{11/4})\frac{\partial h}{\partial\theta}\Bigg\}$$
$$+ 0.0225\,\rho(\nu)^{1/4}\left(\frac{4}{7}\right)\left(\frac{30}{49}\right)\epsilon\Omega\sin(\phi-\theta)\left[\frac{\left|\frac{\Omega}{2}\right|^{3/4}}{\left(\frac{h}{2}\right)^{5/4}}\right](r^{7/4} - R_2^{7/4}) - \left[\frac{r^{3/4} - R_2^{3/4}}{R_1^{3/4} - R_2^{3/4}}\right]\left[0.0225\,\rho(\nu)^{1/4}\left(\frac{4}{7}\right)\left(\frac{30}{49}\right)\epsilon\Omega\sin(\phi-\theta)\left[\frac{\left|\frac{\Omega}{2}\right|^{3/4}}{\left(\frac{h}{2}\right)}\right](R_1^{7/4} - R_2^{7/4})\right] \qquad [39]$$

When $\epsilon = 0$ this equation reduces to the earlier result given in (2) with the sign of the angular velocity reversed.

Leakage Flow

The net leakage-rate is calculated from Eq. [19] using for the velocity profile the formulations given by Eq. [33] and [39]. The resulting leakage rate is given approximately by

$$Q \cong \frac{P_1 - P_2 + \frac{835}{816}\left(\frac{\rho}{2}\right)\left(\frac{R_2\Omega}{2}\right)^2\left[1 - \left(\frac{R_1}{R_2}\right)^2\right]\left[1 + \frac{45}{64}a^2\right]}{\frac{0.0195}{\pi}\left|\frac{\rho^3\mu\Omega^3}{c^9}\right|^{1/4}\left(\frac{4}{3}\right)(R_2^{3/4} - R_1^{3/4})} + \frac{3}{7}\left(\frac{\epsilon\Omega}{2}\right)\frac{R_1^{7/4} - R_2^{7/4}}{R_1^{3/4} - R_2^{3/4}}\int_0^{2\pi} h\sin(\phi-\theta)\,d\theta \qquad [40]$$

neglecting terms of the order a^4.

The last term in this equation is the contribution due to eccentricity. As in the laminar case the integral in this term is non-zero only when $k = 1$. In this case

$$\int_0^{2\pi} h\sin(\phi-\theta)\,d\theta = \pi ca\sin\phi;\ k = 1 \qquad [41]$$

so that the total leakage rate (for $k = 1$) is

$$Q \cong \frac{P_1 - P_2 + \frac{835}{816}\left(\frac{\rho}{2}\right)\frac{R_2\Omega^2}{2}\left[1 - \left(\frac{R_1}{R_2}\right)^2\right]\left[1 + \frac{45}{64}a^2\right]}{\frac{0.0195}{\pi}\left|\frac{\rho^3\mu\Omega^3}{c^9}\right|^{1/4}\left(\frac{4}{3}\right)(R_2^{3/4} - R_1^{3/4})} + \frac{3}{7}\left(\frac{\Omega}{2}\right)\epsilon R_2 ca\pi\sin\phi\left[\frac{1 - \left(\frac{R_1}{R_2}\right)^{7/4}}{1 - \left(\frac{R_1}{R_2}\right)^{3/4}}\right] \qquad [42]$$

Once again it should be noted that eccentricity alone is not sufficient to cause additional leakage, it must be coupled with a specific kind of surface waviness ($k = 1$). Even then the additional leakage can be outward or inward depending on the phase angle ϕ or the direction of rotation.

Torque

The total torque exerted by the fluid on the rotating face of the seal is given by

$$T = \int_0^{2\pi}\int_{R_1}^{R_2} \tau_{z\theta_w} r^2 r\,dr\,d\theta \qquad [43]$$

The wall shear stress used to obtain Eq. [39] is the same as used in (1), namely

$$\tau_{z\theta_w} = 0.0225\rho(\nu)^{1/4}\left(\frac{r\Omega}{2}\right)\left[\frac{\left|\frac{r\Omega}{2}\right|^3}{\left(\frac{h}{2}\right)}\right]^{1/4} \qquad [44]$$

If the film thickness relation given by Eq. [21] is expanded binomially before integration with respect to θ, it is clear that the torque is the same as for the concentric case, i.e.,

$$T = 0.00335\pi\left[\frac{\rho^3\mu\Omega^7 R_2^{19}}{c}\right]^{1/4}\left[1 - \left(\frac{R_1}{R_2}\right)^{19/4}\right]\left[1 + \frac{5}{64}a^2\right] \qquad [45]$$

neglecting terms of the order a^4.

Separating Force

The last two terms in the expression for the pressure distribution present the contribution of the eccentricity. The separating force is determined by the integral

$$F = \int_0^{2\pi}\int_{R_1}^{R_2} pr\,dr\,d\theta$$

When this integration is carried out the separating force for the concentric case will be retrieved with the addition of a term proportional to ϵ. Since both eccentricity terms of Eq. [42] contain $\frac{\sin(\phi-\theta)}{(h/2)^{5/4}}$ the integral of this factor will be determined first. Using the film thickness relation given by Eq. [21]

$$
\begin{aligned}
\int_0^{2\pi} \frac{\sin(\phi-\theta)\,d\theta}{(h/2)^{5/4}}
&= \frac{1}{(c/2)^{5/4}} \int_0^{2\pi} \left[\frac{\sin\phi\cos\theta - \cos\phi\sin\theta}{(1+a\cos k\theta)^{5/4}}\right] d\theta \\
&= \frac{1}{(c/2)^{5/4}} \int_0^{2\pi} (\sin\phi\cos\theta - \sin\theta\cos\phi) \\
&\quad \times \left(1 - \frac{5}{4}a\cos k\theta + \cdots\right) d\theta \\
&= -\frac{5}{4}\left[\frac{a\pi}{(c/2)^{5/4}}\right]\sin\phi;\ k = 1. \qquad [46]
\end{aligned}
$$

It should again be noted that only if $k = 1.0$ is there any contribution of eccentricity to the order of a^2.

The total separating force for $k = 1$ and small a computed from Eqs. [25] and [39] is approximately

$$
\begin{aligned}
F = F_0 + F_\epsilon &= \pi P_1(R_2^2 - R_1^2) + \frac{\pi}{2}\left(\frac{835}{816}\right)\frac{\rho}{2}\left(\frac{\Omega}{2}\right)^2 (R_2^2 - R_1^2) \\
&+ 2\pi\left[\frac{P_2 - P_1 - \left(\frac{835}{816}\right)\frac{\rho}{2}\left(\frac{\Omega}{2}\right)^2 (R_2^2 - R_1^2)}{(R_2^{3/4} - R_1^{3/4})}\right] \\
&\left[\frac{4}{11}(R_2^{11/4} - R_1^{11/4}) - \frac{R_1^{3/4}}{2}(R_2^2 - R_1^2)\right] \\
&+ 0.0225\left(\frac{10}{7}\right)\left(\frac{30}{49}\right)\rho(\nu)^{1/4} \\
&a\epsilon\pi R_2^2 \left|\frac{R_2\Omega}{2}\right|^{3/4}\left(\frac{R_2\Omega}{2}\right)\left[\frac{\sin\phi}{\left(\frac{c}{2}\right)^{5/4}}\right] \\
&\left\{\frac{4}{11}\left[\frac{1-\left(\frac{R_1}{R_2}\right)^{7/4}}{1-\left(\frac{R_1}{R_2}\right)^{3/4}}\right]\right. \\
&\left[1-\left(\frac{R_1}{R_2}\right)^{11/4}\right] - \frac{4}{15}\left[1-\left(\frac{R_1}{R_2}\right)^{15/4}\right] \\
&\left. + \left[1-\left(\frac{1-\left(\frac{R_1}{R_2}\right)^{7/4}}{1-\left(\frac{R_1}{R_2}\right)^{3/4}}\right)\right]\left[\frac{1-\left(\frac{R_1}{R_2}\right)^2}{2}\right]\right\};\ k = 1 \qquad [47]
\end{aligned}
$$

The geometric factor contained in the last term is shown in Fig. 3.

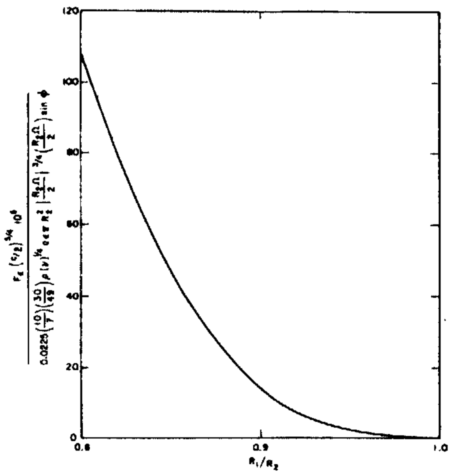

Fig. 3—Geometry factor for turbulent flow (equation 47).

Numerical Example

If the rotational speed is increased to 10^4 rpm the Reynolds number will now be

$$\frac{R_2^2\omega}{\nu} = 1.046(10^6)$$

Assuming that the flow is turbulent at this speed the net leakage rate is given by Eq. [42] as

$$
\begin{aligned}
Q &= \frac{P_1 - P_2 + \frac{835}{816}\left(\frac{\rho}{2}\right)\left(\frac{R_2\Omega}{2}\right)^2\left[1-\left(\frac{R_1}{R_2}\right)^2\right]\left[1+\frac{45}{64}a^2\right]}{\frac{0.0195}{\pi}\left|\frac{\rho^3\mu\Omega^3}{c^9}\right|^{1/4}\left(\frac{4}{3}\right)(R_2^{3/4} - R_1^{3/4})} \\
&+ \frac{3}{7}\left(\frac{\Omega}{2}\right)\epsilon R_2 ca\pi\sin\phi\left[\frac{1-\left(\frac{R_1}{R_2}\right)^{7/4}}{1-\left(\frac{R_1}{R_2}\right)^{3/4}}\right]
\end{aligned}
$$

$$= 0.270 + 0.00234 = 0.272\ \text{in.}^3/\text{sec}$$

Here only about 1% of the total leakage is attributed to eccentricity.

The total separating force is predicted by Eq. [47] (for $k = 1$) as

$$
\begin{aligned}
F = F_0 + F_\epsilon &= \pi P_1(R_2^2 - R_1^2) + \frac{\pi}{2}\left(\frac{835}{816}\right)\frac{\rho}{2}\left(\frac{\Omega}{2}\right)^2 (R_2^2 - R_1^2) \\
&+ 2\pi\left[\frac{P_1 - P_2 - \frac{835}{816}\left(\frac{\rho}{2}\right)\left(\frac{\Omega}{2}\right)(R_2^2 - R_1^2)}{R_2^{3/4} - R_1^{3/4}}\right] \\
&\left[\frac{4}{11}(R_2^{11/4} - R_1^{11/4}) - \frac{R_1^{3/4}}{2}(R_2^2 - R_1^2)\right] \\
&+ 0.0225\left(\frac{10}{7}\right)\left(\frac{30}{49}\right)\rho\nu^{1/4}a\epsilon R_2^2\pi \\
&\left|\frac{R_2\Omega}{2}\right|^{3/4}\left(\frac{R_2\Omega}{2}\right)\left[\frac{\sin\phi}{\left(\frac{c}{2}\right)^{5/4}}\right] \\
&\left\{\frac{4}{11}\left[\frac{1-\left(\frac{R_1}{R_2}\right)^{7/4}}{1-\left(\frac{R_1}{R_2}\right)^{3/4}}\right]\left[1-\left(\frac{R_1}{R_2}\right)^{11/4}\right]\right. - \frac{4}{15}\left[1-\left(\frac{R_1}{R_2}\right)^{15/4}\right] \\
&\left. + \left[1-\left(\frac{1-\left(\frac{R_1}{R_2}\right)^{7/4}}{1-\left(\frac{R_1}{R_2}\right)^{3/4}}\right)\right]\left[\frac{1-\left(\frac{R_1}{R_2}\right)^2}{2}\right]\right\} \\
&= 0 + 0.795 - 0.850 + 1.76(10^{-4}) = -5.5(10^{-2})\ \text{lb}_f
\end{aligned}
$$

The total force F in this case is not significantly influenced by F_ϵ.

CONCLUSIONS

From the analyses of laminar and turbulent flow it is clear that eccentricity coupled with waviness contributes to the leakage rate and to a lesser extent the separating force, while the torque is uneffected. Since both eccentricity and waviness are expected in seals operating under realistic conditions (3), these contributions are of practical significance in seal performance.

For small waviness amplitude relative to the normal clearance the analysis shows that these contributions are proportional to the product of the eccentricity and waviness amplitude. The analysis also shows that a "once per revolution" waviness is the main contributor to the pumping effect.

An unusual feature of the pumping is the effect of rotational direction and phase angle ϕ. Depending on the direction of rotation and the magnitude of ϕ, the pumping due to eccentricity can be a positive or negative contribution to the leakage caused by the applied pressure difference, centrifugal force and the waviness alone. This discovery suggests that otherwise identical seals assembled differently will have different net leakage rates and separating forces. If the maximum expected eccentricity can be estimated, the equations given above predict the maximum expected variation in leakage and separating force which would result from random assembly.

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The Misaligned, Eccentric Face Seal

III. L

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The analytical investigation of a misaligned face seal shows that eccentricity will cause a component of the total leakage to be inward or outward depending on the phase angle between the eccentricity and the misalignment.

INTRODUCTION

The impossibility of constructing a face seal which is neither flat, parallel, nor concentric raises a question as to the effects of these deviations from the ideal geometry. The first two of these geometric perturbations have been investigated elsewhere (*1*), (*2*). It has recently been shown by Findlay (*3*) that eccentricity combined with waviness or misalignment can cause what have heretofore been considered anomolous pumping effects. The analysis to follow is directed toward establishing an analytical description of the combined effect of eccentricity and misalignment.

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LAMINAR FLOW

Momentum Equations

It was shown in (*1*) that the momentum equations for a seal-like configurations are

$$\frac{\partial p}{\partial r} = \rho \frac{v^2}{r} + \frac{\partial \tau_{zr}}{\partial z} \quad [1]$$

NOMENCLATURE

$\bar{e}_r, \bar{e}_\theta, \bar{e}_z$ = unit vectors
F = total separating force
F_ϵ = separating force due to eccentricity
h = local film thickness
h_0 = disk clearance along centerline
$I_1(r,\theta) = \int \frac{r^{11/4}}{(h/2)^{9/4}} dr$
$I_2(r,\theta) = \int \frac{1}{r^{1/4}(h/2)^{9/4}} dr$
$I_3(r,\theta) = \int \frac{r^{3/4}}{(h/2)^{5/4}} dr$
p = pressure in clearance space
P_1 = pressure at the inner radius of the seal
P_2 = pressure at the outer radius of the seal
Q = total volumetric leakage rate
Q_ϵ = volumetric leakage rate due to eccentricity
r = radial coordinate
R_1 = inside radius of seal
R_2 = outside radius of seal
T = total torque
u = temporal mean radial velocity component
v = temporal mean tangential velocity component
$\bar{V}_\omega$ = tilted surface velocity vector
$\bar{V}_\Omega$ = eccentric surface velocity vector
w = temporal mean axial velocity component
z = axial coordinate
ϵ = seal eccentricity
θ = tangential coordinate
λ = angle of axis misalignment
μ = fluid viscosity
$\nu = \mu/\rho$
ρ = fluid density
τ = shear stress
ϕ = phase angle between eccentricity line of centers and tilt reference line
ω = angular velocity of tilted surface
Ω = angular velocity of eccentric surface

$$\frac{\partial \tau_{z\theta}}{\partial z} = 0 \quad [2]$$

$$\frac{\partial p}{\partial z} = 0 \quad [3]$$

When the flow is laminar, these equations take the form

$$\frac{\partial p}{\partial r} = \rho \frac{v^2}{r} + \mu \frac{\partial^2 u}{\partial z^2} \quad [4]$$

$$\frac{\partial p}{\partial z} = \frac{\partial^2 v}{\partial z^2} = 0 \quad [5]$$

Velocity Distribution

The seal geometry is shown in Fig. 1. For small angles of tilt λ the vectorial velocity of a point on the upper tilted disk is given by

$$\bar{V}_\omega \cong [r\omega \cos\theta \sin\theta \tan\lambda \sin\lambda]\bar{e}_r + [r\omega \cos\lambda]\bar{e}_\theta + [r\omega \sin\lambda \sin\theta]\bar{e}_z \quad [6]$$

while the vectorial velocity of a point on the lower disk is given by

$$\bar{V}_\Omega = [\epsilon\Omega \sin(\phi - \theta)]\bar{e}_r + [r\Omega]\bar{e}_\theta \quad [7]$$

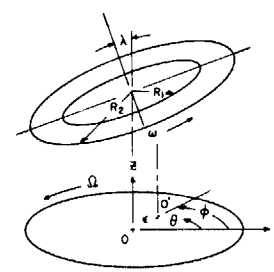

Fig. 1—Seal geometry.

Equation [5] indicates that the tangential velocity profile is linear in the axial coordinate. The linear profile satisfying the boundary conditions of Eqs. [6] and [7] is

$$v = r(\omega \cos\lambda - \Omega)\frac{z}{h} + r\Omega \quad [8]$$

where h is a function of r and θ.

When this velocity is introduced into Eq. [4] and that equation is integrated twice with respect to z, the predicted form of the radial velocity profile is

$$u = \frac{z^2}{2\mu}\frac{\partial p}{\partial r} + F(r,\theta)z + G(r,\theta)$$

$$- \frac{\rho r}{\mu}\left[(\omega\cos\lambda - \Omega)\left(\frac{\Omega}{h}\right)\left(\frac{z^3}{3}\right) + \frac{(\Omega z)^2}{2} + (\omega\cos\lambda - \Omega)^2\left(\frac{z^4}{12h}\right)\right] \quad [9]$$

The boundary conditions on the radial velocity profile are

$$\begin{aligned} u &= \epsilon\Omega \sin(\phi - \theta),\ z = 0 \\ u &= r\omega \sin\theta \cos\theta \tan\lambda \sin\lambda,\ z = h \end{aligned} \quad [10]$$

Although the second of these two conditions obviously yields a very small radial component on the tilted surface, it will be carried through the analysis. Eventually it will be necessary to use expansions involving $\tan\lambda$ at which point the importance of this boundary condition will be discussed.

The radial velocity profile which satisfies the conditions imposed by Eq. [10] is

$$\begin{aligned} u = &\left(\frac{z^2 - hz}{2\mu}\right)\frac{\partial p}{\partial r} + \epsilon\Omega \sin(\phi - \theta)\left[1 - \frac{z}{h}\right] \\ &+ \frac{z}{h}(r\omega \sin\theta \cos\theta \cos\lambda \tan^2\lambda) \\ &- \frac{\rho r}{\mu}\left[(\omega\cos\lambda - \Omega)\left(\frac{\Omega}{h}\right)\left(\frac{z^3 - zh^2}{3}\right) \right. \\ &\left. + \Omega^2\left(\frac{z^2 - zh}{2}\right) + \left(\frac{\omega\cos\lambda - \Omega}{h}\right)^2 \left(\frac{z^4 - zh^3}{12}\right)\right] \end{aligned} \quad [11]$$

Pressure Distribution

The integrated form of continuity equation is

$$\int_0^h \frac{1}{r}\frac{\partial}{\partial r}(ur)\,dz + \int_0^h \frac{1}{r}\frac{dv}{\partial\theta}\,dz + \int_0^h \frac{\partial w}{\partial z}\,dz = 0 \quad [12]$$

The first term of this equation can be written as

$$\frac{1}{r}\int_0^h \frac{\partial}{\partial r}(ur)\,dz = \frac{1}{r}\frac{\partial}{\partial r}\int_0^h (ur)\,dz - \left(u\frac{\partial h}{\partial r}\right)_{z=h} \quad [13]$$

For the geometry shown in Fig. 1 the film thickness is given by

$$h = h_0 - r\cos\theta\tan\lambda \quad [14]$$

so that

$$\frac{1}{r}\int_0^h \frac{\partial}{\partial r}(ur)\,dz = \frac{1}{r}\frac{\partial}{\partial r}\int_0^h (ur)\,dz + r\omega \sin\theta \cos^2\theta \cos\lambda \tan^3\lambda \quad [15]$$

The second term of the continuity equation can be integrated using Eq. [8] to yield

$$\frac{1}{r}\int_0^h \frac{\partial v}{\partial \theta}\, dz = \frac{1}{2}(\omega \cos\lambda - \Omega) r \sin\theta \tan\lambda \quad [16]$$

The third term can also be integrated in a straightforward manner, and making use of Eq. [6] becomes

$$\int_0^h \frac{\partial \omega}{\partial z}\, dz = r\omega \sin\lambda \sin\theta \quad [17]$$

Using Eqs. [15], [16] and [17], the continuity equation becomes

$$\frac{1}{r}\frac{\partial}{\partial r}\int_0^h (ur)\, dz + r\omega \sin\theta \cos^2\theta \cos\lambda \tan^3\lambda + \frac{1}{2} r\omega \sin\theta \sin\lambda + \frac{1}{2} r\Omega \sin\theta \tan\lambda = 0 \quad [18]$$

Integrating this equation with respect to r gives

$$\int_0^h (ur)\, dz = -\frac{r^2}{3}\left[\omega \sin\theta \cos^2\theta \cos\lambda \tan^3\lambda + \frac{1}{2}(\omega\cos\lambda + \Omega)\tan\lambda \sin\theta\right] - \frac{\Phi(\theta)}{r} \quad [19]$$

The integral on the left hand side of this equation can be evaluated directly from the radial velocity Eq. [9] so that

$$\int_0^h u\, dz = -\frac{h^3}{12\mu}\frac{\partial p}{\partial r} + \frac{1}{2}h\Omega\epsilon \sin(\phi - \theta) + \frac{1}{2}(r\omega h \cos\lambda \tan^2\lambda \sin\theta\cos\theta) + \frac{\rho r h^3}{12\mu}\left[\Omega(\omega\cos\lambda - \Omega) + \Omega^2 + \frac{3}{10}(\omega\cos\lambda - \Omega)^2\right] \quad [20]$$

Equating Eqs. [19] and [20] and solving for the pressure gradient results in

$$\frac{\partial p}{\partial r} = \frac{6\mu}{h^2}\left[\epsilon\Omega \sin(\phi-\theta)\right] + \frac{6\mu}{h^2}(r\omega\cos\lambda\tan^2\lambda\sin\theta\cos\theta) + \rho r\left[\Omega(\omega\cos\lambda - \Omega) + \Omega^2 + \frac{3}{10}(\omega\cos\lambda-\Omega)^2\right] + \left(\frac{2\mu r^2}{h^3}\right)\sin\theta\tan\lambda\,[\Omega + \omega\cos\lambda\,(1 + 2\cos^2\theta\tan^2\lambda)] + \left(\frac{12\mu}{rh^3}\right)\Phi(\theta) \quad [21]$$

In typical applications $\tan\lambda$ is of the order 10^{-3} which means that

$$1 + 2\cos^2\theta\tan^2\lambda \cong 1 \quad [22]$$

in Eq. [21].

Using this simplification and integrating Eq. [21] in the radial direction yields

$$p = 6\mu[\epsilon\Omega\sin(\phi-\theta)]\int_{R_1}^r \frac{dr}{h^2} + 6\mu[\omega\cos\lambda\tan^2\lambda\sin\theta\cos\theta]\int_{R_1}^r \frac{r\,dr}{h^2} + \frac{\rho r^2}{2}\left[\omega\Omega\cos\lambda + \frac{3}{10}(\omega\cos\lambda-\Omega)^2\right] + 12\mu\Phi(\theta)\int_{R_1}^r\frac{dr}{rh^3} + 2\mu(\omega\cos\lambda+\Omega)\sin\theta\tan\lambda\int_{R_2}^r \frac{r^2\,dr}{h^3} + \Psi(\theta) \quad [23]$$

The arbitrary functions $\Phi(\theta)$ and $\Psi(\theta)$ are determined from the boundary conditions

$$p = P_1,\ r = R_1 \qquad p = P_2,\ r = R_2 \quad [24]$$

with the result that

$$\Psi(\theta) = P_2 \quad [25]$$

and

$$\Phi(\theta) = \frac{1}{12\mu\int_{R_1}^{R_2}\frac{dr}{rh^3}}\Bigg\{P_1 - P_2 - 6\mu[\epsilon\Omega\sin(\phi-\theta)]\int_{R_2}^{R_1}\frac{dr}{h^2} - 6\mu[\omega\cos\lambda\tan^2\lambda\sin\theta\cos\theta]\int_{R_2}^{R_1}\frac{r\,dr}{h^2} + \frac{\rho}{2}(R_2^2 - R_1^2)\left[\Omega\omega\cos\lambda + \frac{3}{10}(\omega\cos\lambda-\Omega)^2\right] - 2\mu(\omega\cos\lambda+\Omega)\sin\theta\tan\lambda\int_{R_2}^{R_1}\frac{r^2\,dr}{h^3}\Bigg\} \quad [26]$$

Leakage-Rate

The volumetric leakage rate is calculated from the formula

$$Q = \int_0^{2\pi}\int_0^h (ur)\, dz\, d\theta \quad [27]$$

The integral portion of this equation can be determined by combining Eqs. [20] and [21], eliminating $\frac{\partial p}{\partial r}$ so that

$$\int_0^h u\, dz = -\frac{r^2}{6}(\omega\cos\lambda + \Omega)\sin\theta\tan\lambda - \frac{\Phi(\theta)}{r} \quad [28]$$

and Eq. [27] reduces to

$$Q = -\int_0^{2\pi}\Phi(\theta)\, d\theta \quad [29]$$

An inspection of Eq. [25] shows that the only terms which do not vanish upon integration with respect to θ are

$$\frac{1}{12\mu\int_{R_2}^{R_1}\frac{dr}{rh^3}}\left\{P_1 - P_2 + \frac{\rho}{2}\left[\Omega\omega\cos\lambda + \frac{3}{10}(\omega\cos\lambda - \Omega)^2\right](R_2^2 - R_1^2)\right\} \quad [30]$$

and

$$-\left\{\frac{\int_{R_2}^{R_1}\frac{dr}{h^2}}{12\mu\int_{R_2}^{R_1}\frac{dr}{rh^3}}\right\} 6\mu[\epsilon\Omega\sin(\phi-\theta)] \quad [31]$$

The first of these terms is associated with misalignment alone and independent of eccentricity. The contribution of this term was investigated in (*1*).

The contribution of the second term obviously represents the effect of eccentricity.

Since h is a known function of r and θ the integrands may be expanded in power series of the quantity $\frac{r}{h_0}\tan\lambda\cos\theta$ (since it is less than unity) and integrated term by term to form the integrand of Eq. [29]. This infinite series can then be integrated term by term with respect to the variable θ. The result of this rather lengthy manipulation is

$$Q_\epsilon = -\frac{1}{12\mu}\int_0^{2\pi} 6\mu\epsilon\Omega\left[\frac{\int_{R_2}^{R_1}\frac{dr}{h^2}}{\int_{R_2}^{R_1}\frac{dr}{rh^3}}\right]\sin(\phi-\theta)\, d\theta = \pi\epsilon\left(\frac{\Omega}{2}\right)R_2h_0\sin\phi\left\{\left[\frac{3\left(1-\frac{R_1}{R_2}\right)^2}{\ln\left(\frac{R_2}{R_1}\right)} - \frac{1-\left(\frac{R_1}{R_2}\right)^2}{\ln\left(\frac{R_2}{R_1}\right)}\right]\left(\frac{R_2\tan\lambda}{h_0}\right) + 0\left(\frac{R_2\tan\lambda}{h_0}\right)^3 + \cdots\right\} \quad [32]$$

The expansion described above was actually carried to obtain the coefficient of $(R_2\tan\lambda/h_0)^3$. The results of computations including cubic terms are shown in Fig. 2 where it is evident that retaining only the linear term in $R_2\tan\lambda/h_0$ is a very good approximation in the range $0.8 < R_1/R_2 < 1.0$.

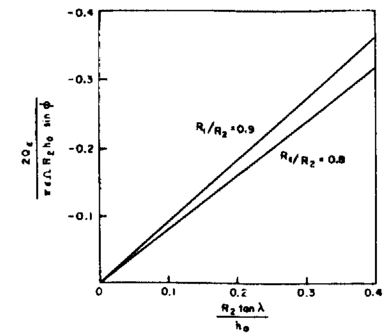

Fig. 2—Laminar flow coefficient.

It was mentioned earlier that expansions of the type just described would be necessary and so terms of the order $\tan^2\lambda$ would be retained. In this case the term involving $\tan^2\lambda$ was eliminated even before the expansion was performed.

Separating Force

The separating force is given by

$$F = \int_0^{2\pi}\int_{R_1}^{R_2} pr\, dr\, d\theta \quad [33]$$

The pressure distribution itself is

$$p - P_2 = 6\mu[\epsilon\Omega\sin(\phi-\theta)]\int_{R_2}^r\frac{dr}{h^2} + 6\mu\omega\cos\lambda\tan^2\lambda\sin\theta\cos\theta\int_{R_2}^r\frac{r\,dr}{h^2} + \frac{\rho}{2}\left[\Omega\omega\cos\lambda + \frac{3}{10}(\omega\cos\lambda-\Omega)^2\right](r^2 - R_2^2) + 2\mu(\omega\cos\lambda+\Omega)\sin\theta\tan\lambda\int_{R_2}^r\frac{r^2\,dr}{h^3} + \frac{\int_{R_2}^r\frac{dr}{rh^3}}{\int_{R_2}^{R_1}\frac{dr}{rh^3}}\Bigg\{P_1 - P_2 - 6\mu[\epsilon\Omega\sin(\phi-\theta)]\int_{R_2}^{R_1}\frac{dr}{h^2} - 6\mu[\omega\cos\lambda\tan^2\lambda\sin\theta\cos\theta]\int_{R_2}^{R_1}\frac{r\,dr}{h^2} + \frac{\rho}{2}(R_2^2 - R_1^2)\left[\Omega\omega\cos\lambda + \frac{3}{10}(\omega\cos\lambda-\Omega)^2\right] - 2\mu(\omega\cos\lambda+\Omega)\sin\theta\tan\lambda\int_{R_2}^{R_1}\frac{r^2\,dr}{h^3}\Bigg\} \quad [34]$$

Once again those terms which survive integration with respect to θ fall into two categories; those dependent on eccentricity and those which are not.

Those in the former category are

$$\frac{\rho}{2}\left[\Omega\omega\cos\lambda + \frac{3}{10}(\omega\cos\lambda - \Omega)^2\right](r^2 - R_2{}^2)$$

$$+ \frac{\int_{R_2}^{r}\frac{dr}{rh^3}}{\int_{R_2}^{R_1}\frac{dr}{rh^3}}\left\{P_1 - P_2 + \frac{\rho}{2}(R_2{}^2 - R_1{}^2)\left[\Omega\omega\cos\lambda\right.\right.$$

$$\left.\left. + \frac{3}{10}(\omega\cos\lambda - \Omega)^2\right]\right\} \qquad [35]$$

Their effect on performance has been investigated in (*1*).

In the later category are the terms

$$6\,\mu\epsilon\Omega\sin(\phi - \theta)\left\{\int_{R_2}^{r}\frac{dr}{h^2} - \int_{R_2}^{R_1}\frac{dr}{h^2}\left[\frac{\int_{R_2}^{r}\frac{dr}{rh^3}}{\int_{R_2}^{R_1}\frac{dr}{rh^3}}\right]\right\} \qquad [36]$$

Following the same expansion procedure as used in the leakage calculations yields

$$F_\epsilon = 6\,\mu\epsilon\Omega\int_{R_1}^{R_2}\int_0^{2\pi}\sin(\phi - \theta)\left\{\int_{R_2}^{r}\frac{dr}{h^2} - \int_{R_2}^{R_1}\frac{dr}{h^2}\left[\frac{\int_{R_2}^{r}\frac{dr}{rh^3}}{\int_{R_2}^{R_1}\frac{dr}{rh^3}}\right]\right\}d\theta\, r\, dr$$

$$= \frac{6\,\mu\epsilon\Omega R_2{}^3\sin\phi}{h_0{}^2}\left\{\left\{\frac{1}{4}\left[1 - \left(\frac{R_1}{R_2}\right)^4\right]\right.\right.$$

$$- \frac{1}{2}\left[1 - \left(\frac{R_1}{R_2}\right)^2\right]$$

$$+ \left[3\left(\frac{1 - \frac{R_1}{R_2}}{\ln\left(\frac{R_2}{R_1}\right)}\right)^2 - \frac{1 - \left(\frac{R_1}{R_2}\right)^2}{\ln\left(\frac{R_2}{R_1}\right)}\right]$$

$$\left[\frac{1}{2}\left(\frac{R_1}{R_2}\right)^2\ln\left(\frac{R_2}{R_1}\right)\right.$$

$$- \frac{1}{4}\left(1 - \left(\frac{R_1}{R_2}\right)^2\right] + 3\left(\frac{1 - \frac{R_1}{R_2}}{\ln\left(\frac{R_2}{R_1}\right)}\right)\left[\frac{1}{2}\left(1 - \left(\frac{R_1}{R_2}\right)^2\right)\right.$$

$$\left.\left. - \frac{1}{3}\left(1 - \left(\frac{R_1}{R_2}\right)^3\right)\right]\right]\left(\frac{R_2\tan\lambda}{h_0}\right)$$

$$\left. + 0\left(\frac{R_2\tan\lambda}{h_0}\right)^3 + \cdots\right\} \qquad [37]$$

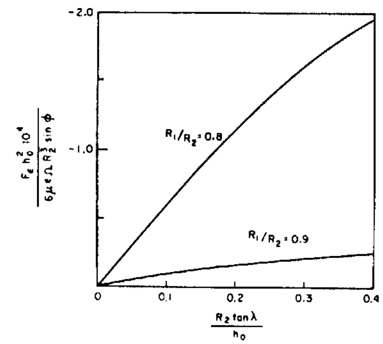

Fig. 3—Laminar force coefficient.

In order to determine the effects of higher order terms the expansion was actually carried out to include the $(R_2\tan\lambda/h_0)^3$ terms. The calculated results are shown in Fig. 3. As in the leakage calculations the deviation from the linear approximation is negligible so that Eq. [37] can be used very good accuracy in the range $0.8 < R_1/R_2 < 0.9$.

Once again the terms of order $\tan^2\lambda$ in Eq. [34] do not affect the performance.

Torque

The torque is obtained from the formula

$$T = \int_{R_1}^{R_2}\int_0^{2\pi}\tau_{z\theta}r^2\,dr\,d\theta = \mu\int_{R_1}^{R_2}\int_0^{2\pi}\frac{\partial v}{\partial z}r^2\,dr\,d\theta \qquad [38]$$

where

$$\frac{\partial v}{\partial z} = \frac{r}{h}(\omega\cos\lambda - \Omega) \qquad [39]$$

so that

$$T = \mu(\omega\cos\lambda - \Omega)\int_{R_1}^{R_2}\int_0^{2\pi}\frac{r^3\,dr\,d\theta}{h} \qquad [40]$$

Recalling that

$$h = h_0\left(1 - \frac{r}{h_0}\cos\theta\tan\lambda\right) \qquad [41]$$

the term by term integration of Eq. [40] yields

$$T = \frac{2\pi\mu}{h_0}(\omega\cos\lambda - \Omega)\left[\frac{R_2{}^4 - R_1{}^4}{4}\right.$$

$$\left. + \frac{1}{12}(R_2{}^6 - R_1{}^6)\left(\frac{\tan\lambda}{h_0}\right)^2 + \ldots\right] \qquad [42]$$

This is the same result as obtained in the concentric case (*1*) which is not surprising since the torque depends only on the tangential velocity profile which was taken to be the same as in the concentric case.

TURBULENT FLOW

Velocity Distribution

In the foregoing analysis of laminar flow the small radial velocity component introduced by the boundary at $z = h$ (see Eq. 10) was retained throughout the analysis. Retaining this term effects the final pressure distribution in two ways. The first is a direct effect which results from the contribution which this term makes to the radial velocity profile, Eq. [11]. The second is an indirect effect which results from a squeeze-film action introduced by the continuity equation, Eqs. [13] and [15]. This latter effect can be eliminated almost immediately on an order-of-magnitude basis, Eq. [22]. The former effect, however, remains in the final equation for the pressure distribution, Eq. [23].

Fortunately, when the gross performance quantities such as leakage, load and torque are evaluated, even the direct effect of the boundary radial velocity disappears because it is self-cancelling for the seal as a whole. The result is that higher order terms involving the tilt angle appear in the equations for leakage, load, and torque only as a consequence of geometric integrals such as appear in Eqs. [32] and [37].

To put it another way, from an analytical point of view the rotation of the upper surface in Fig. 1 is an unnecessary complication which does not contribute to the effects being studied. The absence of ω from Eqs. [32], [37] and [42] confirms this assertion. For this reason the turbulent investigation will be carried-out assuming that $\omega = 0$ which will greatly simplify the analysis.

It was shown in (*2*) that the tangential velocity in turbulent flow could be represented by

$$v_1 = r\Omega\left[1 - \frac{1}{2}\left(\frac{z}{h/2}\right)^{1/7}\right], 0 \le z \le h/2$$

$$v_2 = \frac{r\Omega}{2}\left(\frac{h - z}{h/2}\right)^{1/7}, h/2 \le z \le h \qquad [43]$$

satisfying the boundary conditions

$$\overline{V}_\omega = 0$$

or

$$v = r\Omega, z = 0$$
$$v = 0, z = h \qquad [44]$$

Figure 4 shows the assumed profile.

The radial velocity profile was also shown in (*2*) to be given by

$$u\tau_{z\theta_w} = v\left[f(r,\theta) + \frac{\partial p}{\partial z} - \frac{\rho}{r}\int v^2\,dz\right]$$

$$+ \int\frac{\rho}{r}v^3\,dz - \int v\frac{\partial p}{\partial r}\,dz + g(r,\theta) \qquad [45]$$

The radial velocity boundary conditions are given by Eq. [7], i.e.,

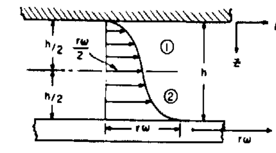

Fig. 4—Turbulent velocity profile.

$$u = \Omega\epsilon\sin(\phi - \theta), z = 0$$
$$u = 0, z = h \qquad [46]$$

With these boundary conditions Eq. [45] becomes

$$u\tau_{z\theta_w} = v\left\{\frac{\epsilon}{r}\tau_{z\theta_w}\sin(\phi - \theta)\right.$$

$$+ \frac{\rho}{r}\left[\frac{1}{r\Omega}\int_0^h v^3\,dz - \int_0^z v^2\,dz\right]$$

$$\left. + \frac{\partial p}{\partial r}\left[z - \frac{1}{r\Omega}\int_0^h v\,dz\right]\right\}$$

$$+ \frac{\rho}{r}\left[\int_0^z v^3\,dz - \int_0^h v^3\,dz\right]$$

$$+ \frac{\partial p}{\partial r}\left[\int_0^h v\,dz - \int_0^z v\,dz\right] \qquad [47]$$

When Eqs. [43] are inserted into Eq. [47] the radial velocity can be written in the form

$$u\tau_{z\theta_w} = \frac{r\Omega}{2}\left(\frac{h}{2}\right)\frac{\partial p}{\partial r}F\left(\frac{z}{h/2}\right) - \frac{\rho}{r}\left(\frac{r\Omega}{2}\right)^3\frac{h}{2}$$

$$G\left(\frac{z}{h/2}\right) + \left[\frac{\upsilon\epsilon\tau_{z\theta_w}}{r}\right]\sin(\phi - \theta) \qquad [48]$$

where

$0 \le z \le \frac{h}{2}$

$$F\left(\frac{z}{h/2}\right) = \left(\frac{z}{h/2}\right)^{1/7}\left[1 - \frac{1}{8}\left(\frac{z}{h/2}\right)\right]$$

$$G\left(\frac{z}{h/2}\right) = \left(\frac{z}{h/2}\right)^{1/7}\left[\frac{13}{12} - \frac{1}{2}\left(\frac{z}{h/2}\right)^{8/7}\right.$$

$$\left. + \frac{7}{18}\left(\frac{z}{h/2}\right)^{9/7} - \frac{7}{90}\left(\frac{z}{h/2}\right)^{10/7}\right]$$

$\frac{h}{2} \le z \le h$

$$F\left(\frac{z}{h/2}\right) = \left(\frac{h - z}{h/2}\right)^{1/7}\left[\frac{3}{4} + \frac{1}{8}\left(\frac{z}{h/2}\right)\right]$$

$$G\left(\frac{z}{h/2}\right) = \left(\frac{h - z}{h/2}\right)^{1/7}\left[\frac{35}{36} - \frac{7}{90}\left(\frac{h - z}{h/2}\right)^{9/7}\right] \qquad [49]$$

Pressure Distribution

When the tangential and radial velocity profiles given above are used to evaluate the integral form of the continuity Eq. [12], the radial pressure gradient is found to be

$$\frac{\partial p}{\partial r} = \frac{835}{816}\rho r\left(\frac{\Omega}{2}\right)^2 - \frac{\tau_{z\theta_w}}{\frac{49}{30}\left(\frac{\Omega}{2}\right)\left(\frac{h}{2}\right)^2}\left[\frac{f(\theta)}{r^2} + \frac{\Omega h\epsilon \sin(\phi-\theta)}{2r} + \frac{r\Omega \sin\theta \tan\lambda}{6}\right] \quad [50]$$

It was shown in (2) that the tangential shear stress $\tau_{z\theta}$ did not vary across the clearance h, and to be consistent with the assumed tangential velocity profile Eq. [43] must be given by

$$\frac{\tau_{z\theta_w}}{\frac{r\Omega}{2}\left(\frac{h}{2}\right)^2} = -\frac{0.0225\rho(\nu)^{1/4}\left|\frac{r\Omega}{2}\right|^{3/4}}{\left(\frac{h}{2}\right)^{9/4}} \quad [51]$$

Equation [50] may now be integrated with respect to r, so that

$$p = \frac{835}{816}\left(\frac{\rho r^2}{2}\right)\left(\frac{\Omega}{2}\right)^2 + \phi(\theta)\int\frac{dr}{r^{1/4}(h/2)^{9/4}} + \frac{30}{49}\left[\frac{0.0225\rho(\nu)^{1/4}\left|\frac{\Omega}{2}\right|^{3/4}\Omega\sin\theta\tan\lambda}{6}\right]\int\frac{r^{11/4}}{(h/2)^{9/4}}dr + \frac{30}{49}\left[0.0225\rho(\nu)^{1/4}\left|\frac{\Omega}{2}\right|^{3/4}\Omega\epsilon\sin(\phi-\theta)\right]\int\frac{r^{3/4}}{(h/2)^{5/4}}dr + \psi(\theta) \quad [52]$$

where for convenience

$$\phi(\theta) = \frac{30}{49}\left[0.0225\rho(\nu)^{1/4}\left|\frac{\Omega}{2}\right|^{3/4}\right]f(\theta) \quad [53]$$

Defining the integrals

$$I_1(r,\theta) = \int\frac{r^{11/4}}{(h/2)^{9/4}}dr$$
$$I_2(r,\theta) = \int\frac{dr}{r^{1/4}(h/2)^{9/4}} \quad [54]$$
$$I_3(r,\theta) = \int\frac{r^{3/4}}{(h/2)^{5/4}}dr$$

and using the boundary conditions of Eq. [24], yields for the arbitrary functions

$$\phi(\theta) = \left[\frac{1}{I_2(R_1,\theta) - I_2(R_2,\theta)}\right]\Bigg\{P_1 - P_2 + \frac{835}{816}\left(\frac{\rho}{2}\right)\left(\frac{\Omega}{2}\right)^2(R_2^2 - R_1^2) + \frac{10}{49}\left[0.0225\rho(\nu)^{1/4}\left|\frac{\Omega}{2}\right|^{3/4}\left(\frac{\Omega}{2}\right)\sin\theta\tan\lambda\right]\times\left[I_1(R_2,\theta) - I_1(R_1,\theta)\right] + \frac{60}{49}\left[0.0225\rho(\nu)^{1/4}\left|\frac{\Omega}{2}\right|^{3/4}\times\left(\frac{\Omega}{2}\right)\epsilon\sin(\phi-\theta)\right]\left[I_3(R_2,\theta) - I_3(R_1,\theta)\right]\Bigg\} \quad [55]$$

and

$$\psi(\theta) = P_2 - \frac{835}{816}\left(\rho\frac{R_2^2}{2}\right)\left(\frac{\Omega}{2}\right)^2 - \frac{10}{49}\left[0.0225\rho(v)^{1/4}\left|\frac{\Omega}{2}\right|^{3/4}\times\left(\frac{\Omega}{2}\right)\sin\theta\tan\lambda\right]I_1(R_2,\theta) - \phi(\theta)I_2(R_2,\theta) - \frac{60}{49}\left[0.0225\rho(v)^{1/4}\left|\frac{\Omega}{2}\right|^{3/4}\times\left(\frac{\Omega}{2}\right)\epsilon\sin(\phi-\theta)\right]I_3(R_2,\theta) \quad [56]$$

Leakage Rate

The volumetric leakage rate evaluated from Eq. [27] is

$$Q = -\int_0^{2\pi} f(\theta)\,d\theta = -\frac{1}{\frac{30}{49}\left[0.0225\rho(\nu)^{1/4}\left|\frac{\Omega}{2}\right|^{3/4}\right]}\int_0^{2\pi}\phi(\theta)\,d\theta \quad [57]$$

The contribution to the pumping due to eccentricity is

$$Q\epsilon = \epsilon\Omega\int_0^{2\pi}\sin(\phi-\theta)\left[\frac{I_3(R_2,\theta) - I_3(R_1,\theta)}{I_2(R_2,\theta) - I_2(R_1,\theta)}\right]d\theta \quad [58]$$

Expanding the integrand as in the laminar case, and integrating yields

$$Q_\epsilon = \pi\epsilon\left(\frac{\Omega}{2}\right)R_2h_0\sin\phi\left\{\frac{3}{7}\left[\frac{1-\left(\frac{R_1}{R_2}\right)^{7/4}}{1-\left(\frac{R_1}{R_2}\right)^{3/4}}\right]\times\left[\frac{35}{44}\left(\frac{1-\left(\frac{R_1}{R_2}\right)^{11/4}}{1-\left(\frac{R_1}{R_2}\right)^{7/4}}\right) - \frac{27}{28}\left(\frac{1-\left(\frac{R_1}{R_2}\right)^{7/4}}{1-\left(\frac{R_1}{R_2}\right)^{3/4}}\right)\right]\left(\frac{R_2\tan\lambda}{h_0}\right) + 0\left(\frac{R_2\tan\lambda}{h_0}\right)^3 + \cdots\right\} \quad [59]$$

Once again the expansion was actually carried-out to obtain the coefficient of $(R_2\tan\lambda/h_0)^3$. The results of these computations are shown in Fig. 5 including the contributions of the cubic terms, which are obviously negligible within the range of the expansion.

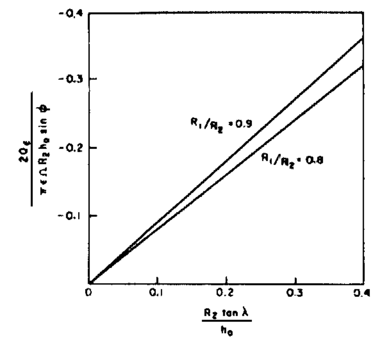

Fig. 5—Turbulent flow coefficient.

Separating Force

The actual pressure distribution in the seal is given by

$$p - P_2 = \frac{835}{816}\left(\frac{\Omega}{2}\right)^2\frac{P}{2}\left(r^2 - R_2^2\right) + \frac{10}{49}\left[0.0225P(v)^{1/4}\left|\frac{\Omega}{2}\right|^{3/4}\times\left(\frac{\Omega}{2}\right)\sin\theta\tan\lambda\right]\left[I_1(r,\theta) - I_1(R_2,\theta)\right] + \frac{60}{49}\left[0.0225\rho(\nu)^{1/4}\left|\frac{\Omega}{2}\right|^{3/4}\left(\frac{\Omega}{2}\right)\epsilon\sin(\phi-\theta)\right]\left[I_3(r,\theta) - I_3(R_2,\theta)\right] + \left[\frac{I_2(r,\theta) - I_2(R_2,\theta)}{I_2(R_1,\theta) - I_2(R_2,\theta)}\right]\Bigg\{P_1 - P_2 + \frac{835}{816}\frac{\rho}{2}\left(\frac{\Omega}{2}\right)^2(R_2^2 - R_1^2) + \frac{10}{49}\left[0.0225\rho(\nu)^{1/4}\left|\frac{\Omega}{2}\right|^{3/4}\left(\frac{\Omega}{2}\right)\sin\theta\tan\lambda\right]\left[I_1(R_2,\theta) - I_1(R_1,\theta)\right] + \frac{60}{49}\left[0.0225\rho(\nu)^{1/4}\left|\frac{\Omega}{2}\right|^{3/4}\left(\frac{\Omega}{2}\right)\epsilon\sin(\phi-\theta)\right]\left[I_3(R_2,\theta) - I_3(R_1\theta)\right]\Bigg\} \quad [60]$$

Considering only those terms associated with eccentricity, the separating force contributed by them is

$$F_\epsilon = \frac{60}{49}\left[0.0225\rho(\nu)^{1/4}\left|\frac{\Omega}{2}\right|^{3/4}\left(\frac{\Omega}{2}\right)\epsilon\right]\int_{R_1}^{R_2}\int_0^{2\pi}\sin(\phi-\theta) \quad [61]$$
$$\times\left\{\left[I_3(r,\theta) - I_3(R_2,\theta)\right] + \left[I_3(R_2,\theta) - I_3(R_1,\theta)\right]\left[\frac{I_2(r,\theta) - I_2(R_2,\theta)}{I_2(R_1,\theta) - I_2(R_2,\theta)}\right]\right\}d\theta\, r\,dr$$

After expanding the integrand and integrating, this contribution to the separating force becomes

$$F_\epsilon = \frac{60}{49}\left[\frac{0.0225\rho(\nu)^{1/4}\left|\frac{R_2\Omega}{2}\right|^{3/4}\left(\frac{R_2\Omega}{2}\right)\pi\epsilon R_2^2\sin\phi}{(h_0/2)^{5/4}}\right]\times$$
$$\Bigg\{\Bigg\{\frac{5}{11}\left[\frac{4}{19}\left(1-\left(\frac{R_1}{R_2}\right)^{19/4}\right) - \frac{1}{2}\left(\frac{R_1}{R_2}\right)^{11/4}\left(1-\left(\frac{R_1}{R_2}\right)^2\right)\right]$$
$$-\frac{27}{49}\left[\frac{4}{15}\left(1-\left(\frac{R_1}{R_2}\right)^{15/4}\right) - \frac{1}{2}\left(\frac{R_1}{R_2}\right)^{7/4}\left(1-\left(\frac{R_1}{R_2}\right)^2\right)\right]\left[\frac{1-\left(\frac{R_1}{R_2}\right)^{7/4}}{1-\left(\frac{R_1}{R_2}\right)^{3/4}}\right]$$
$$-\left[\frac{4}{11}\left(1-\left(\frac{R_1}{R_2}\right)^{11/4}\right) - \frac{1}{2}\left(\frac{R_1}{R_2}\right)^{3/4}\left(1-\left(\frac{R_1}{R_2}\right)^2\right)\right]\times$$
$$\left[\frac{5}{11}\left(\frac{1-\left(\frac{R_1}{R_2}\right)^{11/4}}{1-\left(\frac{R_1}{R_2}\right)^{3/4}}\right) - \frac{27}{49}\left(\frac{1-\left(\frac{R_1}{R_2}\right)^{7/4}}{1-\left(\frac{R_1}{R_2}\right)^{3/4}}\right)^2\right]\Bigg\}\left(\frac{R_2\tan\lambda}{h_0}\right) + 0\left(\frac{R_2\tan\lambda}{h_0}\right)^3 + \cdots\Bigg\} \quad [62]$$

The results of the calculation including the cubic terms are shown in Fig. 6 where it is apparent that F_ϵ is also nearly linear in $(R_2\tan\lambda/h_0)$ as in the laminar case.

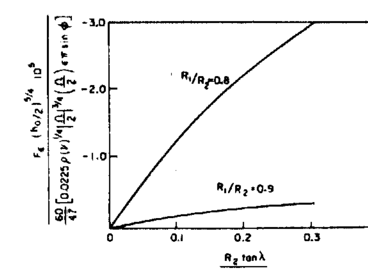

Fig. 6—Turbulent force coefficient.

Torque

The torque calculated from Eq. [38] with the aid of Eq. [51] is independent of eccentricity and therefore the same as given for the concentric case, i.e.,

$$T = 0.00335\pi\left(\frac{\rho^3\mu\Omega^7}{h_0}\right)^{1/4}(R_2^{19/4} - R_1^{19/4}) \times \left\{1 + \frac{19}{27}\left(\frac{5}{64}\right)\left[\frac{1-\left(\frac{R_1}{R_2}\right)^{27/4}}{1-\left(\frac{R_1}{R_2}\right)^{19/4}}\right]\left(\frac{R_2 \tan\lambda}{h_0}\right)^2 + \cdots\right\} \quad [63]$$

CONCLUSIONS

In all the cases so far studied, geometric deviations from the ideal have always had the greatest effect on flow, a modest effect on torque, and a negligible effect on the separating force. This order is not followed by the contributions of eccentricity, where there is no torque contribution what-so-ever.

Another interesting peculiarity of the eccentricity effect is its dependence on some other geometric deviation, such as misalignment. This means that an aligned seal whose boundaries are completely immersed in a sealant will experience no changes in its gross performance if it is made eccentric.

Even more surprising than the geometric coupling requirement just described is the discovery that Q is purely a kinematic effect, independent of the fluid properties. As a consequence, the effect of eccentricity on the separating force can be altered by changing the fluid, but not the effect of eccentricity on the leakage.

In the same vein, it should be noted that the sign of the eccentricity effects can be changed by changing the phase angle ϕ, or reversing the rotational direction of Ω. Since both ϕ and ϵ are extremely difficult to measure, eccentricity in a seal might be more readily detected by observing whether the leakage rate changes when the rotational direction is reversed, all other things unchanged.

ACKNOWLEDGEMENT

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CHAPTER IV

COMPUTER PROGRAMS

For analysis and further discussion of the programs included in this section refer to Chapter 3 Section D. Equations to be referenced are also from that section.

PROGRAM NAME: Short Bearing Seal Program

MODE: Time Sharing System

LANGUAGE: Fortran

INPUT VARIABLES: (Starting with line number 1010 of main program)

IMAX		number of initial geometries
KMAX		number of cases for each geometry
RI	$Ri = \frac{ri}{ro}$	
PO	Po	pressure outside seal
PI	Pi	pressure inside seal
DELTA		increment in theta
KPRINT		printing interval
ALAM	Λ	seal number - $\frac{6\mu\omega}{Po}\left(\frac{ro}{hi}\right)^2$
EP	ϵ	waviness parameter
PC		pressure inside cavity

OUTPUT VARIABLES:

THAPR		theta value at cavity inception point
RA		radius at cavity inception point
THCRPR		theta value at intersection of upstream and downstream
R1CRIT		inside radius at boundary intersection point
R2CRIT		outside radius at boundary intersection point
CL	C_L	total load coefficient
QI	q_i	total flow at inner radius
QO	q_o	total flow at outer radius

Also printed out as a result of stepwise calculations are the values of theta and the corresponding values of R_1 and R_2 that describe both the upstream and downstream boundaries of the cavity. The load coefficients and flows at the inside and outside boundaries are also included for each value of theta. For the definition of these loads and flows refer to section 3D.

SHORT BEARING SEAL PROGRAM

```
5 DIMENSION Y(200),RY(200),THETA(51),RAD(2,51)
10 CØMMØN ALAM,PØ,PI,PC,RI,EP,DELTA,KPRINT,PØMPC,PIMPC,ALAMD4,
11 + RA,YA,THCRIT,YCRIT,THA,R2CRIT,RICRIT,RMIN,RMAX,NMAX,NMAXI,RY,
12 + Y,THETA,RAD,I,RI,R2,SIGCL,SIGØI,SIGQØ,CLA,QIA,QØA,DELTH,
13 + R,TERM1,TERM2,TERM3,TERM4,TERM5
15 READ,IMAX
17 DØ 1500 III=1,IMAX
20 20 READ,KMAX,RI,PØ,PI,DELTA,KPRINT
22 DØ 1400 KKK=1,KMAX
24 READ,ALAM,EP,PC
30 PRINT "      SHØRT BEARING ANALYSIS      -      CØNTAINED CAVITY"
40 PRINT" LAMBDA        PØ         PI          PC            RI       ",
41 PRINT,"EPSILØN"
50 PRINT 60,ALAM,PØ,PI,PC,RI,EP
60 60 FØRMAT(F7.0,2F10.2,F10.4,2F10.2)
80 PRINT"DELTA THETA       PRINT EVERY"
90 PRINT 100,DELTA,KPRINT
100 100 FØRMAT(F7.2,I15)
120 PØMPC=PØ-PC
130 PIMPC=PI-PC
140 ALAMD4=ALAM/4.
150 TRI2=2.*RI**2
160 R=(1.+RI)/2.
170 DELR=(1.-RI)/4.
180 DØ 290 N=1,20
200 CALL TERMS1
210 CALL TERMS2
220 Y2=PØMPC/(ALAMD4*TERM5*(2.*R+TERM4/TERM5))
230 Y1=-PIMPC/(ALAMD4*TERM3*(2.*R+TERM1/TERM3))
240 IF(Y2-Y1) 250,300,270
250 250 R=R+DELR
260 GØ TØ 280
270 270 R=R-DELR
280 280 DELR=DELR/2.
290 290 CØNTINUE
300 300 RA=R
310 YA=0.
320 TH=3.1415926535/2.
330 DELTH=3.1415926535/2.
340 DØ 410 N=1,20
350 E2H=3.*EP*(SIN(2.*TH))**2+CØS(2.*TH)*(1.+EP*CØS(2.*TH))
360 IF (E2H) 370,420,390
370 370 TH=TH+DELTH
380 GØ TØ 400
390 390 TH=TH-DELTH
400 400 DELTH=DELTH/2.
```



```
410 410 CØNTINUE
420 420 THCRIT=TH
430 YCRIT=-2.*EP*SIN(2.*THCRIT)/(1.+EP*CØS(2.*THCRIT))**3
440 TH=3.1415926535/2.
450 DELTH=(THCRIT-TH)/2.
460 DØ 530 N=1,20
470 YTEST=-2.*EP*SIN(2.*TH)/(1.+EP*CØS(2.*TH))**3
480 IF (YA-YTEST) 490,540,510
490  490 TH=TH-DELTH
500 GØ TØ 520
510 510 TH=TH+DELTH
520 520 DELTH=DELTH/2.
530 530 CØNTINUE
540 540 THA=TH
560 R=(1.+RA)/2.
570 DELR=(1.-RA)/4.
580 DØ 660 N=1,20
590 CALL TERMS2
600 Y2=PØMPC/(ALAMD4*TERM5*(2.*R+TERM4/TERM5))
610 IF (YCRIT-Y2) 620,670,640
620 620 R=R-DELR
630 GØ TØ 650
640 640 R=R+DELR
650 650 DELR=DELR/2.
660 660 CØNTINUE
670 670 R2CRIT=R
680 R=(RA+RI)/2.
690 DELR=(RA-RI)/4.
700 DØ 790 N=1,20
710 CALL TERMS1
720 Y1=-PIMPC/(ALAMD4*TERM3*(2.*R+TERM1/TERM3))
730 IF (YCRIT-Y1) 740,790,760
740 740 R=R+DELR
750 GØ TØ 770
760 760 R=R-DELR
770 770 DELR=DELR/2.
780 780 CØNTINUE
790 790 RICRIT=R
800 PRINT"    THETA A          RA"
805 THAPR=THA*57.2958
810 PRINT 820,THAPR,RA
820 820 FØRMAT(F9.4,F12.4)
830 PRINT" THETA CRIT.    RI CRIT.     R2 CRIT."
835 THCRPR=THCRIT*57.2958
840 PRINT 850,THCRPR,RICRIT,R2CRIT
850 850 FØRMAT(F9.4,2F12.4)
860 PRINT" "
870 PRINT"                         UPSTREAM BØUNDARY"
880 PRINT"  THETA      R1         R2         CL1          CL2
881 PRINT"QI             Q2"
890 IF(RICRIT-RI-.000001) 900,900,910
```

```
900 900 RMIN=(RICRIT+RI)/2.
910 GO TO 930
920 920 RMIN=RICRIT-.000001
930 930 IF(1.0-R2CRIT-.000001)940,940,960
940 940 RMAX=(R2CRIT+1.0)/2.
950 GO TØ 970
960 960 RMAX=R2CRIT+.000001
970 970 NMAX=200
980 FNMAX=NMAX
990 NMAX1=FNMAX*(RA-RMIN)/(RMAX-RMIN)+1.
1000 IF(NMAX1-10)1010,1020,1020
1010 1010 NMAX1=10
1020 1020 IF(NMAX-NMAX1-10) 1030,1040,1040
1030 1030 NMAX1=NMAX-10
1040 1040 R=RMIN
1050 NMAX11=NMAX1-1
1060 DEN=NMAX11
1070 DELR=(RA-RMIN)/DEN
1080 DØ 1130 N=1,NMAX11
1090 RY(N)=R
1100 CALL TERMS1
1110 Y(N)=-PIMPC/(ALAMD4*TERM3*(2.*R+TERM1/TERM3))
1120 R=R+DELR
1130 1130 CØNTINUE
1140 R=RA
1150 DEN=NMAX-NMAX1
1160 DELR=(RMAX-RA)/DEN
1170 DØ 1220 N=NMAX1,NMAX
1180 RY(N)=R
1190 CALL TERMS2
1200 Y(N)=PØMPC/(ALAMD4*TERM5*(2.*R+TERM4/TERM5))
1210 R=R+DELR
1220 1220 CØNTINUE
1225 CALL PART2
1230 PRINT↑↑↑↑↑" "
1400 1400 CØNTINUE
1500 1500 CØNTINUE
2000 $USE SUB1A
2010 $USE SUB2A
2020 $USE SUB3A
2030 $USE SUB4A
2040 $USE SUB5A
2050 $USE SUB7A
2060 $USE SUB8A
3000 $DATA
3010 1,
3020 1,.8,2,1,2,10,
3030 10000,.8,.99
```



```
10 SUBRØUTINE PART2
11 DIMENSIØN Y(200),RY(200),THETA(51),RAD(2,51)
15 CØMMØN ALAM,PØ,PI,PC,RI,EP,DELTA,KPRINT,PØMPC,PIMPC,ALAMD4,
16 + RA,YA,THCRIT,YCRIT,THA,R2CRIT,R1CRIT,RMIN,RMAX,NMAX,NMAX1,RY,
17 + Y,THETA,RAD,T,R1,R2,SIGCL,SIGQI,SIGQØ,CLA,QIA,QØA,DELTH
20 LMAX=5
1230 THETA(1)=THA
1235 T=THETA(1)
1240 RAD(1,1)=RA
1245 R1=RAD(1,1)
1250 RAD(2,1)=RA
1255 R2=RAD(2,1)
1260 CALL INTEGR(CL1,CL2,QI,QØ)
1270 THPR=THETA(1)*57.2958
1280 PRINT 1290,THPR,RAD(1,1),RAD(2,1),CL1,CL2,QI,QØ
1290 1290 FØRMAT(F8.4,2F8.5,4E12.4)
1300 SIGCL=0.
1310 SIGQI=0.
1320 SIGQØ=0.
1330 CLA=CL1+CL2
1340 QIA=QI
1350 QØA=QØ
1360 KØUNT=0
1370 DELTH=(THCRIT-THA)/50.
1380 DØ 1690 N=2,51
1385 NN=N
1390 KØUNT=KØUNT+1
1400 THETA(N)=THETA(N-1)+DELTH
1405 T=THETA(N)
1410 YTEST=-2.*EP*SIN(2.*THETA(N))/(1.+EP*CØS(2.*THETA(N)))**3
1415 IF(Y(1)-YTEST) 1460,1420,1420
1420 1420 DØ 1450 MM=2,NMAX
1430 M=MM
1440 IF(Y(M)-YTEST)1480,1480,1450
1450 1450 CØNTINUE
1460 1460 PRINT" UPSTREAM BØUNDARY.  YTEST NØT FØUND"
1470 RETURN
1480 1480 FRACT=(YTEST-Y(M-1))/(Y(M)-Y(M-1))
1490 RAD(1,N)=RY(M-1)+FRACT*(RY(M)-RY(M-1))
1495 R1=RAD(1,N)
1500 DØ 1530 MM=NMAX1,NMAX
1510 M=MM
1520 IF(Y(M)-YTEST)1530,1550,1550
1530 1530 CØNTINUE
1540 GØ TØ 1460
1550 1550 FRACT=(YTEST-Y(M-1))/(Y(M)-Y(M-1))
1560 RAD(2,N)=RY(M-1)+FRACT*(RY(M)-RY(M-1))
1565 R2=RAD(2,N)
1570 CALL INTEGR(CL1,CL2,QI,QØ)
1580 IF (KØUNT-KPRINT) 1630,1590,1590
1590 1590 KØUNT=0
1610 THPR=THETA(N)*57.2958
1620 PRINT 1290,THPR,RAD(1,N),RAD(2,N),CL1,CL2,QI,QØ
1630 1630 CALL SIGMA(CL1,CL2,QI,QØ)
1690 1690 CØNTINUE
```

```
1700 THPR=THETA(NN)*57.2958
1710 IF(KØUNT) 1720,1740,1720
1720 1720 KØUNT=0
1730 PRINT 1290,THPR,RAD(1,NN),RAD(2,NN),CL1,CL2,QI,QØ
1740 1740 PRINT:"                              DØWNSTREAM BØUNDARY"
1750 PRINT:"  THETA       R1        R2          CL1             CL2        ",
1751 PRINT"    QI              QØ"
1760 PRINT 1765,THPR,RAD(1,NN),RAD(2,NN),CL1,CL2,QI,QØ
1765 1765 FØRMAT(F8.3,2F8.5,4E12.4)
1770 CALL PART3
1771 RETURN
```

```
10 SUBRØUTINE PART3
11 DIMENSIØN Y(200),RY(200),THETA(51),RAD(2,51)
15 CØMMØN ALAM,PØ,PI,PC,RI,EP,DELTA,KPRINT,PØMPC,PIMPC,ALAMD4,
16 + RA,YA,THCRIT,YCRIT,THA,R2CRIT,R1CRIT,RMIN,RMAX,NMAX,NMAX1,RY,
17 + Y,THETA,RAD,T,R1,R2,SIGCL,SIGQI,SIGQØ,CLA,QIA,QØA,DELTH
1755 LMAX=5
1760 KØUNT=0
1765 1765 FØRMAT(F8.3,2F8.5,4E12.4)
1770 RAD1A=R1CRIT
1780 RAD1B=RAD1A
1790 RAD2A=R2CRIT
1800 RAD2B=RAD2A
1810 DRDT1A=0.
1820 DRDT2A=0.
1825 NEND=0
1830 TA=THCRIT
1840 DELTH=DELTA/57.2958
1845 1845 KØUNT=KØUNT+1
1850 TB=TA+DELTH
860 1860 DØ 1940 L=1,LMAX
1870 CALL SLØPE(TB,RAD1B,RAD2B,DRDT1B,DRDT2B)
1880 RAD1B=RAD1A+.5*(DRDT1A+DRDT1B)*DELTH
1890 RAD2B=RAD2A+.5*(DRDT2A+DRDT2B)*DELTH
1891 IF (RAD1B-R1CRIT) 1892,1893,1893
1892 1892 RAD1B=R1CRIT
1893 1893 IF (RAD2B-R2CRIT) 1895,1895,1894
1894 1894 RAD2B=R2CRIT
1895 1895 IF (RAD2B-RAD1B) 1900,1900,1940
1900 1900 NEND=1
1905 DELTHP=(RAD2A-RAD1A)/((DRDT1A+DRDT1B-DRDT2A-DRDT2B)/2.)
1910 TB=TA+DELTHP
1920 RAD2B=RAD2A+DELTHP*(DRDT2A+DRDT2B)/2.
1930 RAD1B=RAD1A+DELTHP*(DRDT1A+DRDT1B)/2.
1940 1940 CØNTINUE
1945 IF(NEND)1955,1955,1950
1950 1950 DELTH=DELTHP
1955 1955 T=TB
```



```
1960 R1=RAD1B
1970 R2=RAD2B
1980 CALL INTEGR(CL1,CL2,QI,QØ)
1985 IF(NEND) 1990,1990,2000
1990 1990 IF (KØUNT-KPRINT) 2030,2000,2000
2000 2000 KØUNT=0
2010 THPR=T*57.2958
2020 PRINT 1765,THPR,R1,R2,CL1,CL2,QI,QØ
2030 2030 CALL SIGMA(CL1,CL2,QI,QØ)
2040 IF (NEND) 2050,2050,2180
2050 2050 RAD1A=RAD1B
2060 RAD2A=RAD2B
2070 DRDT1A=DRDT1B
2080 DRDT2A=DRDT2B
2085 TA=TB
2090 GØ TØ 1845
2100 2100 DELTH=(RAD2A-RAD1A)/((DRDT2A+DRDT2B+DRDT1A+DRDT1B)/2.)
2110 T=TA+DELTH
2120 R2=RAD2A-DELTH*(DRDT2A+DRDT2B)/2.
2130 R1=RAD1A+DELTH*(DRDT1A+DRDT1B)/2.
2140 CALL INTEGR(CL1,CL2,QI,QØ)
2180 2180 SIGCL=2.*SIGCL
290 SIGQI=2.*SIGQI
2200 SIGQØ=2.*SIGQØ
2210 HB=1.+EP*CØS(2.*T)
2215 HB2=HB**2
2220 HA=1.+EP*CØS(2.*THA)
2225 HA2=HA**2
2230 ØMRI2=1.-RI**2
240 PØMPI=PØ-PI
2250 PART1=ØMRI2/(4.*LØG(RI))+RI**2/2.
2255 THDI=THA-T+3.1415926535
2260 PART2=(PØMPI*PART1+PØ*ØMRI2/2.)*THDI
2270 PART3=ALAM*(-ØMRI2*ØMRI2/4.-ØMRI2*PART1)/8.
2280 DELCL=2.*(PART2+PART3*(1./HB2-1./HA2))
2290 PART4=(1.+1.5*EP**2)*2.*THDI+(3.*EP+2.*EP**3/3.)
2291 + *(SIN(2.*THA)-SIN(2.*T))
2300 PART5=.75*EP**2*(SIN(4.*THA)-SIN(4.*T))
2310 PART6=EP**3*(SIN(2.*THA)*(CØS(2.*THA))**2-SIN(2.*T)*(CØS(2.*T
2311 +))**2)/3.
2320 PART7=2.*ALAMD4*(2.*RI**2+ØMRI2/LØG(RI))*(HA-HB)
2322 PART8=2.*ALAMD4*(2.+ØMRI2/LØG(RI))*(HA-HB)
2325 PART9=PØMPI*(PART4+PART5+PART6)/LØG(RI)
2327 DELQI=PART9-PART7
2330 DELQØ=PART9-PART8
2340 CL=SIGCL+DELCL
2350 QI=SIGQI+DELQI
2360 QØ=SIGQØ+DELQØ
2370 PRINT:" "
2380 PRINT 2390,CL
2390 2390 FØRMAT(25H TØTAL LØAD CØEFFICIENT =,E12.4)
2400 PRINT 2410,QI
2410 2410 FØRMAT(29H TØTAL FLØW AT INNER RADIUS =,E12.4)
2420 PRINT 2430,QØ
2430 2430 FØRMAT(29H TØTAL FLØW AT ØUTER RADIUS =,E12.4)
2440 RETURN
```

```
10 SUBRØUTINE SIGMA(CL1,CL2,QI,QØ)
11 DIMENSIØN Y(200),RY(200),THETA(51),RAD(2,51)
15 CØMMØN ALAM,PØ,PI,PC,RI,EP,DELTA,KPRINT,PØMPC,PIMPC,ALAMD4,
16 + RA,YA,THCRIT,YCRIT,THA,R2CRIT,R1CRIT,RMIN,RMAX,NMAX,NMAX1,RY,
17 + Y,THETA,RAD,T,R1,R2,SIGCL,SIGQI,SIGQØ,CLA,QIA,QØA,DELTH
20 SIGCL=SIGCL+.5*(CLA+CL1+CL2)*DELTH
30 SIGQI=SIGQI+.5*(QIA+QI)*DELTH
40 SIGQØ=SIGQØ+.5*(QØA+QØ)*DELTH
50 CLA=CL1+CL2
60 QIA=QI
70 QØA=QØ
80 RETURN
```

```
10 SUBRØUTINE SLØPE(TB,RAD1B,RAD2B,DRDT1B,DRDT2B)
11 DIMENSIØN Y(200),RY(200),THETA(51),RAD(2,51)
15 CØMMØN ALAM,PØ,PI,PC,RI,EP,DELTA,KPRINT,PØMPC,PIMPC,ALAMD4,
16 + RA,YA,THCRIT,YCRIT,THA,R2CRIT,R1CRIT,RMIN,RMAX,NMAX,NMAX1,RY,
17 + Y,THETA,RAD,T,R1,R2,SIGCL,SIGQI,SIGQØ,CLA,QIA,QØA,DELTH,
18 +R,TERM1,TERM2,TERM3,TERM4,TERM5
40 RADIUS=RAD1B
50 K=1
60 60 IF (RADIUS-RA) 70,180,180
70 70 DØ 100 II=2,51
80 I=II
90 IF (RADIUS-RAD(1,I)) 100,110,110
100 100 CØNTINUE
110 110 FRACT=(RADIUS-RAD(1,I-1))/(RAD(1,I)-RAD(1,I-1))
120 TH=THETA(I-1)+FRACT*(THETA(I)-THETA(I-1))
130 130 IF (K-1) 250,140,250
140 140 H1U=1.+EP*CØS(2.*TH)
150 K=2
160 RADIUS=RAD2B
170 GØ TØ 60
180 180 DØ 210 II=2,51
190 I=II
200 IF (RADIUS-RAD(2,I)) 220,220,210
210 210 CØNTINUE
220 220 FRACT=(RADIUS-RAD(2,I-1))/(RAD(2,I)-RAD(2,I-1))
230 TH=THETA(I-1)+FRACT*(THETA(I)-THETA(I-1))
240 GØ TØ 130
250 250 H2U=1.+EP*CØS(2.*TH)
260 HD=1.+EP*CØS(2.*TB)
270 HD3=HD**3
280 DHDTH3=-2.*EP*SIN(2.*TB)*ALAMD4/HD3
290 R=RAD1B
295 CALL TERMS1
300 DRDT1B=-HD3/((HD-H1U)*ALAM)*(PIMPC/TERM3+DHDTH3*(2.*RAD1B
301 + +TERM1/TERM3))
310 R=RAD2B
315 CALL TERMS2
320 DRDT2B=-HD3/((HD-H2U)*ALAM)*(-PØMPC/TERM5+DHDTH3*(2.*RAD2B+
321 + TERM4/TERM5))
330 RETURN
```



```
10 SUBRØUTINE INTEGR(CL1,CL2,QI,QØ)
11 DIMENSIØN Y(200),RY(200),THETA(51),RAD(2,51)
15 CØMMØN ALAM,PØ,PI,PC,RI,EP,DELTA,KPRINT,PØMPC,PIMPC,ALAMD4,
16 + RA,YA,THCRIT,YCRIT,THA,R2CRIT,R1CRIT,RMIN,RMAX,NMAX,NMAX1,RY,
17 + Y,THETA,RAD,T,R1,R2,SIGCL,SIGQI,SIGQØ,CLA,QIA,QØA,DELTH,
18 + R,TERM1,TERM2,TERM3,TERM4,TERM5
50 R=R1
51 CALL TERMS1
55 TERM3R=TERM3/R1
60 R=R2
61 CALL TERMS2
65 TERM5R=TERM5/R2
70 P1=R1**2*((1.-1.*TERM2**2)/(4.*TERM3R)+.5*TERM2**2)
80 HTERM1=TERM1/2.
90 P2=-2.*EP*SIN(2.*T)/(1.+EP*CØS(2.*T))**3
100 CL1=PC*HTERM1-PIMPC*P1+PC*(RA**2/2.-1.*R1**2/2.)
101 + +ALAMD4*(-1.*HTERM1**2-TERM1*P1)*P2
110 P1=TERM4/(4.*TERM5R)+.5*R2**2
120 HTERM4=TERM4/2.
130 CL2=PØ*HTERM4+PØMPC*P1+PC*(R2**2/2.-1.*RA**2/2.)+ALAMD4*
131 + (-1.*HTERM4**2-TERM4*P1)*P2
140 H3=(1.+EP*CØS(2.*T))**3
150 QI=-H3*(PIMPC/TERM3R+ALAMD4*P2*(2.*RI**2+TERM1/TERM3R))
160 QØ=-H3*(-PØMPC/TERM5R+ALAMD4*P2*(2.+TERM4/TERM5R))
170 RETURN
```

```
10 SUBRØUTINE TERMS2
11 DIMENSIØN Y(200),RY(200),THETA(51),RAD(2,51)
15 CØMMØN ALAM,PØ,PI,PC,RI,EP,DELTA,KPRINT,PØMPC,PIMPC,ALAMD4,
16 + RA,YA,THCRIT,YCRIT,THA,R2CRIT,R1CRIT,RMIN,RMAX,NMAX,NMAX1,RY,
17 +Y,THETA,RAD,T,R1,R2,SIGCL,SIGQI,SIGQØ,CLA,QIA,QØA,DELTH,
18 + R,TERM1,TERM2,TERM3,TERM4,TERM5
20 TERM4=1.-1.*R**2
30 TERM5=R*LØG(R)
40 RETURN
```

```
10 SUBRØUTINE TERMS1
11 DIMENSIØN Y(200),RY(200),THETA(51),RAD(2,51)
15 CØMMØN ALAM,PØ,PI,PC,RI,EP,DELTA,KPRINT,PØMPC,PIMPC,ALAMD4,
16 + RA,YA,THCRIT,YCRIT,THA,R2CRIT,R1CRIT,RMIN,RMAX,NMAX,NMAX1,RY,
17 + Y,THETA,RAD,T,R1,R2,SIGCL,SIGQI,SIGQØ,CLA,QIA,QØA,DELTH,
18 + R,TERM1,TERM2,TERM3,TERM4,TERM5
20 TERM1=R**2-1.*RI**2
30 TERM2=RI/R
40 TERM3=R*LØG(TERM2)
50 RETURN
```

PROGRAM NAME: NASA SEALS

MODE: Batch (GE 635)

LANGUAGE: Fortran

INPUT VARIABLES:*

RIRO		ri/ro
BVI		pressure at inside boundary
BVO		pressure at outside boundary
ALD	α	angle of orientation between tilt and radial eccentricity
EPR	ϵ_r	rotational eccentricity
BN	Δ	seal number - $\frac{6\,\mu\omega}{po}\left(\frac{ro}{h_I}\right)^2$
A		film thickness parameter, see equation 5
B		"
M		"
EP	ϵ	waviness parameter
N		number of waves
EPT	ϵ_t	tilt eccentricity
PSID	ψ	see equation 6
EPS	ϵ_s	film thickness parameter - see equation 5
IMAX		increments in R direction
JMAX		increments in θ direction
KMAX		maximum number of iterations
KPRINT		printing interval
EPP		convergence error
ACC		first acceleration factor
ACCS		second acceleration factor

KSHOT		frequency of application of acceleration factor
LAMSI		acceleration factor parameter
NCASE		case number
CAV		pressure in cavity
DELACC		acceleration factor increment
ACCMAX		maximum acceleration factor
OMRAT		omega ratio - = 0 for hydrostatic case
EPTDOT	$\dot{\epsilon}_t$	
PSDDOT	$\dot{\psi}$	
EPSDOT	$\dot{\epsilon}_s$	
MOVECA		number of iterations before first cavity boundary calculation
NCAV		cavity parameter
KMOVE		number of iterations between boundary calculations
KRES		result subroutine option, = 1000
XN		portion of field to be calculated
OPTION		control parameter, = 0

OUTPUT VARIABLES *

The output for this program includes the print out of the pressure field. This is tabulated in an R by theta matrix (see output included in this section). Other outputs include:

Lift coefficient
moments about x and y axes -
resultant moment

orientation of resultant moment -
edge flow distribution (flow at inner and outer radius for JMAX theta values)

total flow inner radius

total flow outer radius

moment coefficient

* The definitions and nomenclature for the variables used in this chapter are the same as those given in Chapter 3 Section D.


```
      COMMON  P1(92,25),P2(92,25),H(92,25),T(92),R(25),A1(92,25),
     1A2(92,25),A3(92,25),A4(92,25),A5(92,25),NN(91),NCASE,RIRO,BVI,
     2BVO,ALD,AL,EPR,A,B,M,EP,N,EPT,PSID,PSI,IMAX,JMAX,KMAX,KPRINT,EPP,
     3ACC,ACCS,KSHOT,LAMSI,CAV,DELACC,ACCMAX,IMAXP1,IMAXP2,JMAXM1,
     4JMAXM2,SIGONE,DELT,DELR,K,ERRMAX,IMAXER,JMAXER,SIGP1,TEST,BN,
     5DELQRI(92),DELQRO(92),EPS,OMRAT,EPTDOT,PSDDOT,PSIDOT,EPSDOT,
     6NCAV(92,25),NCAVB(92,25),PSAVE(92,25),XN
      DIMENSION PSHORT(92,25),X(25),Y(25)
      LOGICAL Q5
      REAL     M,N
      NAMELIST/INPUT/NCASE,RIRO,BVI,BVO,ALD,EPR,A,B,M,EP,N,EPT,PSID,
     1IMAX,JMAX,KMAX,KPRINT,EPP,ACC,ACCS,KSHOT,LAMSI,CAV,DELACC,ACCMAX,
     2BN,EPS,OMRAT,EPTDOT,PSDDOT,EPSDOT,MOVECA,NCAV,KMOVE,KRES,XN,
     3OPTION
      CALL FLGEOF(5,Q5)
      READ  (5,1003)
 1003 FORMAT(80H
     1                               )
      READ  (5,1005) RIRO,BVI,BVO,ALD,EPR,BN
      READ  (5,1005) A,B,M,EP,N,EPT,PSID,EPS
 1005 FORMAT(8E10.2)
      READ  (5,1006) IMAX,JMAX,KMAX,KPRINT,EPP,ACC,ACCS,KSHOT,LAMSI
 1006 FORMAT(4I5,3E10.2,2I5)
      KRES=1000
      OMRAT=0.
      OPTION=0.
      MOVECA=5
      KMOVE=0
      CAV=1.
      DELACC=0.
      ACCMAX=1.
      XN=1.
    1 DO 33 I=1,IMAXP2
      DO 32 J=2,JMAXM1
      NCAV(I,J)=0
   32 CONTINUE
   33 CONTINUE
      READ  (5,INPUT)
      IF (Q5) GO TO 500
      AL=ALD/57.2958
      PSI=PSID/57.2958
      PSIDOT=PSDDOT/57.2958
      KSHOTS=LAMSI*KSHOT
      LAMS=LAMSI
      LAMCAV=1
      IMAXP1=IMAX+1
      IMAXP2=IMAX+2
      JMAXM1=JMAX-1
      JMAXM2=JMAX-2
      DO 2 I=1,91
    2 NN(I)=I
      XYZ=IMAX*JMAXM2
      SIGONE=XYZ*(BVI+BVO)/2.
      IF (ABS(BVI-CAV) .GT. 1.E-20) GO TO 5
      DO 4 I=1,IMAXP2
      NCAV(I,1)=1
```

```
    4 CONTINUE
    5 IF (ABS(BVO-CAV) .GT. 1.E-20) GO TO 7
      DO 6 I=1,IMAXP2
      NCAV(I,JMAX)=1
    6 CONTINUE
    7 WRITE (6,3000)
 3000 FORMAT(76H1                                         STEADY-STATE CHARACTE
     1RISTICS OF FACE SEAL)
      WRITE (6,1002)
 1002 FORMAT(1H0)
      WRITE (6,1003)
      WRITE (6,1001)
 1001 FORMAT(1H )
      WRITE (6,1008)
 1008 FORMAT(17H  GEOMETRY INPUTS)
      WRITE (6,1010)
 1010 FORMAT(120H0           CASE NUMBER  RI OVER RO       P INNER         P
     1OUTER         ALPHA         EPSILON R        LAMBDA                    )
      WRITE (6,1012) NCASE,RIRO,BVI,BVO,ALD,EPR,BN
 1012 FORMAT(I15,F15.4,2F15.3,F15.0,2F15.3)
      WRITE (6,1014)
 1014 FORMAT(120H0            A              B              M              EP
     1SILON          N         EPSILON T          PSI         EPSILON S)
      WRITE (6,1016) A,B,M,EP,N,EPT,PSID,EPS
 1016 FORMAT(1P2E15.3,0P1F15.3,F15.4,F15.0,3F15.4)
      IF (OMRAT .EQ. 0.) GO TO 8
      WRITE (6,1017)
 1017 FORMAT(62H0        OMEGA RATIO      EP T DOT        PSI DOT         EP
     1 S DOT)
      WRITE (6,1019) OMRAT,EPTDOT,PSDDOT,EPSDOT
 1019 FORMAT(2F15.3,F15.0,F15.3)
    8 WRITE (6,1018)
 1018 FORMAT(16H0 COMMAND INPUTS)
      WRITE (6,1020)
 1020 FORMAT(65H0        IMAX       JMAX       KMAX      KPRINT       EP P
     1      ACC)
      WRITE (6,1022) IMAX,JMAX,KMAX,KPRINT,EPP,ACC
 1022 FORMAT(4I10,E15.2,F10.1)
      WRITE (6,1024) ACCS,KSHOT,KSHOTS
 1024 FORMAT(39H0  SOLUTION ACCELERATED BY A FACTOR OF F5.2,7H EVERY I3,
     136HTH. ITERATION STARTING AT ITERATION I3)
      IF (XN .NE. 1.) WRITE (6,1026) XN
 1026 FORMAT(27H0  TOTAL CIRCLE DIVIDED BY F4.1)
      DEN=JMAXM1
      DELTA=(BVO-BVI)/DEN
      VALUE=BVI
      DO 13 I=1,IMAXP2
      P1(I,1)=BVI
      P2(I,1)=BVI
      P1(I,JMAX)=BVO
      P2(I,JMAX)=BVO
   13 CONTINUE
      EMAX=IMAX
      DEN=EMAX*XN
      DELT=6.2831853/DEN
      DEN=JMAXM1
      DELR=(1.-RIRO)/DEN
      DTOVDR=DELT/DELR
      T(1)=0.
      DO 15 I=2,IMAXP2
```



```
      T(I)=T(I-1)+DELT
   15 CONTINUE
      R(1)=RIRO
   16 DO 17 J=2,JMAX
      R(J)=R(J-1)+DELR
   17 CONTINUE
      DELT2=DELT**2
      DELR2=DELR**2
      AC=1./(2.*DELT2)
      BC=.75/DELT
      CC=1./(2.*DELR2)
      DC=1./(4.*DELR)
   18 DO 30 J=1,JMAX
      RJ=R(J)
      RJ2=RJ**2
      DJ=1./DELT2+RJ2/DELR2
      AJ=AC/DJ
      CJ=CC*RJ2/DJ
      CSI=(RJ-RIRO)/(1.-RIRO)
   19 URP=0.
      UTP=RJ
      DURPDR=0.
   20 DO 28 I=1,IMAXP2
      TI=T(I)
      COSTIP=COS(TI+PSI)
      SINTIP=SIN(TI+PSI)
      TERM1=1.+A*CSI+B*CSI**2
      TERM2=EP*COS(N*TI)
      TERM3=EPT*RJ*COSTIP
      H(I,J)=TERM1**M+TERM2+TERM3+EPS
      HIJ=H(I,J)
      DHDT=-EP*N*SIN(N*TI)-EPT*RJ*SINTIP
      DHDR=M*TERM1**(M-1.)*(A+2.*B*CSI)/(1.-RIRO)+EPT*COSTIP
   21 BIJ=BC*DHDT/(DJ*HIJ)
      DIJ=(DC/DJ)*(3.*RJ2*DHDR/HIJ+RJ)
   22 A1(I,J)=AJ+BIJ
      A2(I,J)=AJ-BIJ
      A3(I,J)=CJ+DIJ
      A4(I,J)=CJ-DIJ
   24 IF ((AL .EQ. 0.) .AND. (EPR .EQ. 0.)) GO TO 27
      BE=AL+TI
      SINB=SIN(BE)
      COSB=COS(BE)
      URP=-EPR*COSB
      UTP=RJ+EPR*SINB
      DUTPDT=EPR*COSB
   27 TERM4=URP*(DHDR+HIJ/RJ)+(UTP*DHDT+HIJ*DUTPDT)/RJ
      TERM5=0.
      IF (OMRAT .EQ. 0.) GO TO 26
      TERM5=OMRAT*(EPTDOT*RJ*COSTIP-PSIDOT*EPT*RJ*SINTIP+EPSDOT)
   26 A5(I,J)=(TERM4+TERM5)*RJ2*BN/(2.*DJ*HIJ**3)
   28 CONTINUE
   30 CONTINUE
      LAMK=1
       LAMR=1
      LAMUPA=1
      SIGP2=0.
      K=0
   34 K=K+1
      DO 40 I=1,IMAXP2
```

```
      DO 38 J=2,JMAXM1
      NCAVB(I,J)=0
   38 CONTINUE
   40 CONTINUE
      ERRMAX=0.
      SIGP1=0.
      DO 45 I=2,IMAXP1
      X(1)=0.
      Y(1)= BVI
      DO 310 IXY=2,JMAXM1
      DENOM= 1.-(A4(I,IXY)*X(IXY-1))
      IF(K-1)309,308,309
  308 SUB=(DELR/(R(IXY)*DELT))**2+1.
      SUB3=A3(I,IXY)*SUB
      SUB4=A4(I,IXY)*SUB
      SUB5=A5(I,IXY)*SUB
      SUBDN=1.-(SUB4*X(IXY-1))
      X(IXY)=SUB3/SUBDN
      Y(IXY)=(-SUB5+SUB4*Y(IXY-1))/SUBDN
      GO TO 310
  309 X(IXY)=A3(I,IXY)/DENOM
      Y(IXY)=(A1(I,IXY)*P2(I+1,IXY)+A2(I,IXY)*P2(I-1,IXY)
     1-A5(I,IXY)+A4(I,IXY)*Y(IXY-1))/DENOM
  310 CONTINUE
      DO 44 J=2,JMAXM1
      IF (NCAV(I,J) .EQ. 0) GO TO 43
      P1(I,J)=CAV
      GO TO 42
   43 IBACK=JMAX-J+1
      P1(I,IBACK)=X(IBACK)*P1(I,IBACK+1)+Y(IBACK)
      P1(I,IBACK)=ACC*P1(I,IBACK)+(1.-ACC)*P2(I,IBACK)
      IF(K-1)41,401,41
  401 P2(I,IBACK)=P1(I,IBACK)
   41 IF (P1(I,IBACK) .GE. CAV) GO TO 42
      NCAV(I,IBACK)=1
      P1(I,IBACK)=CAV
   42 SIGP1=SIGP1+ABS(P1(I,IBACK)-PSHORT(I,IBACK))
      ERR=(P1(I,IBACK)-P2(I,IBACK))/P1(I,IBACK)
      IF (ABS(ERR) .LE. ABS(ERRMAX)) GO TO 44
      ERRMAX=ERR
      IMAXER=I
      JMAXER=IBACK
   44 CONTINUE
   45 CONTINUE
      DO 46 J=1,JMAX
      P1(1,J)=P1(IMAXP1,J)
      P1(IMAXP2,J)=P1(2,J)
      P2(1,J)=P1(1,J)
      P2(IMAXP2,J)=P1(IMAXP2,J)
      NCAV(1,J)=NCAV(IMAXP1,J)
      NCAV(IMAXP2,J)=NCAV(2,J)
   46 CONTINUE
      IF(K-1)461,464,461
  461 DO 463 I=1,IMAXP2
      DO 462 J=2,JMAXM1
      P2(I,J)=P1(I,J)
  462 CONTINUE
  463 CONTINUE
  464 TEST=(SIGP1-SIGP2)/SIGP1
      SIGP2=SIGP1
```



```
   47 IF (K+5-KSHOTS) 48,48,51
   48 IF (K-LAMUPA*5) 56,49,56
   49 LAMUPA=LAMUPA+1
      ACC=ACC+DELACC
      IF(K-1) 56,56,50
   50 IF (ACC .GT. ACCMAX) ACC=ACCMAX
      GO TO 56
   51 IF (K-LAMS*KSHOT) 56,52,56
   52 LAMS=LAMS+1
      DO 54 J=2,JMAXM1
      DO 53 I=1,IMAXP2
      P1(I,J)=ACCS*P1(I,J)+(1.-ACCS)*P2(I,J)
   53 CONTINUE
   54 CONTINUE
   56 IF(((K-LAMK*KPRINT) .GE. 0) .OR. (K .EQ. 1)) GO TO 57
      GO TO 74
   57 LAMK=LAMK+1
   58 WRITE (6,1058)
 1058 FORMAT(24H1  PRESSURE DISTRIBUTION)
      IF(K-1) 600,600,60
  600 DO 300 I=1,IMAXP2
      DO 301 J=1,JMAX
      PSHORT(I,J)=P1(I,J)
  301 CONTINUE
  300 CONTINUE
   60 WRITE (6,1060) K,ERRMAX,IMAXER,JMAXER,TEST,ACC,ACCS
 1060 FORMAT(20H0        CYCLE NUMBER I3,21H, MAX. REL. ERROR IS 1P1E12.5,
     16H AT I=I2,7H AND J=I2,7H, TEST=1P1E12.5,7H,  ACC=0P1F6.3,7H, ACCS
     2=F6.3)
      WRITE (6,1001)
      ISTART=1
      IEND=10
      NSWB=0
   65 IF (IMAXP1-IEND) 66,66,68
   66 IEND=IMAXP1
      NSWB=1
   68 WRITE (6,1028) (NN(I),I=ISTART,IEND)
 1028 FORMAT(15H              I=I2,9H        I=I2,9H        I=I2,9H        I
     1=I2,9H        I=I2,9H        I=I2,9H        I=I2,9H        I=I2,9H
     2   I=I2,9H        I=I2)
      WRITE (6,1001)
      DO 70 J=1,JMAX
      ISTART=ISTART
   70 WRITE (6,1030) NN(J),(P1(I,J),I=ISTART,IEND)
 1030 FORMAT(6H      J=I2,F12.7,9F11.7)
      IF (NSWB)74,72,74
   72 WRITE (6,1002)
      ISTART=ISTART+10
      IEND=IEND+10
      GO TO 65
   74 IF((K .GE. LAMR*KRES) .OR. (K .EQ. 1)) GO TO 705
      GO TO 75
  705 LAMR=LAMR+1
      CALL RESULT
   75 IF (K-LAMCAV*MOVECA)255,155,255
  155 LAMCAV=LAMCAV+1
      MTEST=0
      DO 132 I=1,IMAXP2
      DO 130 J=2,JMAXM1
      PSAVE(I,J)=P1(I,J)
```

```
  130 CONTINUE
  132 CONTINUE
      DO 158 J=2,JMAXM1
      DO 156 I=2,IMAXP1
      IF (NCAV(I,J) .EQ. 0) GO TO 156
      IF ((NCAV(I-1,J) .EQ. 0) .OR. (NCAV(I,J+1) .EQ. 0) .OR.
     1(NCAV(I+1,J) .EQ. 0) .OR. (NCAV(I,J-1) .EQ. 0)) NCAVB(I,J)=1
  156 CONTINUE
      NCAVB(1,J)=NCAVB(IMAXP1,J)
      NCAVB(IMAXP2,J)=NCAVB(2,J)
  158 CONTINUE
      DO 205 J=2,JMAXM1
      JJ=J
      ISKIP=0
      DO 200 II=2,IMAXP1
      I=II
      NPTEST=0
      IF (ISKIP .EQ. 0) GO TO 159
      ISKIP=0
      GO TO 200
  159 IF ((NCAV(I,J) .EQ. 1) .AND. (NCAV(I+1,J) .EQ. 0)) GO TO 161
  160 GO TO 200
  161 IF (NCAV(I-1,J) .EQ. 0) GO TO 200
  162 DO 168 ISUB=2,IMAX
      ITRY=I-ISUB
  164 IF (ITRY .LT. 1) ITRY=ITRY+IMAX
  166 IF (NCAV(ITRY,J) .EQ. 0) GO TO 172
  168 CONTINUE
      WRITE (6,1168) J
 1168 FORMAT(40H0 CAVITY PRESSURE EXISTS ALONG ENTIRE J=I2,5H LINE)
      STOP
  172 H1=H(ITRY+1,J)
      H2=H(I,J)
      CORMUL=0.5
      IF ((ABS(P1(I,J+1)-CAV) .LT. 1.E-20) .OR. (ABS(P1(I,J-1)-CAV) .LT.
     1 1.E-20)) CORMUL=1.0
      IP1=I+1
      IF (IP1 .GT. IMAXP2) IP1=IP1-IMAX
      DTDR=-DTOVDR*(P1(I,J+1)-P1(I,J-1))/(P1(IP1,J)-P1(I-1,J))
      DTDR=CORMUL*ABS(DTDR)
  196 TERM1=(H1-H2)*BN/H2**3
      DEN=R(J)**2*DELT
      TERM2=(P1(IP1,J)+P1(I-1,J))/DEN
      TERM3=(P1(I,J+1)+P1(I,J-1))*DTDR/DELR
      TERM4=2./DEN+2.*DTDR/DELR
      PRESS=(TERM1+TERM2+TERM3)/TERM4
  197 IF (KMOVE .NE. 0) GO TO 200
  198 IF (PRESS .LE. CAV) GO TO 175
      PSAVE(I,J)=PRESS
      IF (NCAV(I,J) .EQ. 1) MTEST=1
      NCAV(I,J)=0
      GO TO 200
  175 IF (NPTEST .EQ. 1) GO TO 180
      NPTEST=1
      I=I+1
      GO TO 172
  180 PSAVE(I,J)=CAV
      NCAV(I,J)=1
      MTEST=1
      ISKIP=1
```



```
  200 CONTINUE
  205 CONTINUE
      DO 210 J=2,JMAXM1
      DO 208 I=3,IMAXP2
      IF (ABS(P1(I,J)-CAV) .LT. 1.E-20) NCAV(I,J)=1
  208 CONTINUE
      NCAV(1,J)=NCAV(IMAXP1,J)
      NCAV(2,J)=NCAV(IMAXP2,J)
      PSAVE(1,J)=PSAVE(IMAXP1,J)
      IF (ABS(PSAVE(IMAXP2,J)-P1(IMAXP2,J)) .GT. 1.E-20) PSAVE(2,J)=
     1  PSAVE(IMAXP2,J)
      IF (ABS(PSAVE(2,J)-P1(2,J)) .GT. 1.E-20) PSAVE(IMAXP2,J)=
     1  PSAVE(2,J)
  210 CONTINUE
      DO 222 I=1,IMAXP2
      DO 220 J=2,JMAXM1
      P1(I,J)=PSAVE(I,J)
  220 CONTINUE
  222 CONTINUE
  255 IF (MTEST .NE. 0) GO TO 76
      IF(TEST)759,76,759
  759 IF(ABS(TEST)-ACC*EPP)120,120,76
   76 IF (K .LT. KMAX) GO TO 34
   89 WRITE (6,1099)
 1099 FORMAT(28H1  SOLUTION DID NOT CONVERGE)
  110 WRITE (6,1100)
 1100 FORMAT(46H1  PRESSURE DISTRIBUTION ON NEXT TO LAST CYCLE)
      WRITE (6,1001)
      ISTART=1
      IEND=10
      NSWB=0
  111 IF (IMAXP1-IEND) 112,112,114
  112 IEND=IMAXP1
      NSWB=1
  114 WRITE (6,1028) (NN(I),I=ISTART,IEND)
      WRITE (6,1001)
      DO 116 J=1,JMAX
      ISTART=ISTART
  116 WRITE (6,1030) NN(J),(P2(I,J),I=ISTART,IEND)
      IF (NSWB)120,118,120
  118 WRITE (6,1002)
      ISTART=ISTART+10
      IEND=IEND+10
      GO TO 111
  120 WRITE (6,1200)
 1200 FORMAT(38H1  PRESSURE DISTRIBUTION ON LAST CYCLE)
      WRITE (6,1060) K,ERRMAX,IMAXER,JMAXER,TEST,ACC,ACCS
      WRITE (6,1001)
      ISTART=1
      IEND=10
      NSWB=0
  121 IF (IMAXP1-IEND) 122,122,124
  122 IEND=IMAXP1
      NSWB=1
  124 WRITE (6,1028) (NN(I),I=ISTART,IEND)
      WRITE (6,1001)
      DO 126 J=1,JMAX
      ISTART=ISTART
  126 WRITE (6,1030) NN(J),(P1(I,J),I=ISTART,IEND)
  128 WRITE (6,1002)
```

```
      ISTART=ISTART+10
      IEND=IEND+10
      GO TO 121
   90 CALL RESULT
      GO TO 1
  500 WRITE (6,5000)
 5000 FORMAT(45H1  END OF INPUT DATA.  LAST RUN HAS BEEN MADE)
      STOP
      END
```



```
      SUBROUTINE RESULT
      COMMON  P1(92,25),P2(92,25),H(92,25),T(92),R(25),A1(92,25),
     1A2(92,25),A3(92,25),A4(92,25),A5(92,25),NN(91),NCASE,RIRO,BVI,
     2BVO,ALD,AL,EPR,A,B,M,EP,N,EPT,PSID,PSI,IMAX,JMAX,KMAX,KPRINT,EPP,
     3ACC,ACCS,KSHOT,LAMST,CAV,DELACC,ACCMAX,IMAXP1,IMAXP2,JMAXM1,
     4JMAXM2,SIGONE,DELT,DELR,K,ERRMAX,IMAXER,JMAXER,SIGP1,TEST,BN,
     5DELORI(92),DELORO(92),EPS,OMRAT,EPTDOT,PSDDOT,PSIDOT,EPSDOT,
     6NCAV(92,25),NCAVB(92,25)
      WRITE (6,1000)
 1000 FORMAT(68H1                                                       INPU
     1T CONDITIONS)
      WRITE (6,1002)
 1002 FORMAT(1H0)
      WRITE (6,1005) NCASE,RIRO,BVI,BVO,ALD,EPR
 1005 FORMAT(10H  CASE NO.I4,18H,   RADIUS RATIO =F6.3,13H,   P INNER =F
     16.2,13H,   P OUTER =F6.2,11H,   ALPHA =F6.2,15H,   EPSILON R =F8.5
     2)
      WRITE (6,1010) A,B,M,EP,N,EPT,PSID
 1010 FORMAT(5H0 A =1P1E10.3,9H,     B =1P1E10.3,9H,     M =0P1F7.4,15H,
     1      EPSILON =F8.5,9H,     N =F5.1,17H,     EPSILON T =F8.5,11H,
     2   PSI =F6.2)
      WRITE (6,1015) IMAX,JMAX,K,BN
 1015 FORMAT(29H0 NO. OF ANGULAR INCREMENTS =I3,32H,    NO. OF RADIAL GR
     1ID POINTS =I3,30H,    TOTAL NO. OF ITERATIONS =I4,
     211H, LAMBDA = F8.2)
      WRITE (6,1002)
      WRITE (6,1017)
 1017 FORMAT(71H0                                                      NON-DIME
     1NSIONAL RESULTS)
      NEW=IMAX
      CL=0.
      XMOM=0.
      YMOM=0.
      DELDEL=DELT*DELR
      DO 70 J=1,JMAXM1
      RAV=(R(J)+R(J+1))/2.
      DO 60 I=1,IMAX
      PAVI=(P1(I,J)+P1(I,J+1))/2.
      PAVI1=(P1(I+1,J)+P1(I+1,J+1))/2.
      IF (PAVI-PAVI1)10,15,15
   10 TOP=.5*PAVI+(PAVI1-PAVI)/3.
      GO TO 20
   15 TOP=.5*PAVI1+(PAVI-PAVI1)/6.
   20 BOT=(PAVI+PAVI1)/2.
   25 IF (BOT .GT. .000001) GO TO 30
      FACT=.5
      GO TO 32
   30 FACT=TOP/BOT
   32 TMEAN=T(I)+FACT*DELT
      PAVJ=(P1(I,J)+P1(I+1,J))/2.
      PAVJ1=(P1(I,J+1)+P1(I+1,J+1))/2.
      IF (PAVJ-PAVJ1)35,40,40
   35 TOP=.5*PAVJ+(PAVJ1-PAVJ)/3.
      GO TO 45
   40 TOP=.5*PAVJ1+(PAVJ-PAVJ1)/6.
   45 IF (BOT .GT. .000001) GO TO 50
      FACT=.5
      GO TO 52
   50 FACT=TOP/BOT
```

```
   52 RMEAN=R(J)+FACT*DELR
      DELCL=BOT*RAV*DELDEL
      CL=CL+DELCL
      DELCLR=DELCL*RMEAN
      XMOM=XMOM+DELCLR*SIN(TMEAN)
      YMOM=YMOM-DELCLR*COS(TMEAN)
   60 CONTINUE
   70 CONTINUE
      RMOM=SQRT(XMOM**2+YMOM**2)
      ANGMOM=ATAN2(XMOM,YMOM)
      ANGMOM=57.2958*ANGMOM
      IF (ANGMOM .LT. 0.) ANGMOM=ANGMOM+180.
      WRITE (6,1020) CL
 1020 FORMAT(21H0 LIFT COEFFICIENT = 1P1E12.4)
      WRITE (6,1025) XMOM,YMOM
 1025 FORMAT(24H0 MOMENT ABOUT X-AXIS = 1P1E12.4,47H
     1        MOMENT ABOUT Y-AXIS= 1P1E12.4)
      WRITE (6,1030) RMOM,ANGMOM
 1030 FORMAT(21H0 RESULTANT MOMENT = 1P1E12.4,69H
     1        ORIENTATION OF RESULTANT MOMENT VECTOR = 1P1E12.4,8H DEGREES
     2)
      JMAXM3=JMAX-3
      QIN=0.
      QOUT=0.
      DO 120 I=1,NEW
      IF (EPR .NE. 0.) GO TO 100
      URP=0.
      UTPI=R(1)
      UTPO=R(JMAX)
      GO TO 110
  100 BE=AL+T(I)
      URP=-EPR*COS(BE)
      EPSINB=EPR*SIN(BE)
      UTPI=R(1)+EPSINB
      UTPO=1.+EPSINB
  110 DPDRI=.5*(-4.*P1(I,1)+7.*P1(I,2)-4.*P1(I,3)+P1(I,4))/DELR
      DPDRO=.5*(-P1(I,JMAXM3)+4.*P1(I,JMAXM2)-7.*P1(I,JMAXM1)
     1        +4.*P1(I,JMAX))/DELR
      DELQRI(I)=(BN*H(I,1)*URP-H(I,1)**3*DPDRI)*R(1)*DELT
      DELQRO(I)=(BN*H(I,JMAX)*URP-H(I,JMAX)**3*DPDRO)*DELT
      QIN=QIN+DELQRI(I)
      QOUT=QOUT+DELQRO(I)
  120 CONTINUE
      WRITE (6,1130)
 1130 FORMAT(27H0  EDGE FLOW DISTRIBUTIONS.//)
      ISTART=1
      IEND=10
      NSWB=0
  125 IF (NEW-IEND) 130,130,135
  130 IEND=NEW
      NSWB=1
  135 WRITE (6,1135) (NN(I),I=ISTART,IEND)
 1135 FORMAT(22H                          I=I2,9H        I=I2,9H        I=I2,9H
     1       I=I2,9H        I=I2,9H        I=I2,9H        I=I2,9H        I=I2
     2,9H        I=I2,9H        I=I2)
      WRITE (6,1001)
 1001 FORMAT(1H )
      WRITE (6,1140) (DELQRI(I),I=ISTART,IEND)
```



```
 1140 FORMAT(16H  INNER RADIUS  1P10E11.3)
      WRITE (6,1142) (DELQRO(I),I=ISTART,IEND)
 1142 FORMAT(16H  OUTER RADIUS  1P10E11.3)
      IF (NSWB) 145,140,145
  140 WRITE (6,1002)
      ISTART=ISTART+10
      IEND=IEND+10
      GO TO 125
  145 WRITE (6,1145) QIN,QOUT
 1145 FORMAT(31H0 TOTAL FLOW AT INNER RADIUS = 1P1E13.5,39H             101
     1AL FLOW AT OUTER RADIUS = 1P1E13.5)
      TDELT=2.*DELT
      CF=0.
      DO 165 J=1,JMAXM1
      ESINAL=EPR*SIN(AL)
      UTP=R(J)+ESINAL
      UTP1=R(J+1)+ESINAL
      DPDT=(P1(2,J)-P1(IMAX,J))/TDELT
      DPDT1=(P1(2,J+1)-P1(IMAX,J+1))/TDELT
      PART=(BN*UTP/H(1,J)+3.*H(1,J)*DPDT/R(J))*R(J)**2
      PART1=(BN*UTP1/H(1,J+1)+3.*H(1,J+1)*DPDT1/R(J+1))*R(J+1)**2
      TERM1=.5*(PART+PART1)
      DO 160 I=2,IMAXP1
      IF (EPR .NE. 0.) GO TO 150
      UTP=R(J)
      UTP1=R(J+1)
      GO TO 155
  150 BE=AL+T(I)
      EPSINB=EPR*SIN(BE)
      UTP=R(J)+EPSINB
      UTP1=R(J+1)+EPSINB
  155 DPDT=(P1(I+1,J)-P1(I-1,J))/TDELT
      DPDT1=(P1(I+1,J+1)-P1(I-1,J+1))/TDELT
      PART=(BN*UTP/H(I,J)+3.*H(I,J)*DPDT/R(J))*R(J)**2
      PART1=(BN*UTP1/H(I,J+1)+3.*H(I,J+1)*DPDT1/R(J+1))*R(J+1)**2
      TERM2=.5*(PART+PART1)
      DELCF=.5*(TERM1+TERM2)*DELDEL
      TERM1=TERM2
      CF=CF+DELCF
  160 CONTINUE
  165 CONTINUE
      WRITE (6,1165) CF
 1165 FORMAT(23H0 MOMENT COEFFICIENT = 1P1E13.5)
      RETURN
      END
```

TYPICAL INPUT - NASA SEALS PROGRAM

```
 CAVITY INCEPTION  --  NEW METHOD
.665      10.       10.       0.        0.        0.
0.        0.        0.        0.        1.        .2        0.        0.
  36   13  100  100  1.00E-06 1.        1.              5     5
$INPUT,NCASE=1,BN=10.,EPT=.9,EPR=.01,MOVECA=101$
$INPUT,NCASE=2,BN=1000.,EPT=.2,EPR=.001$
$INPUT,NCASE=3,RIRO=.5,EPR=.01$
$INPUT,NCASE=4,BN=100.,EPT=.5$
$INPUT,NCASE=5,EPT=.9,BN=10.$
$INPUT,NCASE=6,RIRO=.9,BN=1000.,EPT=.2$
$INPUT,NCASE=7,BN=100.,EPT=.5$
$INPUT,NCASE=8,BN=10.,EPT=.9$
$INPUT,NCASE=9,RIRO=.665,BVI=11.,BN=0.,EPT=.2,EPR=0.$
$INPUT,NCASE=10,EPT=.5$
$INPUT,NCASE=11,EPT=.9$
```


CHAPTER V

NON-NEWTONIAN BEHAVIOR

IN FACE SEALS

CHAPTER V

HIGH SHEAR, NON-NEWTONIAN PROPERTIES OF A POLYMER FORTIFIED OIL

L. H. Bernd

I. Summary

Based on an examination of the characteristics of shear viscosity versus shear rate, it was postulated that high tensile and compressive stresses might exist in certain liquids at very high shear rates. If so, this would be important in high speed machine elements, and as a sealing mechanism in radial face seals. Such stresses should be evident in a polymer fortified oil, or in a liquid comprised of molecules possessing an appreciable length to width ratio.

Therefore, a jet reaction viscometer reaching 10^7 sec.$^{-1}$ shear rate was developed. A long chain molecule was used to produce elastic, non-Newtonian effects.

At high shear rates, elastic stresses were found to be dominant with respect to viscous stresses. The life of the long chain molecule was short. Tensile stresses up to more than 1,000 psi were obtained.

II. Introduction

Objective

Interesting results have been obtained by various experimenters over the past decade indicating that certain types of visco-elastic, non-Newtonian fluids produce compressive and tensile stresses when sheared. The tensile stress occurs parallel to the plane of shear, the compressive stress occurs perpendicular to the plane of shear. These stresses are generally termed normal stresses, and are elastic in nature. Fluids in which they occur contain long molecules; for example, solutions of high molecular weight polymers, also gels, molten plastics, silicone oils.

Typically, stress measurements have been made at relatively low shear rates. With liquids of lubricant viscosity (100 cs or less) containing dissolved polymers, the stresses obtained have been low, from a hundredth psi or so, up to a few psi at best. The question arises whether these normal stresses might become appreciable, say in the order of 100 to 1,000 psi, at high shear rates such as occur in high speed bearings or seals. If so, normal stresses might then be used to maintain a bearing load, seal, or lubricate, and perhaps compete with other means of achieving these functions. A practical limitation would be the expectancy of short life for a long chain polymer under high shear.

Thus a normal stress would probably find use in short-lived applications, such as occur in a rocket engine, or a "once through" system.

The objective of this investigation was therefore to develop test equipment to reach high shear rates of from 10^4 to 10^7 reciprocal seconds, go beyond the shear range hitherto reached in normal stress measurements, and determine the magnitude and general characteristics of normal stresses. It must be realized that the subject of normal stresses is somewhat controversial in certain aspects. It was planned that experiments in a rotating geometry follow the present work [21].

To make normal stresses predominate in a fluid for study, the highest molecular weight polyisobutylene available was dissolved in a kerosene grade of mineral oil. Small clearances were used in the test viscometer to approach the clearances found in machine elements; similarly rectangular passages were used (in addition to circular capillaries) as the flow passages.

Normal Stresses of Interest in a Machine Element

Normal stresses were first observed in a rotating geometry by Weissenberg, and reported in 1948 [1]. In a Weissenberg rheogoniometer, the normal stress is termed the force per unit area forcing apart parallel, closely-spaced rotating members in the form of disks shearing the fluid. This is shown in Figure 1. This is the normal stress often referred to in the literature (11, 17, 18). These normal stresses should be useable to provide load bearing between the faces of a mechanical face seal or in a plain thrust bearing. In addition, sealing should be obtainable in a mechanical face seal. That is, fluid is pumped by means of a centripetal force towards the center of rotation if a central outlet is provided. Thus one can oppose a pressure head causing leakage in an outward radial direction.

These effects are due to the creation of a tensile stress τ_{11} in the liquid. The tensile stress acts parallel to the flow streamlines, producing a hoop stress because of the circular flow path, thus creating an inward centripetal pressure component. This results in a pressure head p which increases as one progresses from the outer edge of the rheogoniometer to the center of rotation, summing up the pressure contributions from each element of fluid under tensile stress. That is,

$$\frac{dp}{dr} = - \frac{\tau_{11}}{r} \qquad (1)$$

Then, the maximum head created at the center of the rheogonometer for sealing or load bearing is

$$p_m = \int_{r=0}^{r=R} \frac{\tau_{11}}{r} \, dr \qquad (2)$$

where τ_{11} is a function of the shear rate, which is proportional to the radius for a constant clearance between faces. If sufficiently high rotational speeds and/or radii exist, centrifugal forces must be taken into account since they reduce p_m. The tensile stress produced is elastic, acting to return the deformed fluid and the plates of the rheogonometer back to a previous position.

Normal stresses have also been found in geometries where there is parallel flow without curvature; i.e., where a hoop stress cannot exist to aid in producing pressure. For instance, Sakiades (13) measured the pressure on the sidewall of a tube of circular cross-section within which fluid was flowing (polyvinyl alcohol). A linear extrapolation to the outlet of the tube, as in Figure 2, showed a compressive pressure acting to force the walls apart beyond the static pressure head causing flow. Similar results are reported herein on flow between two parallel flat plates. A reasonable assumption is that this compressive stress also exists when two parallel flat surfaces are translated to shear the fluid. That is, the stress is a result of shear in the fluid, and hence should be independent of the means by which the shear is produced. This compressive stress might be useful to provide load bearing in a slider bearing configuration, or its extension--a journal bearing.

Given the existence of a compressive stress, one expects that the total normal stress in a rotating geometry should actually be separable into two effects: the fluid acting indirectly via a tensile hoop stress to produce the normal stress as previously mentioned, plus the compressive stress acting directly on the walls. Sakiades (13) determined both the tensile stress (τ_{11}) and the compressive stress (p_{22}) acting in flow through a circular tube. Examination of his curves showed that a constant proportionality existed over the range of his tests, i.e.,

$$p_{22} = 0.70 \ \tau_{11} \qquad (3)$$

Somewhat similar results were obtained in our tests for flow between flat plates. Thus in considering a total normal stress i.e., load bearing as a consequence of both effects, the hoop-produced stress should dominate at small radii, while the compression-produced stress should predominateat large radii. Weissenberg or plate and cone rheogoniometers, generally speaking, are often small so that the compressive stress could go unnoticed if both load bearing mechanisms were present.

Low Shear Values of Normal Stresses

Much of the experimental work to date has been done with rotary viscometers at low shear rates below 10^2 to 10^3 sec.$^{-1}$. Generally speaking, at low shear rates the normal stress is proportional to the shear rate squared.

$$\tau_{11} = k\, s^n \quad , \text{where } n = 2 \qquad (4)$$

Using jet viscometers, various investigators (2, 3, 4, 16) have reached a region of higher shear where the stress is approximately proportional to the shear rate; $n = 1$. The maximum shear rate reported is about 10^5 sec.$^{-1}$.

Tensile stress values obtained in this linear region are given in Table I for a shear rate of 10^4 sec.$^{-1}$ for purposes of comparison. From this, it can be seen that a molecule must have an appreciable length to diameter ratio to produce a normal stress. Thus many ordinary liquids, such as water and various petroleum oils, are not likely to produce significant normal stresses because their molecules tend to be round rather than elongated shapes or long linear chains.

Also, comparing the silicones with polymer solutions, shows that the use of a solution is necessary in order to obtain a high normal stress while still retaining a reasonable shear viscosity. Polymers are often used to confer desirable properties to an oil; i.e., decrease the change in viscosity versus temperature, and produce a less viscous oil under high shear. Thus it is a natural step to consider using a polymer additive to produce normal stresses in a liquid. The SAE 10W30 oil tested showed that oils now on the market do possess sufficient quantities of polymer additives to produce normal stresses that may be able to play a minor role at high shear rates. In fact, Okrent (5) showed that polymer additives do reduce wear in a dynamically loaded journal bearing.

Possibility of High Normal Stresses at High Shear Rates

However, linear extrapolation of the stress values given in Table I to 10^6 sec.$^{-1}$ shear rate does not generally predict tensile stresses high

enough to be dominant or even equal in magnitude to conventional viscosity-produced load bearing. But, examination of a simple shear viscosity curve versus shear rate implies that perhaps an extrapolation greater than linear may exist to make for higher normal stresses.

The reasoning is as follows. When a stress/shear rate curve (or viscosity versus shear rate curve), is taken from low to exceedingly high shear stresses, two definite breaks in the curve are often to be noted. This is shown schematically in Figure 3 to emphasize the discontinuities. The viscosity tends to remain essentially constant at low shear ranges, decreasing through a range of moderate shear rates, ultimately becoming a constant at high shear rates. The first two regions have been found in linear polymers when the polymer comprises the entire oil as in a silicone oil, or when the polymer is dissolved in a base liquid (6, 17). The third region is rarely reached in a viscometer because of the high shear rates or shear stresses required.

One expects that there should be a correspondence between normal stresses and viscosity; both must be evidence of the same intermolecular action comprising "viscosity". As evidence, the discontinuity in slope for a normal stress (for the limited data available) was found to occur near the same shear rate as the discontinuity in slope for viscosity. Hence one would look in the high shear ranges for a second discontinuity in a normal stress curve corresponding to the second discontinuity in the viscosity curve.

To extend the argument further, one should be able to relate a normal stress to the viscosity of a given fluid. Since the shear viscosity is essentially constant for both the low shear region and the high shear regions, might not the normal stress at high shear be related to the shear rate by a large exponent as at low shear rates, i.e., $n > 1$. This would then produce higher normal stresses than predicted by a linear extrapolation of $n = 1$. This was the argument, admittedly speculative, that directed obtaining a shear rate high enough to enter the region beyond the second discontinuity of a given fluid to look for high normal stresses.

Normal Stress Differences

For a viscoelastic fluid under steady state conditions there are three stress quantities of significance, the normal stress differences $(p_{22} - p_{33})$ and $(p_{11} - p_{22})$, and the shear stress τ_{12}. The axes along which these stresses act are shown according to convention in Figure 4 for various geometries. The normal stresses are usually considered as the stress deviatrics, i.e., they are stresses superposed on the hydrostatic pressure of the fluid. The hydrostatic pressure is uniform in all directions at any given point in a fluid, whereas the normal stresses are not. The tensile stress τ_{11} is $-p_{11}$; the compressive stress is p_{22}.

Discussions of the normal stress differences, their significance, and how they are obtained from rheological measurements are given by White and Metzner (7), Lodge (8), Frederickson (9). Coleman and Noll (10, 8) have considered a wide class of rheological flow equations for isotropic materials to show that steady state shear flow should result in a general form that can be reduced to:

$$\begin{cases} p_{11} - p_{22} = k_1 s^2 \\ p_{22} - p_{33} = k_2 s^2 \\ \tau_{12} = \tau_{21} = k_3 s \end{cases} \qquad (5)$$

The shear rate is s; k_1, k_2, k_3 are constants.

Roberts (11), from experimental evidence in a Weissenberg rheogonometer on various fluids concluded that $p_{22} - p_{33} = 0$. This is customarily called the Weissenberg hypothesis. This was previously implied when it was stated for equation (2) that the head for load bearing (p_{33}) was the same as the head for centripetal pumping (p_{22}). However, Markovitz et al (12), examining test data on polyisobutylene solutions, concluded that $p_{22} - p_{33}$ is not necessarily zero. The reasons for this are still unresolved. One can suggest as a possible cause the superposition of different effects as previously outlined in the discussion of a rotating geometry. Variations then might or might not appear in different equipment depending upon the equipment size, specific geometry, and the relation between the tensile stress, compressive stress and shear rate.

Viscometry

It is necessary to use tube viscometers at high shear rates because they are less susceptible to heating errors than a rotational viscometer. A tube is a "once-through" device; the fluid is sheared briefly as it passes through the tube. In a rotating viscometer the fluid is sheared and heated continuously. A bearing should fall somewhere in between the two when oil is circulated and used for cooling.

The flow through a tube has been made use of in various ways to measure normal stresses: (1) Measurement of the swelling of the jet after the fluid has exited from the tube. (2) Measurement of the reaction of the jet. (3) Measurement of the pressure on the wall of the tube. The last two methods are suitable for high shear rates; the first is impracticable because friction between the emergent jet and the surrounding air shatters the jet before swelling can take place to be measured. The use of tubes in this way has been questioned since it is customarily assumed that the hydrostatic pressure in the jet is at atmospheric pressure as it exits from the tube.

Metzner et al (2, 3, 7) have done considerable work with jet viscometers. However, they presume that $(p_{11} - p_{22})$, rather than τ_{11}, is measured by a jet viscometer when using either the swelling of a jet or its reaction. Nevertheless, their derivations consider only the tensile stresses τ_{11} acting along the lines of shear at the exit to the tube. In effect, p_{22} is considered to be zero. They found that the tensile stress obtained from swelling measurements coincided fairly well with the stress obtained from reaction measurements. This shows that p_{22} does not aid in the swelling of a jet, thus confirming the basic applicability of their derivations.

Broadly speaking, the rheological equivalence of a rotating device and flow through a tube has not been proved. But, as has been pointed out by Metzner (14), experimental data indicates at least an approximate numerical equivalence between the two techniques. That is, Shertzer and Metzner (2), in experiments using a rotational viscometer with polyisobutylene dissolved in decalin, found that τ_{11} plotted versus the shear rate on log-log coordinates extended towards, and had the same slope as data for τ_{11} measured with a jet viscometer.

Molecular Action

We are concerned with elongated molecules, or clusters of molecules that rotate and deform in a shear field. Various aspects of this are summarized and dealt with by Philippoff (19). For large, non-deformable microscopic particles carried by a liquid, the average speed of rotation of a particle has been shown to be proportional to the shear rate (15). The instantaneous rate of rotation varies, and is highest when the length of the particle is normal to the plane of shear, and lowest when parallel. Thus, on the average, there is an alignment of particles parallel to the stream lines in the plane of shear. A similar situation should exist with elongated molecules. Thus, it is possible that fluid properties may differ normal to the direction of shear compared to the direction of shear; the fluid may not be isotropic. Derivations to date for normal stresses are based on the assumption that the fluid is isotropic.

In addition, there is evidence to suggest that the fluid molecules interact as a consequence of rotation. That is, the area swept out by a molecule increases as the shear rate increases. Then, molecules increasingly contact, perhaps entangle, and so exhibit a high frictional drag to produce high tensile stresses. Also, because the molecules for the greater percentage of the time are parallel to one another, they must occupy a minimum volume when in the parallel position. Hence, as molecules rotate out of a parallel position, the molecules may then be "constricted" and interact to deform one another normal to the lines of flow, thus inducing a compressive stress normal to the plane of shear.

The point in mentioning these possibilities is to conclude that an experimental approach is necessary to evaluate and sum such actions to give the flow properties of the fluid. The characteristics of the molecules are not well enough known to deduce a priori the resultant flow effects, assuming one had a clear picture of the actions at hand to postulate a practical physical or mathematical model.

III. Jet Viscometer

The general approach followed is that of Shertzer and Metzner (2) for measuring the reaction of a jet, modified to achieve high shear rates. The viscosity of the fluid tested versus shear rate is found from pressure/flow measurements made with various capillaries. The flow is obtained by collecting the efflux from the capillary. From this, velocity profiles at the exit of the capillary are obtained. This enables the force required to accelerate the fluid to fully developed flow to be computed. The measured reaction of the jet less the acceleration force, gives the tensile forces acting over the cross section of the jet as the fluid leaves the capillary. By suitable algebraic manipulations, the maximum tensile stress at the wall of the capillary is obtained versus the maximum shear rate at the wall. The derivations for this are outlined in the Appendix for rectangular capillaries; Shertzer (20) has given derivations for round capillaries.

The compressive stress was obtained by measuring the pressure on the wall at four different points and extrapolating a plotting of these pressures out to the exit of the capillary.

Description of Viscometer

A schematic of the jet viscometer is given in Figure 5. A high pressure hydraulic system capable of reaching 10,000 psi provides a constant pressure source to force fluid through a capillary. The hydraulic system is separated from the fluid being tested by an isolating piston. This permits the use of a conventional hydraulic power supply consisting of pump, accumulator and hydraulic oil storage reservoir. The fluid to be tested is stored in a pressure vessel. A tube feeds the fluid to the capillary. To achieve flexibility and thereby also be free from Bourdon tube effects, the tube is long, with several offsets, using the highest possible stress limit to obtain a thin wall. The capillary is mounted on a spring-mounted reaction table. The springs are stiff so that the maximum full scale deflection is about 0.005"; thus resulting in a negligible reaction from the feed tube when moved. The spring constant is approximately 1 gram per 80 microinch deflection. Deflection is measured by a sensitive differential transformer with a useable resolution of 2 microinches. Linearity of the springing and the readout system was achieved over 4 decades of thrust measurement. However, an air conditioned room with a regulated temperature was essential to obtain repeatability to within the 2 microinch accuracy. The reaction table is mounted on a massive cast iron table placed on vibration isolators in order to be free of floor-transmitted vibration.

Capillaries

A small capillary is necessary in order to reach high shear rates for various reasons. A primary limitation making for a small capillary is the need to keep the pressure within reasonable limits. This can be seen from equations (6) and (7). Too high a pressure increases the viscosity and density of the fluid excessively, and distorts the capillaries. The maximum pressure that one can tolerate in a jet viscometer is about 10,000 psi. This set the maximum pressure for the hydraulic power supply. Much of this pressure is used to overcome entrance losses and initially accelerate the fluid; it does not remain to markedly affect viscosity relations for the major length of the tube.

The price paid for using a small capillaries is the need to have an extremely sensitive measurement of the jet reaction; i.e., the reaction force decreases as the cross sectional area of the jet decreases. Also, small capillaries have an annoying tendency to clog from the polymer no matter how well the base fluid is filtered.

Capillaries were made in sizes from 0.0465" to 0.0005" as shown in

Figure 6, where they are plotted versus the range of shear over which useful data was obtained. Each capillary is approximately twice the size of the next smaller capillary. The larger capillaries are round; the smaller capillaries are rectangular. The size of the round capillaries was checked by flow tests with water. The rectangular capillaries were disassembled; dimensions were measured directly.

Rectangular capillaries were chosen for extremely high shear rates; in effect this made the viscometer practical. The rectangular shape in particular enabled a measurable reaction to be obtained for a small capillary height by using a large width of capillary to make for a large cross-sectional area. Also, rectangular capillaries do not clog as readily as circular capillaries. Ease of manufacture was obtained by making the rectangular capillaries from two lapped blocks clamped together, with a cut-out shim inserted between the lapped faces to form the capillary proper. The thickness of the shim formed the height of the capillary; the cut-out, width.

The reason a small capillary must be used to obtain a low pressure drop can be seen from the relations for the flow of a Newtonian fluid in a circular or rectangular tube; the kinetic energy term increases as the square of the size for a given shear rate, whereas the viscous term is proportional to the size:

Circular:

$$\Delta p = 2 S_m \left[\frac{L}{R}\right] + \frac{S_m^2 R^2}{16} \left[\frac{\rho}{2g}\right] \quad 3.44 \qquad (6)$$

Rectangular:

$$\Delta p = \mu S_m \left[\frac{L}{H}\right] + \frac{S_m^2 H^2}{9} \left[\frac{\rho}{2g}\right] \quad 2.70 \qquad (7)$$

Where:

Δp = Pressure drop across tube

S_m = Maximum shear rate at wall

R = Radius of tube

ρ/g = Mass density

μ = Viscosity

The cross-section of the rectangular tube is 2H x W, where 2H W.

A small diameter capillary is also necessary in order to keep the Reynolds number low (R_e<2000) to ensure laminar flow. The Reynolds number for Newtonian flow given in terms of the shear rate is:

$$R_e = \frac{S_m R^2}{2} \frac{\rho}{\mu} \qquad (8)$$

A small Reynolds number also makes it possible to keep the entrance length down to a fraction of the length of the capillary. This is necessary for accurate results, since the pressure drop incurred over the entrance length per unit length of capillary is higher than for the fully developed Poiseuille flow. The entrance length for Newtonian flow is:

$$X = \frac{1}{100} S_m R^3 \frac{\rho}{\mu} \qquad (9)$$

Capillaries with a length to diameter ratio of 200:1 were standardized on. a series of capillaries of different length to diameter ratios was tested. A length of 150:1 showed the beginnings of a reduced tensile strength; further reductions occurred with shorter lengths. In other words, a length of at least 150:1 is necessary to achieve consistent velocity profile data.

IV. Test Results

Tensile Stress

Tensile stresses of 1,000 psi and more were obtained in the 10^6 to 10^7 shear range with a 3.2% solution of polyisobutylene in kerosene. The molecular weight of the polyisobutylene was 2 x 10^5. The tensile stress obtained is plotted versus shear rate in Figure 7 for three decades of shear from 10^4 to 10^7 sec.$^{-1}$. The tensile stress given is the maximum stress τ_W occurring at the wall of the capillary. The shear rate given is the maximum shear S_m occuring at the wall, assuming the flow to be Newtonian. This slightly displaces values along the abscissa from their true position. Below 10^5 sec.$^{-1}$, the exponent n relating stress to shear is one; above 10^5 it ranges from 3 to 1.4 depending on the capillary size. However, the tensile stress decreases rapidly as the capillary size decreased. Tensile stress is plotted versus capillary size in Figure 8 to show this. For example, the tensile strength in the 0.001" capillary is 41 psi at a shear rate of 2.0 x 10^6. Contrast this with the highest value obtained in any of the tests; 25,000 psi for the 0.022" capillary at the same shear rate. Also, the tensile stress obtained in a rectangular capillary was about 1.5:1 lower than for a round capillary having the same minimum channel size, for a given maximum shear rate.

Compressive Stress

Sakiades (13), in a relatively large circular tube (3/8" diameter, length/diameter = 75), using a different fluid, and with flow at low shear rates, showed essentially a constant compressive stress along the tube equal to 0.7 times the tensile stress. As previously mentioned, his approach was to measure wall pressures and plot these pressures versus their axial position along the tube. A straight line was drawn through these test points. Extrapolated to the discharge end of the capillary, the wall pressure was not zero as would have occurred for an ordinary fluid. Because of the linear plot, one concludes that the compressive stress is constant versus tube position. Hence the compressive stress is not produced by the transient acceleration of the fluid as it enters the capillary, but by the steady-state flow taking place thereafter between the parallel walls.

The same experimental procedure was followed as used by Sakiades, except that high shear rates were used along with a rectangular cross-section flow channel. The capillary size was 0.005" high by 0.5" wide in cross-section, and 1" long. Four small pressure taps were provided in the wall.

The maximum values of the compressive stress obtained in each test were found to range from 0.7 to 0.3 of the tensile stress. The highest value obtained was 400 psi at a shear rate of 8×10^6. The test results are shown in Figure 9. The maximum compressive stress as well as the average of the four pressure taps is plotted versus shear rate. Below a shear rate of about 1.5×10^6, the maximum compressive stress is 0.7 of the tensile stress, showing agreement with Sakiades results. Above this shear rate there is a falling off to about 0.3 of the tensile stress. The sharp break in the curve from 2 to 3 occurred when a new batch of fluid was used.

The compressive stress was not constant as had been the case for Sakiades, but decreased downstream within the capillary. The average value given is the average of four measuring points. With the number of tests allocated to this task, it could not be satisfactorily established whether this decrease -- which at times was appreciable -- occurred because of fluid breakdown, unusual flow conditions due to the use of viscoelastic fluid, or because of the inherent nature of the stress. Because Sakiades did not obtain a decrease, one presumes that fluid breakdown was probably the cause.

Viscosity

Auxiliary data obtained relating to the normal stress measurements were fluid density, viscosity versus shear rate, and viscosity versus pressure. The viscosity versus shear rate data was obtained by using the selfsame capillaries that had been used to measure the reaction of the jet. As shown in Figure 10, the results from various differently-sized capillaries are in good agreement. In obtaining the viscosity, the pressure drop caused by kinetic losses was corrected for by assuming a parabolic (Newtonian) velocity profile, as in equations (6) and (7). This smoothed out data adequately, showing that a more accurate correction based on the actual non-Newtonian velocity profiles was not needed.

Heating effects started to reduce viscosity at shear rates above 10^5, but had less of an effect as the capillary size decreased. That is, the change in slope denoting heating started at a higher shear rate as the capillary size decreased. The heating appeared to have little effect on the tensile stresses. The average increase in temperature of the fluid after passing through a capillary at the maximum shear rate was about 20°C. At extremely high shear rates, an abrupt change in slope occurred denoting the onset of fluid breakdown. At this point, the tensile stress also decreased abruptly.

The viscosity of the fluid versus pressure was obtained by timing the fall of a ball in a glass tube that had been inserted in the pressurized fluid. The fall was timed by noting a change in capacitance as the ball passed through rings that had been mounted on the outside of the tube. The data obtained is given in Figure 11. Corrections for the increase in viscosity due to pressure were not found necessary.

Life/Breakdown Tests

Life tests at very high shear rates produced rapid breakdown. The test procedure used was to repeatedly cycle fluid through a capillary, and then use viscosity as a measure of breakdown. The reciprocal of the relative viscosity if plotted versus the number of cycles in Figure 12 for a few shear rates.

V. Commentary

Tensile and compressive stresses higher than any heretofore measured have been obtained in a visco-elastic fluid. A high shear region wherein the stresses are related to the shear rate by an exponent n greater than one has been found. As postulated, a "discontinuity" in the viscosity and tensil stress curves was found to occur at about the same shear rate (10^5 sec.$^{-1}$) to denote the beginning of this region. However, at these shear rates fluid life was exceedingly short, measured by but a few times circulated through a short tube. If longer life were wanted, a compromise would have to be sought between the magnitude of the normal stress desired and the life of the fluid. A lower molecular weight polymer would undergo less breakdown, but produce lower normal stresses. A strong molecule not subject to breakdown is needed.

Confirmation of Existence of Tensile Stresses Up to 10^5 sec.$^{-1}$

An effort was made to check the tensile stress by other means. It was possible experimentally to show the equivalence between the jet viscometer and a rotating geometry up to a shear rate of 10^5 sec.$^{-1}$. Orsino (21) used a parallel face seal geometry in which the clearance between the faces could be adjusted. The adjacent faces between which fluid was placed had an outer and an inner diameter; the mean diameter was 1.25". Fluid was introduced at the outer radius, and the fluid pressure generated was measured at the inner radius, where a tube connection led to a pressure transducer. Negligible flow took place since the pressure tap was dead ended. A typical pattern established itself. At low clearances between the faces (0.002", 0.001", 0.0005"), and at a shear rate of 10^4 sec.$^{-1}$ and less, the pressure obtained checked closely with the pressure predicted from the tensile stress (obtained from the jet viscometer) acting via a hoop stress. However, as the shear rate increased above 10^4 sec.$^{-1}$, the pressure produced when using smaller clearances increasingly diverged to go below the predicted pressure. A clearance of 0.004" reached a 20 psi hoop stress at 10^5 sec.$^{-1}$ before diverging, to give the highest tensile stress obtained.

This then showed that the tensile stress could be the same in both geometries, and correspond over a range of shear rates. However, small clearances reduced the tensile stress, this reduction increasing with increasing shear rate. It was presumed that fluid breakdown prevented reaching higher tensile stresses at higher shear rates.

Lack of Tensile Stress Immediately at Wall

It is interesting to note that the tensile strength obtained in the jet viscometer was seen by the fluid, but not by the wall of the capillary surrounding the fluid. Suppose one assumed that the tensile stress τ_W did act on the wall. Taking a force balance between the pressure Δp causing flow

and a resisting stress τ_w at the wall gives for a circular capillary:

$$\Delta p = 2 \tau_w \frac{L}{R} \tag{11}$$

Taking a typical test where the tensile stress was 12.6 psi (i.e., within the range substatiated with a rotating viscometer), one obtains a Δp of 10,850 psi. The actual pressure causing flow was 163 psi. Therefore one must conclude that the tensile stress ceases to exist close to the wall, and that only a small fraction of the tensile stress probably shows up as viscosity in the viscosity measurements using the capillary.

Clearance Sizes

Certain generalizations appear possible for the high shear ranges of 10^5 to 10^7 sec.$^{-1}$. The approximately constant ratio between the compressive stress p_{22} and the tensile stress τ_{11}; for both our tests and Sakiadis tests, for different fluids at widely different shear rates; implies that these stresses are related. It is possible of course that this may be chance; a wide field of data would be necessary to check this.

If a constant ratio of p_{22}/τ_{11} is assumed, then the reduction in tensile stress in a small capillary or clearance must also result in a reduction in the compressive stress. In fact, the rectangular capillary used to measure the compressive stress did have a reduced tensile stress due to its small size. This then says that high tensile or compressive stresses are not to be expected between extremely close fitted machine parts. Referring to Figure 8, clearances of at least a few thousands of an inch are seen to be needed to obtain substantial stresses. Most high speed bearings have clearances of this order. This means that high speed journal bearings can probably have compressive stresses of the same order as viscous-produced load bearing to aid lubrication. Stresses an order of magnitude higher, say 1,000 to 10,000 psi, to dominate over viscous load apparently are not attainable in the normal clearance one would expect in a bearing. It also follows that a "contacting" mechanical face seal, wherein the clearance between faces may be 0.0001" or less, will not develop useable normal stresses; however, a controlled clearance seal or a thrust bearing possessing a larger clearance should benefit.

The effects obtained substantiate the picture of rotating molecules as a necessary part of the phenomenon of normal stresses. One puts forth that long molecules cannot rotate in an exceedingly small passageway or near a wall.

Hence rotary motion must be necessary for the molecule to sweep out a large enough area to contact other molecules; otherwise drag or entanglement between molecules cannot result to produce a tensile stress.

Interestingly, it was found possible to filter the polyisobutylene solution through a 40 micron filter, but not through a 10 micron filter. This implied that the effective dimensions of the molecules -- or the largest discrete particles present in the solution presuming agglomeration may have taken place -- fall somewheres between the two filter sizes, i.e., between 0.0016" and 0.0004". One also sees that the effect of capillary size to reduce the stress probably ceases somewheres between 0.02" and 0.03"; i.e., the trend of the curves in Figure 8 is to become asymptotic to lines parallel to the abscissa. This then leaves one with the rough guide that the passage width must be somewhere between 10:1 to 75:1 larger than the effective molecule or particle size to permit free rotation. Thus, if one wanted normal stresses in small passages at high shear rates, molecules possessing a correspondingly lower molecular weight would have to be used.

Distance to Develop Normal Stresses

Again, invoking the probability of a constant ratio of p_{22}/τ_{11}; the first pressure tap in a capillary (when measuring the compressive stress) showed a fully developed compressive stress. This implies that the distance required to elastically stress the molecules to obtain normal stresses for the shear rates employed is no longer than 0.125", the distance between the entrance to the capillary and the pressure tap. This confirms that the length of capillaries chosen was reasonable to achieve fully developed normal stresses. Similarly, normal stresses should not take an excessive distance to develop in a bearing.

Rectangular versus Round Capillary

The tensile stress in a rectangular capillary was approximately 1.5:1 lower than for a round capillary. In a practical sense this shows that the results from a round capillary are roughly comparable to a rectangular capillary, and hence applicable to shear between parallel walls.

The normal stress differences that occur in a viscoelastic fluid are $(p_{11} - p_{22})$, and $(p_{22} - p_{33})$. When using a circular capillary it is conceivable that $(p_{22} - p_{33})$ may not be negligible, and if so may then possibly affect the flow. The use of a rectangular capillary makes $(p_{22} - p_{33}) = 0$ because of the symmetry of the rectangular cross section. Thus, the difference in tensile stress shows that a $(p_{22} - p_{33})$ must exist in the circular cross section in this case, and be taken into account for accurate viscometry.

APPENDIX

Reaction of a Jet Emerging from a Rectangular Capillary

Capillary dimensions, terminology:

Length = L, width = W, height = 2H = D, W>> 2H, pressure drop over length L = Δ p; h = distance from centerline of height to a plane moving at velocity u, and upon which a tensile stress τ_{11} acts parallel to the centerline of the jet, $\tau_{11}\big|_w$ = tensile stress at wall.

Shear stress on an element:

$$\tau = \frac{\Delta p\ h}{L} \tag{12}$$

Viscosity relation:

$$\tau = k\left[\frac{du}{dh}\right]^n \tag{13}$$

Total flow Q:

$$Q/W = 2\left(\frac{n}{1+2n}\right)\left(\frac{p}{k\,L}\right)^{\frac{1}{n}} H^{\frac{1}{n}+2} \tag{14}$$

Average velocity V:

$$V = \frac{Q}{2HW} \tag{15}$$

Velocity profile:

$$u = V\left(\frac{1+2n}{1+n}\right)\left[1-\left(\frac{h}{H}\right)^{\frac{1}{n}+1}\right] \tag{16}$$

Maximum shear rate at wall:

$$S_m = \frac{du}{dh}\bigg|_w = \left(\frac{1+2n}{n}\right)\frac{V}{H} = \frac{6V}{D} \quad \text{for } n = 1 \tag{17}$$

Taking a force balance at the capillary exit, and assuming the hydrostatic pressure at the tube exit to be zero, the total thrust T is:

$$T = 2W\int_o^H \rho\, u^2 dh - 2W\int_o^H \tau_{11}\, dh$$

$$= 2WH\left[\rho\, V^2\int_o^1\left(\frac{u}{V}\right)^2 d\,\frac{h}{H} - \int_o^1 \tau_{11}\, d\,\frac{h}{H}\right] \tag{18}$$

From equation (12), the shear stress τ_w at the wall is related to τ:

$$h/H = \tau/\tau_w \tag{19}$$

This leads to an expression for the tensile stress at the wall:

$$\tau_{11}\Big|_w = \rho\ V^2\ I + 2\rho\tau_w\ IV\ \frac{\partial V}{\partial \tau_w} + \rho\tau_w\ V^2 \frac{\partial I}{\partial \tau_w} - \left[\frac{T}{2WH} + \frac{\tau_w}{2WH}\ \frac{\partial T}{2\tau_w}\right]$$

$$\text{where } I = \int_o^1 \left(\frac{u}{V}\right)^2 d\,\frac{h}{H} \qquad (20)$$

In evaluating the above terms, using 6V/D as an approximate value for the maximum shear rate at the wall, one obtains:

$$\frac{\partial V}{\partial \tau_w} = \frac{1}{n'}\ \frac{V}{\tau_w} \qquad \text{where } n' = \frac{\partial \ln \tau_w}{\partial \ln\ (6V/D)} \qquad (21)$$

$$\frac{\partial I}{\partial \tau_w} = 0 \qquad \text{for } n = \text{constant, since the velocity profilt does not change} \qquad (22)$$

$$\tau_w\ \frac{\partial T}{\partial \tau_w} = \frac{T\ \partial \ln\ T}{\tau_w \partial \ln \tau_w} = T\left[\frac{m}{n'}\right] \qquad (23)$$

$$\text{where } m = \frac{\partial \ln\ T}{\partial \ln\ (6V/D)}$$

$$I = 2\left(\frac{1+2n}{2+3n}\right) \text{ using equation (16)} \qquad (24)$$

Substituting (21) through (24) in (20)

$$\tau_{11}\Big|_w = \rho V^2\ \left[2\left(\frac{1+2n}{2+3n}\right)\left(1+\frac{2}{n'}\right)\right] - \frac{T}{2WH}\ \left[1 + \frac{m}{n'}\right] \qquad (25)$$

However n' = approximately $\frac{\partial \ln \Delta p}{\partial \ln Q} = n$

This finally gives for the maximum tensile stress at the wall:

$$\tau_{11}\Big|_w = \rho\ V^2\ \left[2\ \left(\frac{1+2n}{2+3n}\right)\ \left(\frac{2+n}{n}\right)\right] - \frac{T}{2WH}\ \left[1 + \frac{m}{n}\right] \qquad (26)$$

The comparable equation for a circular jet, emerging from a tube of radius R, derived by Shertzer (20) is:

$$\tau_{11}\Big|_w = \rho\ V^2\left[\left(\frac{1+3n}{1+2n}\right)\ \left(\frac{1+n}{n}\right)\right] - \frac{T}{\pi R^2}\ \left[1 + \frac{m}{2n}\right] \qquad (27)$$

The constants n and m are obtained from thrust/shear, and pressure/flow plottings. The above equations presume n and m to be constant, and hence do not apply close to an abrupt change in n or m. Generally speaking, n and m vary somewhat over several decades of shear. This does not appreciably affect the end results obtained with these equations.

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Table 1 Normal stress in linear range (measured at $10^4\ sec^{-1}$)

Fluid	Low shear viscosity centistokes	Approximate L/D of molecule	Measured tensile stress at $10^4\ sec^{-1}$ psi	Extrapolated tensile stress at $10^6\ sec^{-1}$ psi[c]
Kerosene	1.8	1	0.00	0.0
Oleic acid		6.4	0.13	13
Methyl silicones				
	10	13.6	0.00	0.0
	100	27.1	0.19	19
	500	43.1	0.25	25
	1000	53.1	0.29	29
Polymer solutions	Solvent	Molecular weight		
Motor oil, SAE 10W30			0.18	18
Polyisobutylene				
2½ percent	kerosene	2×10^4	2.2	220
5 percent	kerosene	1×10^4	0.70	70
3 percent[a]	decalin		0.87	87
0.5 percent J-100[a]	water		0.69	69
Polyox				
0.01 percent[b]	water		0.03	3
0.05 percent[b]	water		0.12	12
ET597		8×10^6		
30 ppm	water		1.8	18
300 ppm	water		2.1	21
3000 ppm	water		1.8	18

[a] From Shertzer and Metzner, reference [2].
[b] From Oliver, reference [16].
[c] Previous column multiplied by 100, assuming linear relation between tensile stress and shear rate.
Note: The tensile stress values just listed, except where other sources are given, were done by the author with a jet viscometer, measuring the swelling of the emerging jet, following the techniques of Metzner, et al. [3].

LIST OF CAPTIONS

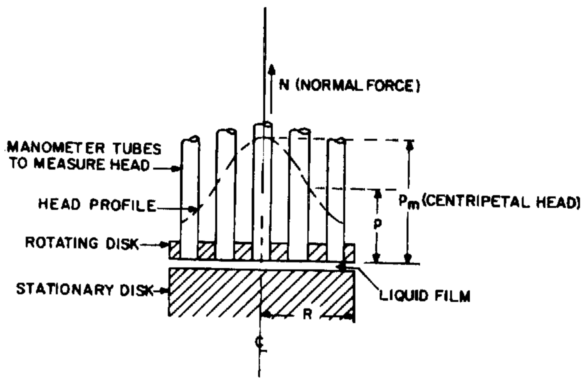

Fig. 1 Weissenberg rheogoniometer

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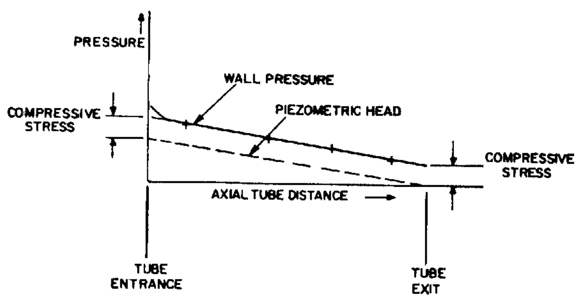

Fig. 2 Compressive stress on tube wall

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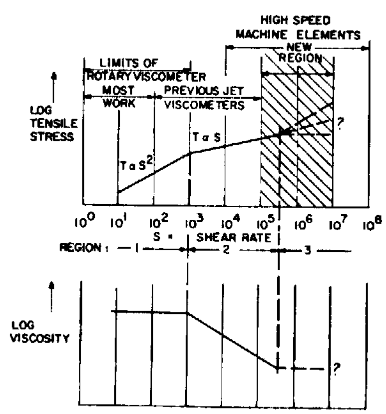

Fig. 3 Status: Normal stress versus shear rate

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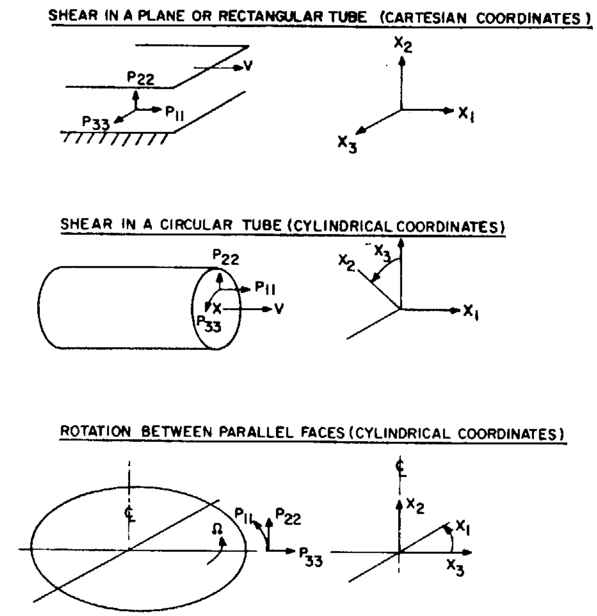

Fig. 4 Normal stresses in different geometries

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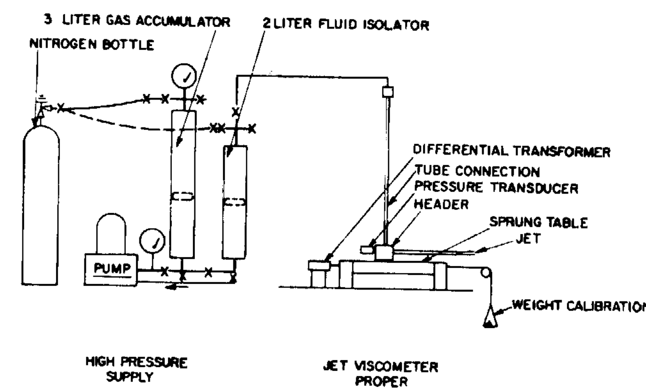

Fig. 5 Jet viscometer

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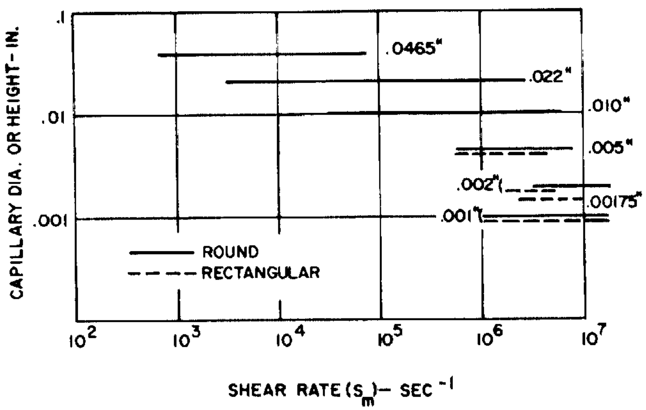

Fig. 6 Capillary sizes and range of test data

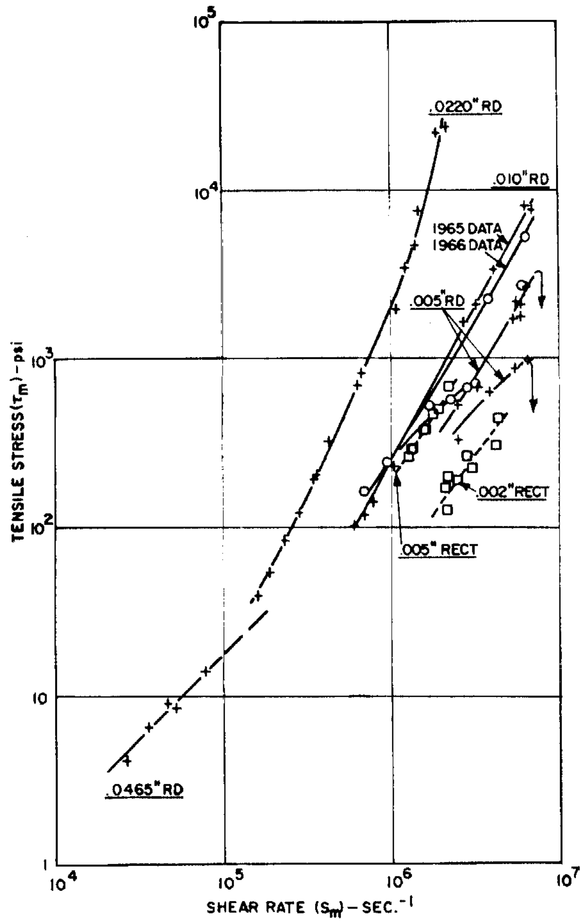

Fig. 7 Tensile stress versus shear

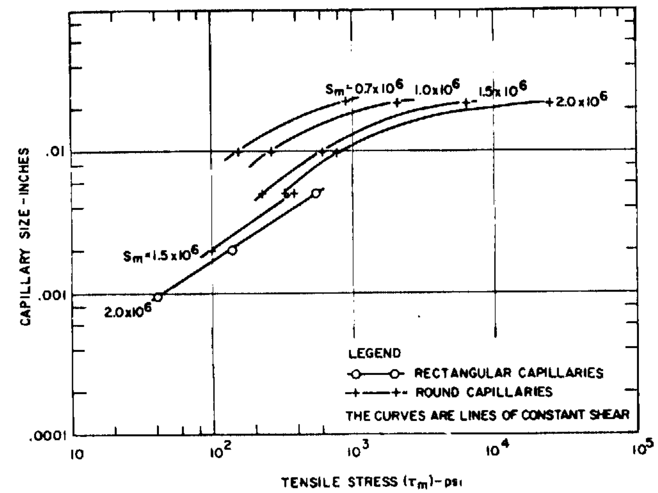

Fig. 8 Tensile stress versus capillary clearance

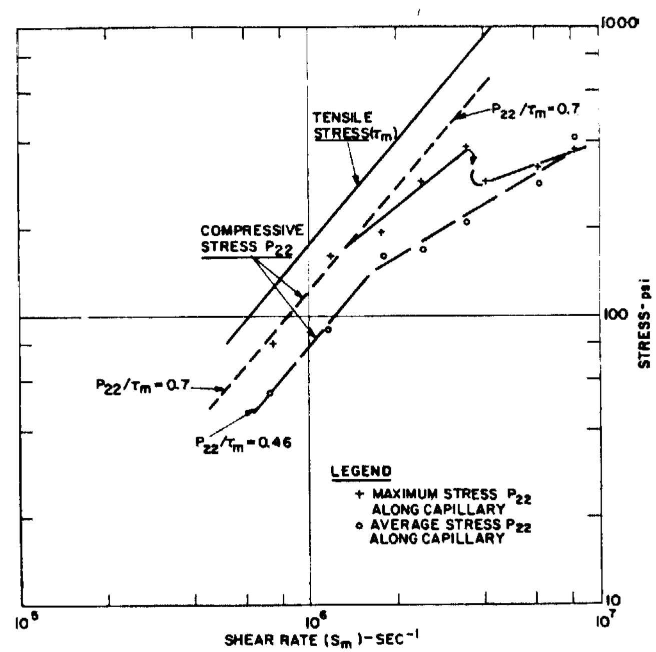

Fig. 9 Compressive stress versus shear rate

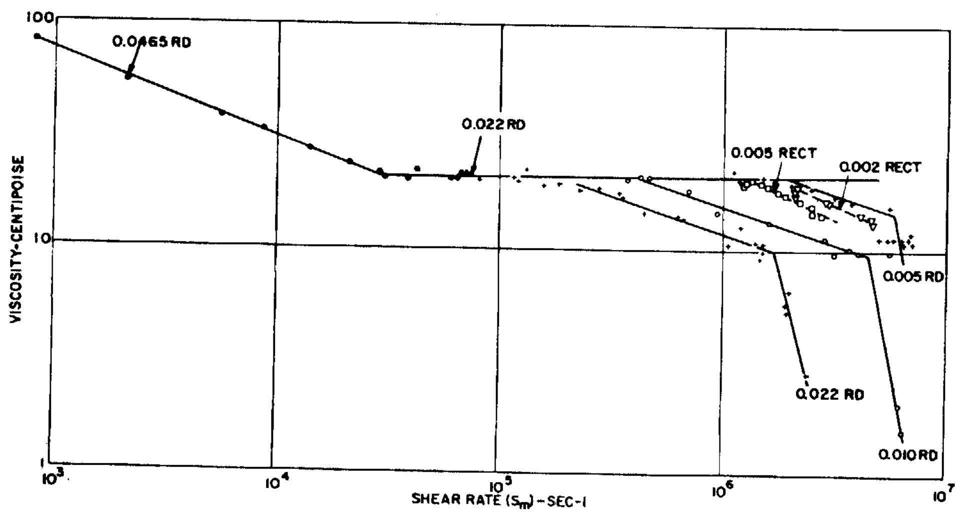

Fig. 10 Viscosity versus shear rate

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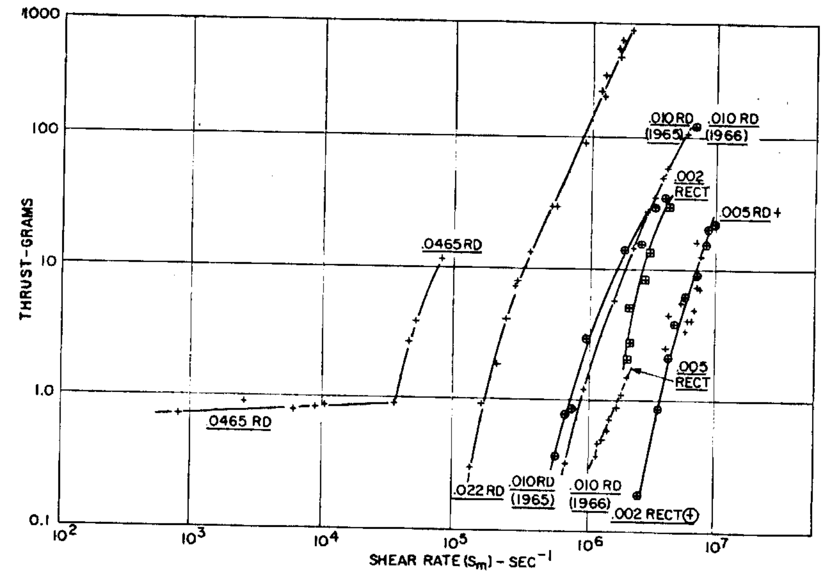

Fig. 11 Thrust of jet versus shear rate

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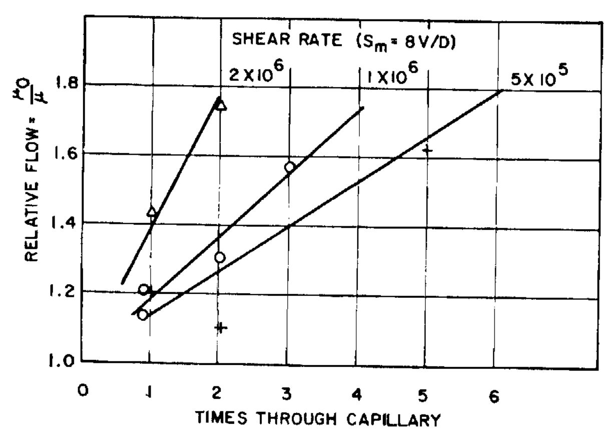

NOTE: .010" DIAMETER CAPILLARY

Fig. 12 Life tests

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NON-NEWTONIAN FACE SEAL EXPERIMENT

A. J. Orsino

I. Introduction

The experimental study by Bernd, Reference 1, predicted increased sealing effectiveness in a face type dynamic seal for fluids containing small amounts of certain polymer additive. As a result of the work reported in Reference 1, an experimental program using a seal like configuration was undertaken in an effort to obtain a more detailed understanding of non-Newtonian fluid behavior in dynamic face seals. The results of this effort are reported in Reference 2 and show only a minimal degree of correlation with the results of the jet capillary experiments by Bernd. The fundamental reasons for the poor correlation is probably due to the significant differences in geometry between the capillary and the dynamic face seal. Transit time for the fluid in the capillary test is extremely short and perhaps of a lower value than the fluid relaxation time. This is not the case for the seal configuration. The comparatively tortuous path for the fluid in the seal, with its correspondingly longer transit time, may give rise to increased possibility for rupture of intramolecular bands. Rupture of fluid molecular bands has a strong influence on fluid properties.

In order to better understand the behavior of non-Newtonian fluids in thin films, and thereby establish a basic understanding of face seals operating with such fluids, a new series of experiments was conducted. The objective of this new testing was the acquisition of experimental data for use in evolving a suitable correlative model. The model would provide a basis for making meaningful predictions regarding face seal performance. Testing during this contract phase was an extension of previous work using the identical test equipment plus some modification to allow for torque measurements. It was expected that torque data would provide an independent means for development of a suitable correlative model. Knowledge of the torque present at the fluid interface allows detailed evaluation of viscosity and its variation with shear rate.

II. Low Shear Rate Viscosity Test

Viscosity data, employing a conventional Cannon and Fenske glass capillary viscometer, was obtained for samples of JP5 containing 3.2% Vistanex polymer additive. The results indicate a viscosity of 4471 centistokes (20,650 Saybolt Seconds) at 77°F. This value of viscosity is approximately 2200 times greater

than the viscosity of the base JP5 fluid. In effect this data provides the zero shear rate viscosity value and agrees quite well with data for the same fluid taken on the Weissenberg Rheogoniometer.

Additional viscosity data was obtained from the rheogoniometer located at the General Electric Research and Development Center. The data for JP5 with 3.2% Vistanex of molecular weight 2 x 10^5 and 10^5 are shown in Figure 1. The data covers only the low shear rate range. Also shown is the normal stress for the Vistanex 200 mixture. This data was obtained in an effort to provide additional information useful in establishing a correlative model.

III. Discussion of Test Data

Tests were conducted at nominal clearances of 0.001, 0.002, 0.003 and 0.004 inches with JP5 plus 3.2% polyisobutylene. The major portion of the testing was conducted with a 1/2 inch wide interface land as used in prior testing. Tests with a 1/4 inch wide land were also completed. Data for the JP5 plus 3.2% polyisobutylene is contained in Tables 1, 2, 3 and 4 for each of the nominal clearances and at several pressures. The data is plotted in Figures 2, 3, 4 and 5 and shows the same trend as previously seen with other viscoelastic fluids. This is characterized by a change in rate of torque increase as speed is increased. In each case the change occurs between 1000 and 1500 RPM. This slope variation is due to the influence of shear rate on viscosity. The specific relationship will be analyzed in greater depth in the next section. The data covers testing to 12000 RPM. In the lower speed range, below 100 RPM, the torque readings contain some scatter due to the low level of interface torque. This difficulty was recognized at an early date and provision was made to obtain low shear rate data from a separate source. The low shear data was obtained on a rheogoniometer available at the General Electric Research and Development Center. The data at the lower shear is shown in Figure 1. This low shear rate data will be used as an additional input to the correlation study. It should be noted that the rheogoniometer is limited to shear rates of 10^3 reciprocal seconds.

As a further comparison Table 5 presents data for plain JP5 at a reservoir pressure of 5 PSIG. Due to the extremely low viscosity it was possible to test at a clearance of 0.001 inches only. In Figure 6 the torque verses speed from Table 5 is plotted. The straight line relationship characteristic of such a fluid is evident.

The data from the tests with the narrow land is shown in Table 6 and Table 7 for the 0.002 inch clearance and the 0.003 inch clearance respectively. The distinct change in slope is seen for each clearance in Figures 7 and 8 for the same clearances.

As an additional source of information for correlations of the viscosity data, several tests were performed with silicone fluids. Silicone SF96 of two different viscosity values (50 and 100) were tested up to 1000 RPM. Fairly good correlation was achieved between measured torque values and torque values computed from flow rates. Test data for the silicone fluids evaluated is shown in Table 8.

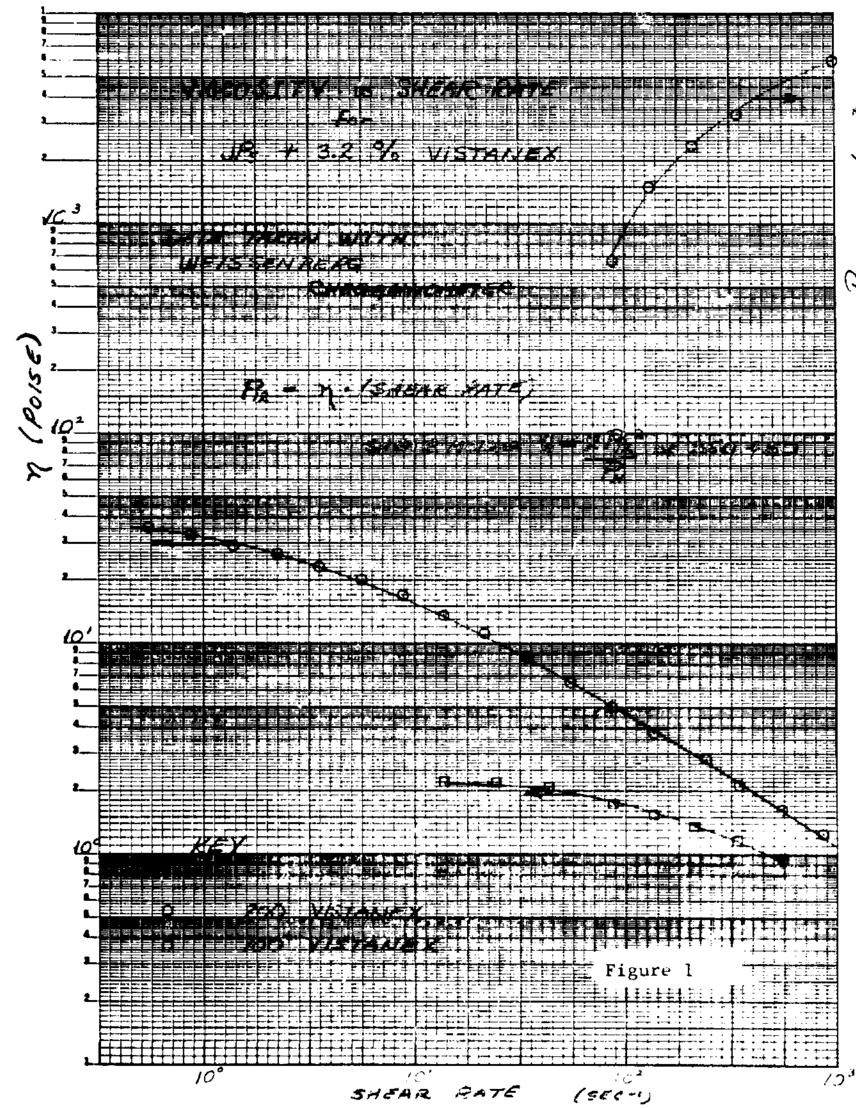


Figure 1

INTERFACE TORQUE VS. SHAFT SPEED

JP5 + 3.2% POLYISOBUTYLENE

NOMINAL CLEARANCE .001 in

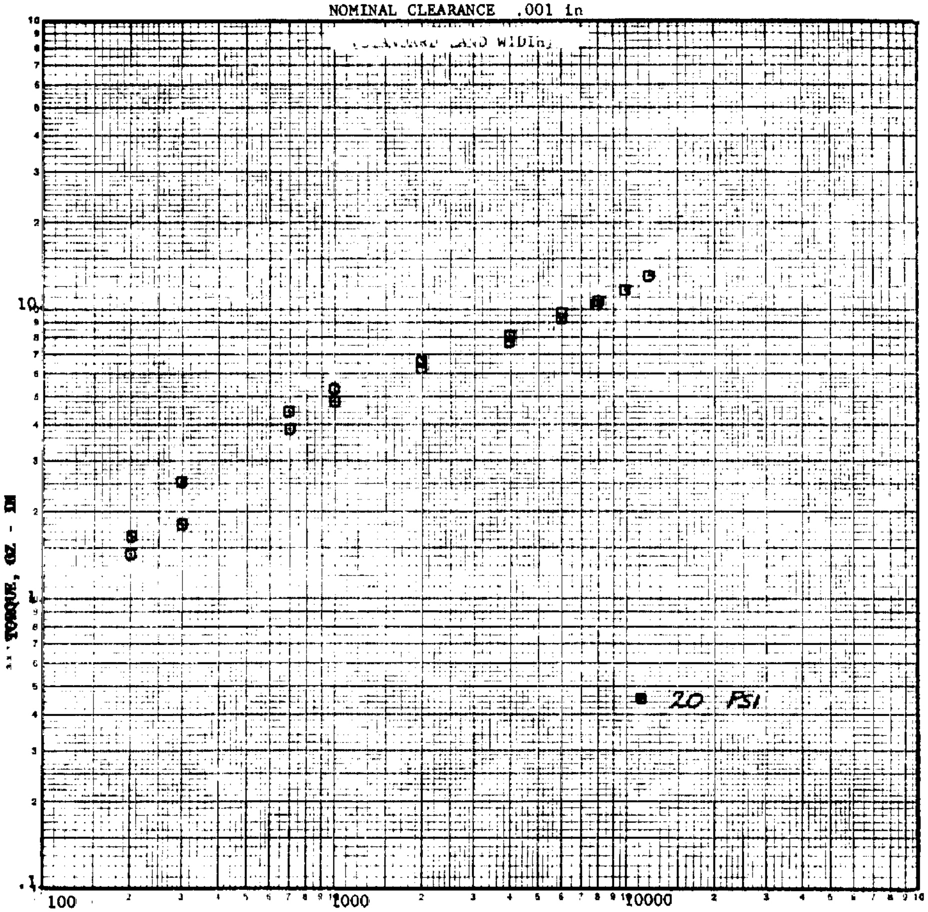


FIGURE 2

INTERFACE TORQUE VS SHAFT SPEED

JP5 + 3.2% POLYISOBUTYLENE

NOMINAL CLEARANCE .002 IN

(STANDARD LAND WIDTH)

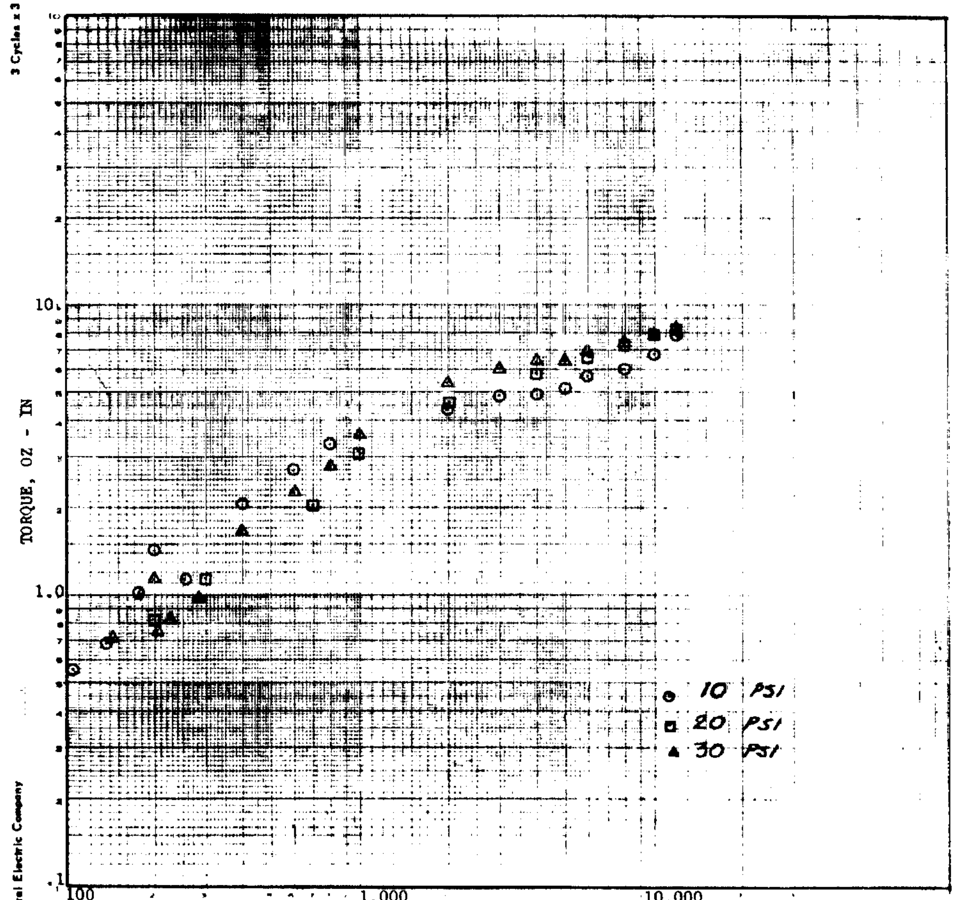


FIGURE 3

INTERFACE TORQUE VS SHAFT SPEED

JP5 + 3.2% POLYISOBUTYLENE

NOMINAL CLEARANCE .003 IN

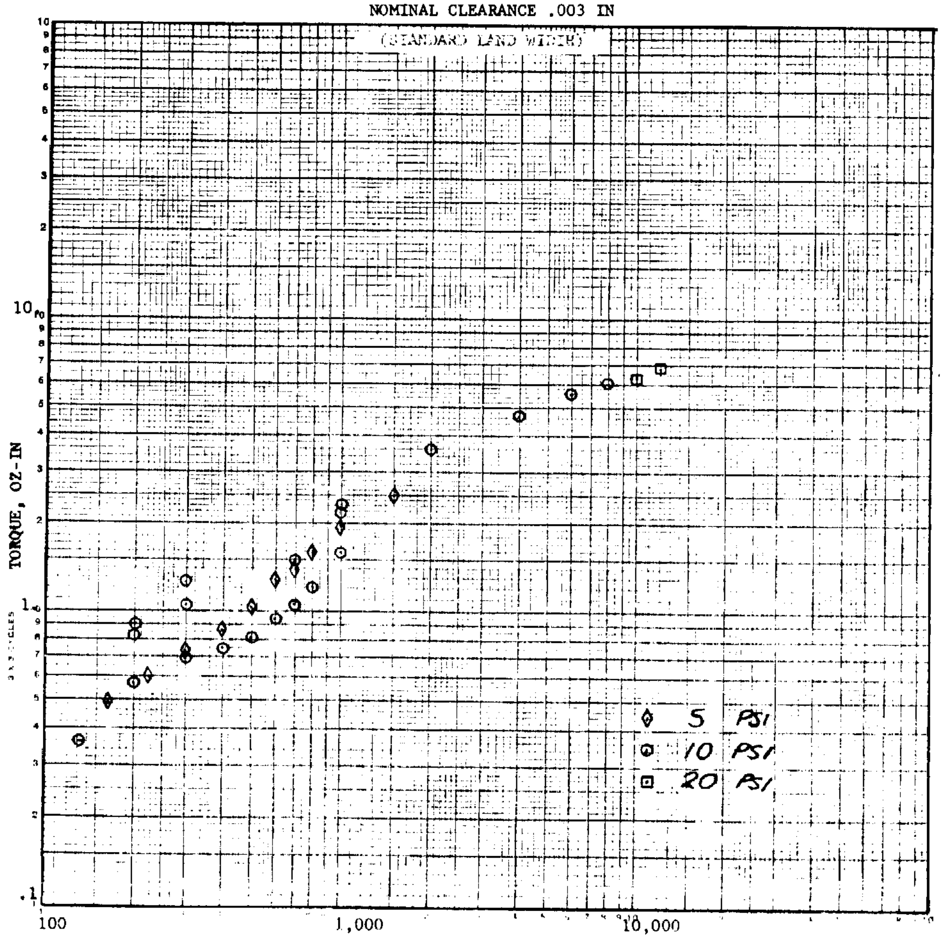


FIGURE 4

INTERFACE TORQUE VS SHAFT SPEED

JP5 + 3.2% POLYISOBUTYLENE

NOMINAL CLEARANCE .004 IN

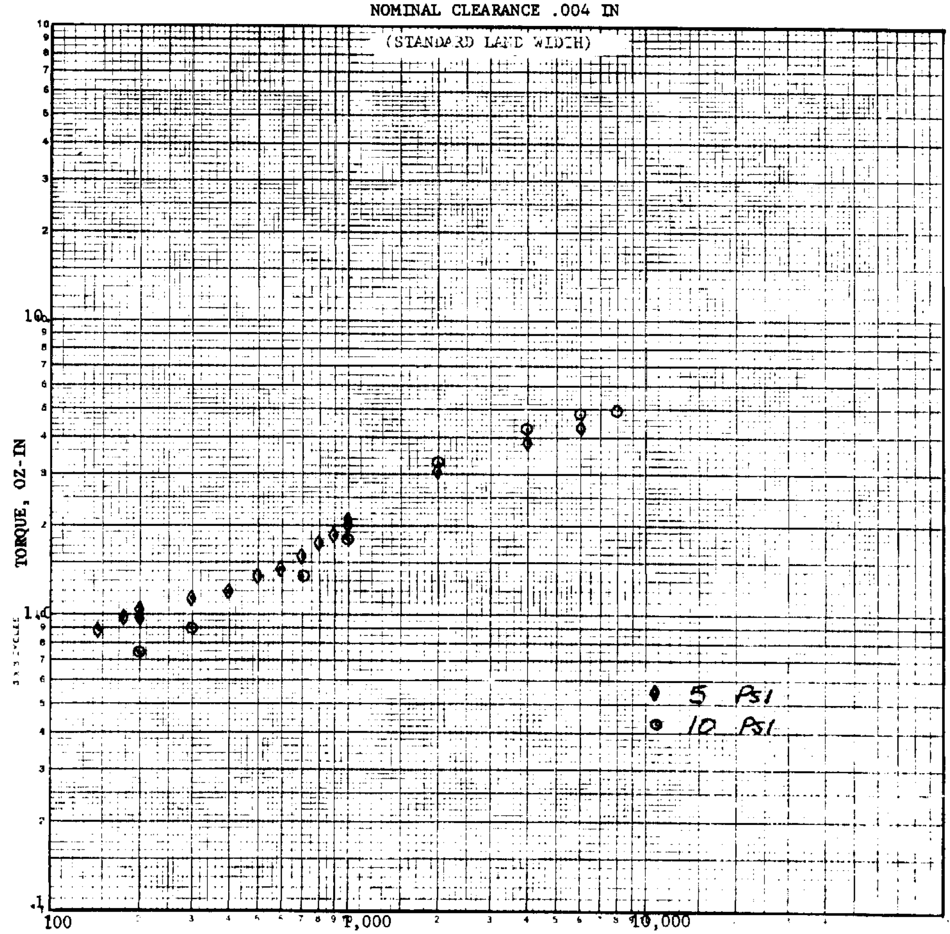


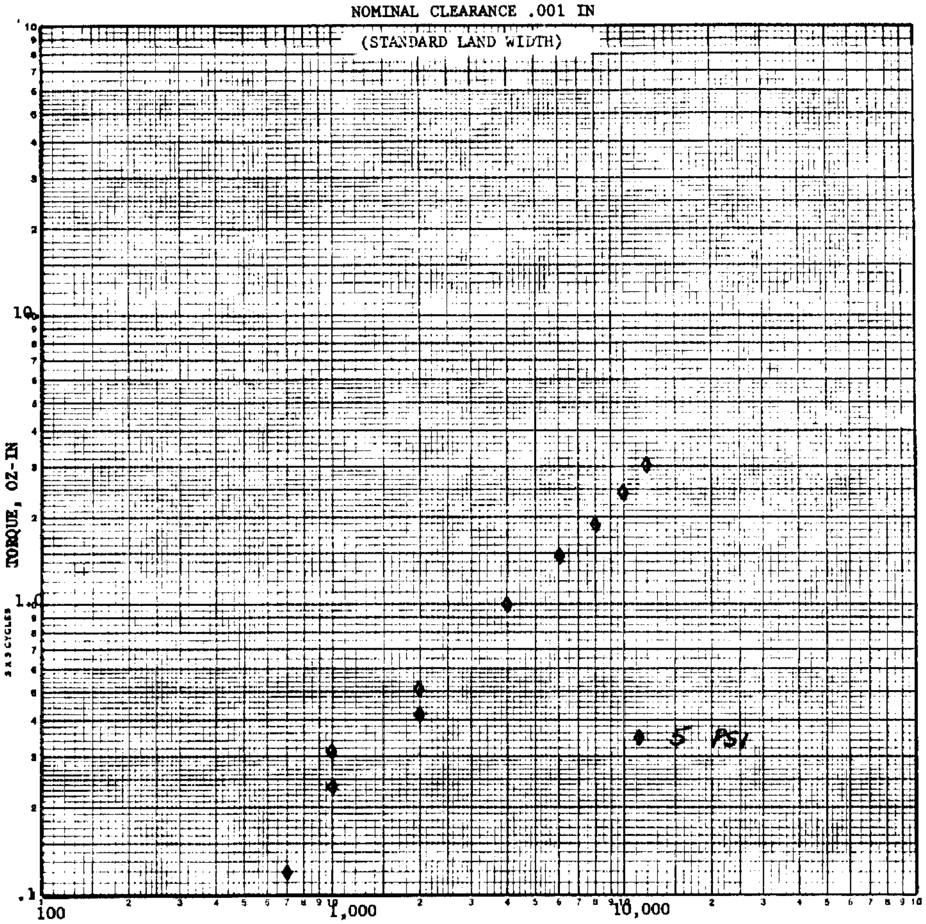
FIGURE 5

INTERFACE TORQUE VS SHAFT SPEED

JP-5

NOMINAL CLEARANCE .001 IN

(STANDARD LAND WIDTH)



SPEED, RPM

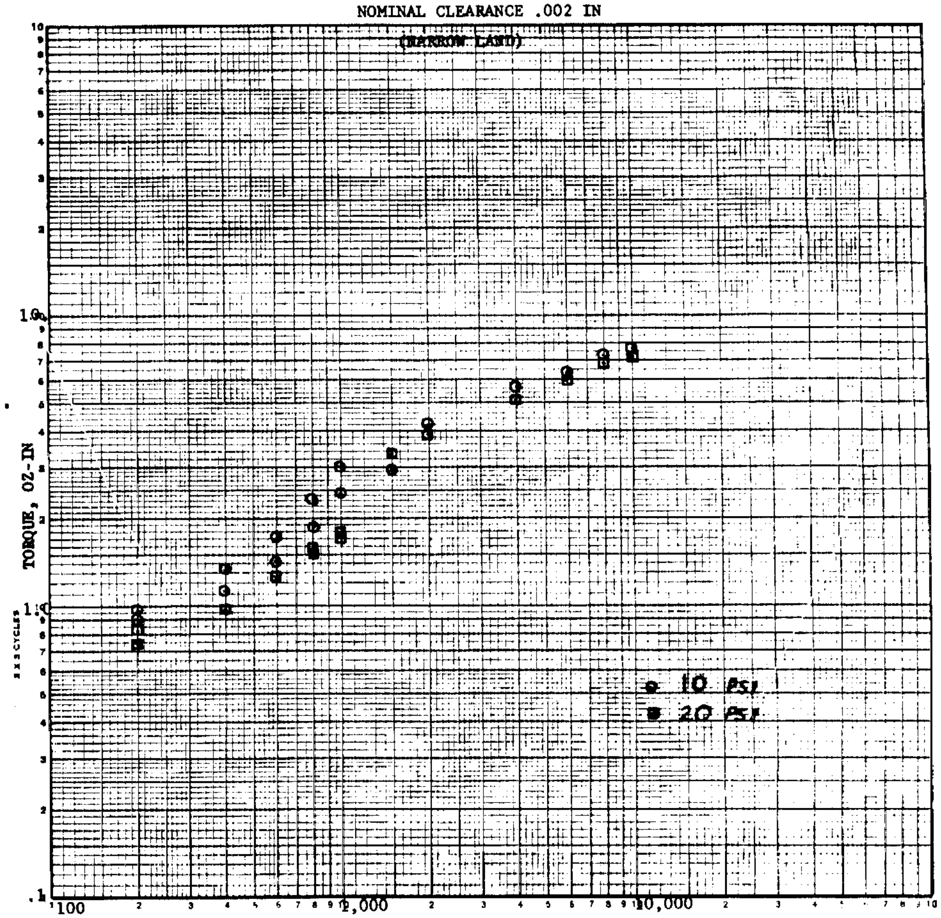
FIGURE 6

INTERFACE TORQUE VS SHAFT SPEED

JP5 + 3.2% POLYISOBUTYLENE

NOMINAL CLEARANCE .002 IN

(NARROW LAND)



SPEED, RPM

FIGURE 7

INTERFACE TORQUE VS SHAFT SPEED

JP5 + 3.2% POLYISOBUTYLENE

NOMINAL CLEARANCE .003 IN

(NARROW LAND)

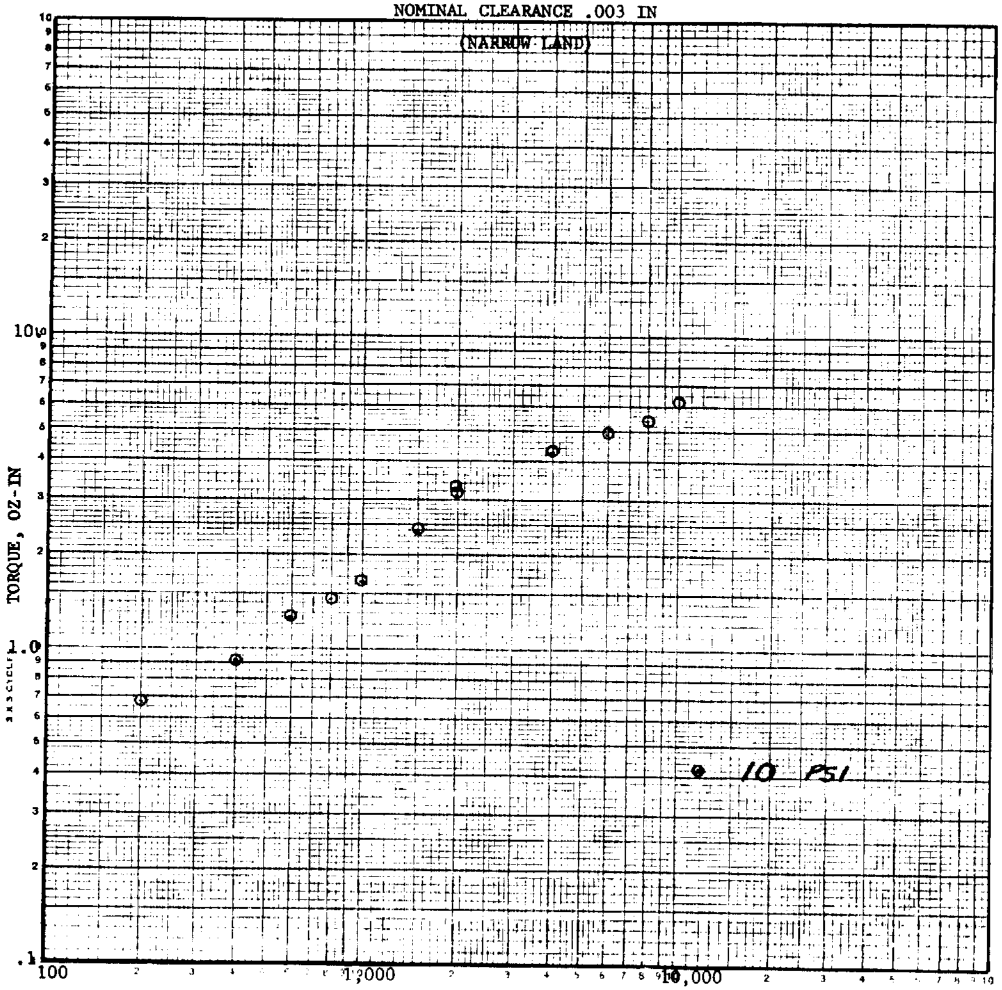


FIGURE 8

TABLE 1

JP5 + 3.2% VISTANEX 200

NOMINAL CLEARANCE 0.001 IN.
(STANDARD LAND WIDTH)

RESERVOIR PRESSURE 20 PSI
AMB. TEMPERATURE 77°F

RPM	Q in^3/sec	h in.	Torque oz-in
INITIAL TEST			
0	.00218	.00114	--
200	.0048	.00114	1.65
300	.00451	.00114	2.55
700	.00188	.00113	4.5
1000	.00204	.00113	4.8
2000	.0019	.00108	6.45
4000	.0025	.00103	8.1
6000	.00277	.00097	9.6
8000	.00371	.00086	10.8
10000	.00389	.00077	11.85
12000	.00399	.00070	13.05
REPEAT TEST			
0	.0015	.00111	--
200	.00383	.00113	1.42
300	.0040	.00112	1.8
700	.00191	.00111	3.83
1000	.00164	.00110	5.34
2000	.00156	.00109	6.83
4000	.00183	.00105	7.8
6000	.00222	.00100	9.16
8000	.00297	.00095	10.65
10000	.00356	.00087	11.86
12000	.00361	.00078	13.07

TABLE 2

JP5 + 3.2% VISTANEX 200

NOMINAL CLEARANCE 0.002 IN
(STANDARD LAND WIDTH)

Reservoir Pressure - 10 PSI
Amb. Temperature 78°F

RPM	Q in^3/sec	h in.	Torque Oz-In
0	.00407	.00216	-
8	.00428	.00206	.225
25	.00500	.00206	.27
36	.00473	.00206	.27
60	.00623	.00206	.375
105	.00718	.00206	.555
137	.0078	.00206	.69
179	.00877	.00206	1.01
200	.00948	.00208	1.42
262	.00865	.00206	1.12
400	.0104	.00208	2.03
600	.0838	.00211	2.7
800	.00509	.00214	3.3
1000	.00314	.00226	3.15
2000	.00169	.00212	4.35
3000	.00305	.00198	4.80
4000	.00337	.00197	4.80
5000	.00198	.00200	5.10
6000	.00300	.00194	5.65
8000	.00359	.00183	6.00
10000	.00363	.00175	6.75
12000	.00451	.00161	7.65

Reservoir Pressure - 20 PSI
Amb. Temperature 77°F

RPM	Q in^3/sec	h in.	Torque Oz-In
0	.0225	.00216	-
200	.0269	.00216	.825
300	.0293	.00215	1.12
700	.0271	.00214	2.03
1000	.132	.00213	3.15
2000	.00501	.00211	4.65
4000	.00446	.00207	5.85
6000	.00594	.00203	6.6
8000	.00687	.00197	7.2
10000	.00800	.00188	7.8
12000	.00983	.00179	8.25

Reservoir Pressure - 30 PSI
Amb. Temperature

RPM	Q in^3/sec	h in.	Torque Oz-In
0	.0539	.00222	--
18	.0444	.00218	.30
35	.0504	.00218	.375
85	.0473	.00218	.69
143	.0533	.00218	.705
202	.0515	.00218	.75
226	.0377	.00218	.84
286	.0584	.00218	.975
200	.0558	.00223	1.12
400	.0633	.00222	1.65
600	.0702	.00222	2.25
800	.0592	.00220	2.78
1000	.0526	.00220	3.53
1000	.0406	.00218	3.45
2000	.0103	.00216	5.4
3000	.0084	.00213	6.0
4000	.00808	.00209	6.45
5000	.00838	.00206	6.6
6000	.00850	.00204	6.9
8000	.0100	.00199	7.2
10000	.0107	.00195	7.8
12000	.0128	.00189	8.25

TABLE 3

JP5 WITH 3.2% VISTANEX 200

NOMINAL CLEARANCE 0.003 IN.**

Reservoir Pressure - 5 PSI
Amb. Temperature 78°F

RPM	Q in^3/sec	h in.	Torque Oz-In
0	.00411	.00303	--
52	.00688	.00303	.405
83	.00801	.00301	.435
166	.0140	.00300	.495
228	.00936	.00298	.60
300	.00920	.00297	.735
400	.00920	.00295	.855
500	.00935	.00294	1.02
600	.00773	.00294	1.29
700	.00676	.00294	1.38
800	.00495	.00293	1.59
1000	.00197	.00293	1.98
1500	.00125	.00293	2.50

Reservoir Pressure - 10 PSI
Amb. Temperature 78°F

RPM	Q in^3/sec	h in.	Torque Oz-In
0	.0163	.00308	--
200	.0272	.00308	..825
300	.0301	.00307	1.05
700	.0332	.00307	1.5
1000	.0217	.00307	2.17
2000	.00371	.00305	3.6
4000	.00252	.00303	4.65
6000	.00421	.00301	5.55
8000	.00395	.00294	6.0
*10000	.0151	.00276	6.3
*12000	.0177	.00271	6.75
0	.0164	.00312	--
200	.0275	.00312	.9
300	.0316	.00312	1.26
1000	.025	.00312	2.32

Reservoir Pressure - 10 PSI
Amb. Temperature 78°F

RPM	Q in^3/sec	h in.	Torque Oz-In
0	.0160	.00307	--
35	.016	.00306	.15
73	.0213	.00306	.27
131	.0232	.00306	.36
200	.0259	.00305	.57
300	.0269	.00303	.69
400	.0303	.00303	.75
500	.0320	.00303	.81
600	.0324	.00303	.93
700	.0287	.00303	1.05
800	.0271	.00302	1.2
1000	.0186	.00302	1.59

* Fluid Pressure - 20 PSI
** Standard Land Width

TABLE 4

JP5 WITH 3.2% VISTANEX 200

NOMINAL CLEARANCE 0.004 IN.*

Reservoir Pressure - 5 PSI
Amb. Temperature 79°F

RPM	Q in^3/sec	h in.	Torque Oz-In
0	.00944		--
37	.0118	.00400	.6
63	.0140		.75
91	.0148		.825
144	.0190		.9
179	.0190		.975
200	.0192		1.05
300	.0200		1.12
400	.0219		1.2
500	.0235		1.35
600	.0225		1.42
700	.0242		1.57
800	.0281		1.72
900	.0172		1.87
1000	.0154		2.02
0	.0102	.00398	--
200	.0203	.00400	.976
300	.0214	.00399	1.12
700	.0210	.00399	1.57
1000	.0148	.00397	2.1
2000	.00225	.00397	3.08
4000	.00218	.00395@	3.82
6000	.00383	.00390	4.27
8000	.00562	.00385	3.45

Reservoir Pressure - 10 PSI
Amb. Temperature 80°F

RPM	Q in^3/sec	h in.	Torque Oz-In
0	.0409	.00399	--
200	.0536	.00399	.75
300	.0583	.00399	.90
700	.0648	.00398	1.35
1000	.0565	.00397	1.8
2000	.0078	.00397	3.3
4000	.00451	.00396	4.28
6000	.0060	.00395	4.8
8000	.00823	.00398	4.95

*Standard Land Width

TABLE 5

JP5

NOMINAL CLEARANCE 0.001 IN
(STANDARD LAND WIDTH)

RESERVOIR PRESSURE - 5 PSI
AMB. TEMPERATURE 80°F

RPM	Q in^3/sec	h in.	Torque Oz-In
0	.0289	.00113	--
200	.0294	.00107	.045
300	.0278	.00107	.0528
700	.0279	.00105	.12
1000	.0266	.00104	.233
2000	.0277	.00104	.51
0	.0259	.00109	--
1000	.0271	.00106	.308
2000	.0232	.00105	.413
4000	.0258	.00102	.99
6000	.0294	.00095	1.47
8000	.0367	.00089	1.89
10000	.0402	.00081	2.4
12000	.0281	.00070	3.00

TABLE 6

JP5 WITH 3.2% VISTANEX 200

NOMINAL CLEARANCE 0.002 IN*

Reservoir Pressure - 20 PSI
Amb. Temperature 75°F

RPM	Q in^3/sec	h in.	Torque Oz-In
0	.0877	.00216	--
200	.0998	.00216	.75
400	.0998	.00216	.975
600	.118	.00216	1.27
800	.0948	.00216	1.5
1000	.0998	.00216	1.72
2000	.0149	.00215	3.98
4000	.0155	.00214	5.18
6000	.0180	.00210	6.00
8000	.0225	.00207	6.9
10000	.0300	.00201	7.2
200	.0748	.00216	..826
400	.0860	.00216	..976
600	.0972	.00216	1.28
800	.0842	.00216	1.57
1000	.0842	.00216	1.80
1500	.0220	.00216	3.38

Reservoir Pressure - 10 PSI
Amb. Temperature 70°F

RPM	Q in^3/sec	h in.	Torque Oz- In
0	.0152	.00199	--
200	.0177	.00199	.975
400	.0214	.00199	1.35
600	.0204	.00199	1.72
800	.0103	.00199	2.32
1000	.00518	.00199	3.00
2000	.00366	.00197	4.2
4000	.00421	.00195	5.77
6000	.00633	.00190	6.3
8000	.00712	.00183	7.05
10000	.0109	.00175	7.65
0	.0136	.00206	--
200	.0211	.00206	..90
400	.0252	.00206	1.12
600	.0223	.00206	1.42
800	.00945	.00206	1.88
1000	.00043	.00206	2.48
1500	.00484	.00206	2.93

*Standard Land Width

TABLE 7

JP5 WITH 3.2% VISTANEX 200

NOMINAL CLEARANCE 0.003 IN
(STANDARD LAND WIDTH)

RESERVOIR PRESSURE - 10 PSI
AMB. TEMPERATURE 75°F

RPM	Q in^3/sec	h in.	Torque Oz-In
0	.0563	.00306	--
200	.0728	.00306	.675
400	.0789	.00306	.90
600	.0823	.00306	1.27
800	.0823	.00306	1.43
1000	.0823	.00306	1.65
2000	.0104	.00303	3.3
4000	.00992	.00303	4.28
6000	.0118	.00298	4.87
8000	--	.00292	5.4
10000	.0203	.00290	6.15
1000	.0842	.00296	1.65
1500	.0315	.00294	2.4
2000	.0115	.00292	3.15

NON-NEWTONIAN CORRELATIVE MODEL

BY

H. J. SNECK

I. INTRODUCTION

Viscometric tests of Newtonian fluids containing viscoelastic additives has shown that it is possible to generate a radial pressure gradient between two rotating disks such that the pressure at the center of the disks is larger than at the perimeter. If such a pressure gradient can also be created in a face seal, this pressure gradient could have a beneficial effect on the outward leakage rate.

Early experiments (Ref. 1) indicated that such a beneficial effect can be obtained. Further tests have been performed, this time including torque measurements, with the view to obtaining if possible a simple correlative model which describes the seal performance characteristics in terms of measurable fluid properties. The results of this attempt are reported on the following pages.

TABLE 8

NOMINAL CLEARANCE 0.002 IN.
(STANDARD LAND WIDTH)

Silicone SF 96 (50)

Reservoir Pressure = 10 psi

rpm	Q in^3/sec	h in	torque oz-in
0	0.0114	0.00209	--
109	0.0119	0.00205	0.36
300	0.0104	0.00205	0.99
700	0.00924	0.00198	1.83
1000	0.0146	0.00195	2.76

Silicone SF 96 (100)

Reservoir Pressure = 40 psi

rpm	Q in^3/sec	h in	torque oz-in
0	0.0297	0.00222	--
200	0.0271	0.00219	1.12
300	0.0260	0.00218	1.87
700	0.0277	0.00218	4.72
1000	0.0257	0.00207	6.82

II. CALIBRATION TESTS USING A NEWTONIAN FLUID

An initial check-out of the test-rig performance was made using several known Newtonian fluids. The fluids used were JP-5 and two silicone fluids, SF96(50) and SF96(100).

Because JP-5 has such a low viscosity it was not feasible to make tests with an applied pressure differential above 5 psi. The viscosity calculated from torque measurements was found to be constant at $1.5(10^{-7})$ reyns over the shear-rate range 10^4 to 10^6 $\sec^{-1}$. This viscosity was then used to compute the leakage rate from the equation

$$Q = \frac{\pi C^3}{6\mu \ln \frac{R_2}{R_1}} \left\{ P_1 - P_2 + \frac{3}{20}\rho (R_2\omega)^2 \left[1 - \left(\frac{R_1}{R_2}\right)^2 \right] \right\}$$

The agreement between measured and predicted leakage-rate was within 10%, which is an acceptable correlation considering the effects which small clearance measurement errors can introduce. (The nominal clearance at 5 psi was 0.001 inches.)

The two silicone fluids were also tested at pressures of 10, 30, and 40 psi using a nominal gap of 0.002 inches. Torque measurements yielded viscosities of 29 centipoise for SF96(50) and 69 centipoise for SF96(100). As with the JP-5 the calculated leakage was somewhat larger than the measured values, the maximum error in this case being of the order of 15% for shear rates ranging from 5×10^3 to 5×10^4 $\sec^{-1}$.

The measured viscosities are 60 to 70% of the manufacturers indicated viscosity and about 80 to 90% of the values required to compute the measured leakage-rates. The Newtonian fluid check-out of the apparatus indicates that reasonably accurate viscosities can be obtained from the torque measurements, and what is more important, these viscosities are constant throughout the shear-rate range and consistant with regard to leakage-rates.

III. ANALYTICAL FORMULATION OF THE PROBLEM

The equations of motion for the fluid are

$$-\rho \frac{v^2}{r} = \frac{\partial}{\partial r}(p - P_{rr}) + \frac{P_{rr} - P_{\theta\theta}}{r} + \frac{\partial \tau_{zr}}{\partial z} \qquad (1)$$

$$0 = -\frac{\partial}{\partial z}(p - P_{zz}) \qquad (2)$$

$$0 = \frac{\partial \tau_{z\theta}}{\partial z} \qquad (3)$$

Where the P_{rr}, $P_{\theta\theta}$, and P_{zz} are the deviatoric components of the stress. The last equation indicates that the shear stress is constant across the clearance. For laminar flow this suggests the approximation

$$v = r\omega\left(\frac{z}{c}\right) \qquad (4)$$

Integrating the first equation across the clearance yields

$$\tau_{zr} = \frac{\partial \bar{P}}{\partial r} z + \int \left(\frac{P_{\theta\theta} - P_{rr}}{r}\right) dz \qquad (5)$$

$$-\frac{\rho}{r}\left(\frac{r\omega}{c}\right)^2 \int_0^z z^2 dz + f(r,\theta) \qquad (6)$$

where $\bar{P} = p - P_{rr}$. Assuming that the viscosity isotropic allows the shear-stresses to be related by

$$\tau_{zr} = \tau_{z\theta}\left(\frac{du}{\partial z}\Big/\frac{\partial v}{\partial z}\right) \qquad (7)$$

where $\tau_{z\theta}$ is independent of z.

Integrating the radial momentum equation once again with respect to z gives

$$\tau_{z\theta} u = \frac{r\omega}{c}\left\{ z f(r,\theta) + \frac{z^2}{2}\frac{\partial \bar{P}}{\partial r} + \int_0^z \int_0^\xi \left(\frac{P_{\theta\theta} - P_{rr}}{r}\right) d\xi dz - \frac{\rho}{r}\left(\frac{r\omega}{c}\right)^2 \frac{z^4}{12} \right\} + g(r,\theta) \qquad (8)$$

If it is assumed that $P_{\theta\theta} - P_{rr}$ is independent of z, then

$$\int_0^z \int_0^\xi \left(\frac{P_{\theta\theta} - P_{rr}}{r}\right) d\xi dz = \frac{z^2}{2}\left(\frac{P_{\theta\theta} - P_{rr}}{r}\right) \qquad (9)$$

Applying the boundary conditions that $u = o$ at $z = o, c$ allows the arbitrary functions $f(r,\theta)$ and $g(r,\theta)$ to be evaluated. As a result

$$\frac{\tau_{z\theta} u}{\left(\frac{r\omega}{c}\right)} = \left[\frac{\partial \bar{P}}{\partial r} + \frac{P_{\theta\theta} - P_{rr}}{r}\right]\left(\frac{z^2 - cz}{z}\right) - \frac{\rho}{r}\left(\frac{r\omega}{c}\right)^2 \left(\frac{z^4 - c^3 z}{12}\right) \qquad (10)$$

The volumetric leakage rate is calculated from

$$Q = \int_{o}^{c} 2\pi u r dr \tag{11}$$

When this flow-rate is evaluated it becomes

$$\frac{Q\tau_{z\theta}}{2\pi r\left(\frac{r\omega}{c}\right)} = -\left[\frac{\partial \bar{p}}{\partial r} + \frac{P_{\theta\theta}-P_{rr}}{r}\right]\frac{c^3}{12} + \frac{1}{12}\left(\frac{3}{10}\right)\frac{\rho}{r}\left(\frac{r\omega}{c}\right)^2 c^5 \tag{12}$$

Integrating this equation with respect to r using the boundary conditions

$$\bar{P} = \bar{P}_1 \ @ \ r = R_1$$
$$\bar{P} = \bar{P}_2 \ @ \ r = R_2 \tag{13}$$

yields

$$\frac{6Q}{c^3\pi}\int_{R_1}^{R_2}\frac{\tau_{z\theta}}{r\left(\frac{r\omega}{c}\right)}dr = \bar{P}_1 - \bar{P}_2 + \frac{3}{20}\rho\omega^2\left(R_2^2 - R_1^2\right) + \int_{R_1}^{R_2}\left(\frac{P_{rr} - P_{\theta\theta}}{r}\right)dr \tag{14}$$

The left hand side of this equation may also be written in the form

$$\frac{6Q}{\pi c^3}\int_{R_1}^{R_2}\frac{\tau_{zr}}{r\left(\frac{\partial v}{\partial z}\right)}dr \tag{15}$$

IV. NO-ROTATION LEAKAGE CORRELATIONS

For the purpose of determining the basic behavior of the fluid the seal may be considered as a slightly divergent rectangular slit. When there is no rotation the slit may be considered analogous to a circular pipe in that the flow is caused only by a pressure differential. Rabinowitsch and Mooney (1, 2) have devised a general expression relating pressure drop, leakage-rate and geometry which is independent of fluid properties. A log-log plot of the parameters in this expression is shown in Figure 1 for the no-rotation test results for 3.2% Vistanex 200 in JP-5. These results are for pressure differences between 5 and 30 psi and clearances of 1, 2, 3 and 4 mils. It may be concluded that the fluid follows a power-law relation from the fact that the results generally lie along a straight line in this plot.

The quantity

$$c\frac{\left(\bar{P}_1-\bar{P}_2\right)}{R_2-R_1}$$

is related to the radial shear stress τ_{zr} for parallel disks by the expression

$$\frac{c\left(\bar{P}_1-\bar{P}_2\right)}{R_2-R_1} = \frac{6Q}{\pi c^2\left(R_2-R_1\right)}\int_{R_1}^{R_2}\frac{\tau_{zr}}{r\left(\frac{\partial u}{\partial r}\right)}dr \tag{16}$$

The data shown in Figure 1 is correlated using the expression

$$\frac{c\left(\bar{P}_1-\bar{P}_2\right)}{R_2-R_1} = 2.86\ (10^{-3})\left(\frac{Q}{c^2}\right)^{0.4} \tag{17}$$

If it is assumed that

$$\tau_{zr}\Big/ r\frac{\partial u}{\partial z} = \frac{K}{r^{0.4}} \tag{18}$$

the last two equations yield the power law expression

$$\tau_{zr} = 1.222\ (10^{-3})\left(\frac{Q}{c^2 r}\right)^{-0.6}\frac{\partial u}{\partial z} \tag{19}$$

where the viscosity can be taken to be

$$\mu = 1.222\ (10^{-3})\left(\frac{Q}{c^2 r}\right)^{-0.6} \quad \text{(reyns)} \tag{20}$$

V. RESULTS OF TORQUE TESTS

The torque test data can be reduced assuming that

$$\tau_{z\theta} = \mu\,\dot{\gamma}_{z\theta} = \mu\frac{rw}{c} \tag{21}$$

in the absence of actual velocity profile data. If μ is constant, then the torque is given by

$$T = \frac{1}{2}\left(\frac{\mu\omega\pi}{c}\right)\left(R_2^4-R_1^4\right) \tag{22}$$

The viscosity calculated from the data using this equation is shown in Figure 2 as a function of

$$\frac{R_2 \omega}{c} .$$

There is considerable scatter in the results up to about

$$\frac{R_2 \omega}{c} \sim 5(10^5)$$

after which there is a good correlation for all applied pressure differences and clearances. In this range it is obvious that the viscosity follows a power-law relationship of the form

$$\mu = \mu_o \left(\frac{r\omega}{c}\right)^n \tag{23}$$

If the torque is now calculated on the basis of the power-law viscosity relation, the equation for the torque becomes

$$T = 2 \left(\frac{\mu_o \omega \pi}{c}\right)\left(\frac{\omega}{c}\right)^n \left[\frac{R_2^{\,n+4} - R_1^{\,n+4}}{n+4}\right] \tag{24}$$

A first estimate of n can be obtained from Figure 2 and then the test data processed through the revised form of the torque equation to determine the constant μ_o. Carrying out this calculation yields the correlation

$$\mu = 2.7(10^{-3}) \left(\frac{r\omega}{c}\right)^{-0.60} \quad \text{(reyns)} \tag{25}$$

for

$$\frac{r\omega}{c} > 5(10^4)$$

VI. INVARIANT POWER LAW CORRELATION

The scatter in the low shear rate viscosity is undoubtedly due to the fact that the radial and tangential shear rates are of the same order of magnitude in this range. At higher rotational speeds, the tangential shear rate

predominates so that the simple correlation given above is quite accurate. The invariant form of the power-law is

$$\tau_{ij} = \mu_o \left| \left(\frac{\partial u}{\partial z}\right)^2 + \left(\frac{\partial v}{\partial z}\right)^2 \right|^{\frac{n-1}{2}} \dot{\gamma}_{ij} \tag{26}$$

where the viscosity is given by

$$\mu = \mu_o \left| \left(\frac{\partial u}{\partial z}\right)^2 + \left(\frac{\partial v}{\partial z}\right)^2 \right|^{\frac{n-1}{2}} \tag{27}$$

From the two previous correlations it is clear that n-1 = -0.6. Since

$$\frac{dv}{dz} \simeq \left(\frac{r\omega}{c}\right)$$

the quantity μ_o is equal to 2.7 (10^{-3}). The quantity Q/c^2r used in equation (20) is not equal to $\partial u/\partial z$ but only a measure of it. Bearing this in mind, the discrepancy in coefficients can be accommodated by writing the viscosity as

$$\mu = 2.7\ (10^{-3}) \left| \left(\frac{r\omega}{c}\right)^2 + \left(\frac{kQ}{c^2 r}\right)^2 \right|^{-0.30} \tag{28}$$

If k is chosen equal to 3.75, then this expression also agrees with the no-rotation correlation for ω = o.

The final expression for the viscosity is then

$$\mu = 2.7\ (10^{-3}) \left| \left(\frac{r\omega}{c}\right)^2 + \left(\frac{3.75\ Q}{c^2 r}\right)^2 \right|^{-0.30} \quad \text{(reyns)} \tag{29}$$

The total torque can now be calculated from the expression

$$T = 2.7\ (10^{-3})\ 2\pi \int_{R_1}^{R_2} \frac{\left(\frac{r\omega}{c}\right) r^2 dr}{\left| \left(\frac{r\omega}{c}\right)^2 + \left(\frac{3.75Q}{rc^2}\right)^2 \right|^{0.3}} \tag{30}$$

The predictions of this equation are in fairly good agreement with the experimental results over the entire operating range of the seal.

VII. NO-LEAKAGE PRESSURE RISE

Tests were also performed with no initially applied internal pressure. It was found that the internal pressure increased with time due to inward pumping of the fluid. This pressure eventually reached an equilibrium value when the net leakage-rate vanished. This effect is obviously a manifestation of the so-called "Weissenberg effect."

Figure 3 shows the result of these tests where $\Delta\bar{P}_{tot}$ is the measured pressure corrected for centrifugal effects. The pressure difference is plotted in the figure against the parameter

$$\frac{R_2\omega}{c}\sqrt{c}$$

because the data indicates a size effect.

It should be recalled that this pressure difference is related to the normal stresses through the expression

$$\bar{P}_1-\bar{P}_2 = -\int_{R_1}^{R_2}\left(\frac{P_{rr}-P_{\theta\theta}}{r}\right)dr \tag{31}$$

When the right-hand-side of this equation is evaluated for a laminar Newtonian flow it is found to be dependent on the shear-rate, as well as the clearance. This normal stress term for Newtonian flow is extremely small and ordinarily neglected. The important fact is that even in such a simple case this term is not just a function of the shear rate.

It should also be noted that it was assumed earlier that $P_{\theta\theta}-P_{rr}$ was independent of the coordinate z, and following that assumption three integrations with respect to z where performed on $P_{\theta\theta}-P_{rr}$. The effect of performing these integrations is to introduce the clearance into the integrated result. The way in which the normal stresses depend on the local shear rate determines how the clearance enters the integrated expression given above.

If it is assumed that $P_{rr}-P_{\theta\theta}$ is proportional to shear stress square as some experimenters suggest, then

$$\int\left(\frac{P_{\theta\theta}-P_{rr}}{r}\right)dr \propto \left(\frac{r\omega}{c}\right)^{0.8} \tag{32}$$

Since

$$\frac{r\omega}{c}$$

is only an approximation for the tangential shear rate, a small error in it could account for the experimentally observed $\sqrt{c}$ factor.

Correlating the result of Figure 3 yields

$$\int_{R_1}^{R_2}\left(\frac{P_{\theta\theta}-P_{rr}}{r}\right)dr = 1.582\ (10^{-3})\ \frac{R_2\omega}{c}\sqrt{c} \tag{33}$$

where the constant absorbs the effects of the radius R_1.

VIII. LEAKAGE AT LOW RPM WITH AN APPLIED PRESSURE DIFFERENTIAL

Having examined the performance of the fluid under the special conditions of no-leakage and no-rotation, we are now in a position to compare these results with the actual performance of the fluid under more general operating conditions.

Since such an excellent correlation for the non-rotating performance was obtained, it seems logical to first investigate the perturbing effects of modest rotational speeds. Because the rotational speeds are small, the centrifugal pumping effect will be negligible and any changes in leakage rate can be attributed to changes in fluid properties.

Of the quantities so far investigated, probably the term containing $P_{rr}-P_{\theta\theta}$ is the one in greatest doubt and least understood from a fundamental point of view. On the other hand the term containing

$$\tau_{z\theta}\Big/\frac{r\omega}{c}$$

which is equal to the viscosity, is by comparison a much better understood quantity and, therefore, on a much firmer foundation. This suggests that the best way to examine the experimental data for general operating conditions is to compute the quantity

$$\bar{P}_1-\bar{P}_2 + \frac{3}{20}\rho\omega^2\left(R_2^2-P_1^2\right) - \frac{6Q}{\pi c^3}\int_{R_1}^{R_2}\frac{\mu}{r}\,dr \tag{34}$$

which is equal to the apparent value of

$$\int_{R_1}^{R_2} \frac{P_{\theta\theta}-P_{rr}}{r}\, dr \qquad (35)$$

and compare the result with the values of the latter quantity obtained for the special case of Q = o. In the first expression above (33), μ is given by the invariant power law derived earlier [equation (28)].

The results of such computations clearly indicate that when $Q/rc^2 > r\omega/c$ the first expression (33) is very nearly equal to zero (i.e., within probable experimental error), so that the Weissenberg effect or contribution of

$$\int_{R_1}^{R_2} \frac{P_{\theta\theta}-P_{rr}}{r}\, dr$$

is negligible or overwhelmed by the viscous effects. Typically, this occurs at the higher pressure differentials ($\bar{P}_1-\bar{P}_2 \sim 20$ to 30 psi) which, of course, correspond to high leakage rates. Actually, for tests made, $r\omega/c$ was always less than $10^4 sec^{-1}$ when $r\omega/c < Q/rc^2$. According to the no-leakage test results, negligible Weissenberg effects would be expected under these circumstances.

The only disquieting aspect of the calculated results was the fact that the small residual was always negative indicating that μ_o may be a little too large by a factor of about 1.2.

IX. LEAKAGE AT HIGH RPM WITH AN APPLIED PRESSURE DIFFERENTIAL

When the data is run through the equation above for high $r\omega/c$ where $r\omega/c$ is about ten times Q/cr^2, the apparent value of

$$\int_{R_1}^{R_2} \left(\frac{P_{\theta\theta}-P_{rr}}{r}\right) dr$$

exhibits several interesting features.

The first is the rapidity with which the viscous term becomes insignificant. At total shear rates of the order of $10^5 sec^{-1}$, where the total shear rate is given by

$$\dot{\gamma} = \left[\left(\frac{r\omega}{c}\right)^2 + \left(\frac{3.75Q}{rc^2}\right)^2\right]^{1/2}$$

the pressure and inertial terms entirely overwhelm the viscous term so that the apparent value of

$$\int_{R_1}^{R_2} \left(\frac{P_{\theta\theta}-P_{rr}}{r}\right) dr$$

is positive and practically independent of Q. This condition is almost the same as the no-leakage case studied earlier and warrants comparison with it. Such a comparison reveals that the apparent values are always higher than those measured for the no-leakage case, sometimes almost twice as much for the same shear rate without leakage. The no-leakage form of the equation

$$\int_{R}^{R_2} \left(\frac{P_{\theta\theta}-P_{rr}}{r}\right) dr = 1.582\ (10^{-3})\ \frac{P_2^{\mu}}{c}\ \sqrt{c}$$

is, therefore, a conservative estimate of its value for seal operation when $Q \neq o$.

X. LEAKAGE AT INTERMEDIATE RPM WITH AN APPLIED PRESSURE DIFFERENTIAL

This regime of operation is the most interesting and also the most difficult, since it cannot be compared directly with either of the limiting cases previously examined.

The first important feature of note in this regime is the relative weakness of the viscous effects. While more influential than in the limiting case of extremely high shear rate, the viscous term is still quite small compared to the pressure and inertia effects. The apparent value of the normal stress term amounts to a variation around the applied pressure differential with slight corrections for inertial effects. The result is that in the total shear rate range 2 (10^4) to (10^6) the quantity $P_{\theta\theta}-P_{rr}$ is proportional to

$$\left[\left(\frac{r\omega}{c}\right)^2 + \left(\frac{3.75Q}{rc^2}\right)^2\right]^{0.15}$$

for a given clearance and pressure difference.

In this range the leakage contribution to the total shear-rate is quite small so that to a high degree of accuracy

$$P_{\theta\theta}-P_{rr} \propto \left(\frac{r\omega}{c}\right)^{0.3}$$

which is clearly quite different from the no-leakage results.

Because the pressure term dominates the viscous term in equation 33 the normal stress difference is also proportional to the applied pressure difference P_1-P_2 so that for a given clearance

$$P_{\theta\theta}-P_{rr} \propto (P_1-P_2) \left[\left(\frac{r\omega}{c}\right)^2 + \left(\frac{3.75Q}{rc^2}\right)^2\right]^{0.15}$$

There seems to be no simple expression which will correlate the effects of clearance. As noted, the proportion above holds only for a given nominal clearance. When the clearance is changed the normal stress difference changes, but not in a regular or consistent manner. In fact, there seems to be an optimum clearance (about 2 mils) at which the viscoelastic effects are the strongest. This is at variance with the no-leakage results where there was a definite nonotonic trend with clearance changes.

XI. CONCLUSION

It is clear from these findings, as well as those for very high shear-rates, that the advent of radial leakage has a significant effect on the performance of the viscoelastic fluids on face seals at high shear-rates (of the order 10^4 or greater). Since these effects can only be inferred indirectly, rather than measured directly, it is natural that one would like to assign observed unexplainable peculiarities to poor experimental values of the measured quantities.

In particular, it would be nice if the peculiar behavior at moderate rpm could be laid at the doorstep of poorly measured viscosity. This is not a plausible explanation since it is in this range of shear-rates where the most consistent viscosities were obtained.

Even more surprising is the fact that the best overall correlation (and the most believable) was obtained at low shear-rates where the viscosity correlation is its poorest. At least in this region the normal stress difference is relatively unimportant and, therefore, does not introduce added complications.

Considering all the evidence, it is obvious that the normal stress difference $P_{\theta\theta}-P_{rr}$ is dependent on more than just the tangential shear-rate as might be supposed from the examination of viscometer test data where the net leakage-rate is zero. As pointed out by several investigators, zero net leakage flow fields cannot be the simple flows which are usually pictured, but must contain rather complicated secondary flows involving radial and axial velocity components.

From the results obtained here, it is obvious that radial velocity components play a significant role in the behavior of the fluid. The major clue in this regard is the normal stress difference dependence on the applied pressure difference. This pressure difference is probably unimportant in itself, its importance lies in the fact that it is symptomatic of higher leakage-rates and high radial shear-rates.

The key to understanding high shear performance is in determining how normal stress difference depends on the radial shear rate. The analysis used above avoids a detailed investigation of the velocity field and deals only with the gross features of the seal performance, i.e., leakage, applied pressure difference, etc. Obviously, this is not sufficient when the normal stress difference is of the order of the applied pressure difference. Its dependence on clearance is evidence that more detailed knowledge is required.

To carry out a detailed analysis is by no means a simple task. It requires first a model constitutive equation, and second, a computer program for solving the equations of motion and predicting the overall performance characteristics since these equations will not be simple. This is a formidable task, but unfortunately there seems to be no alternative.

FIGURE 1

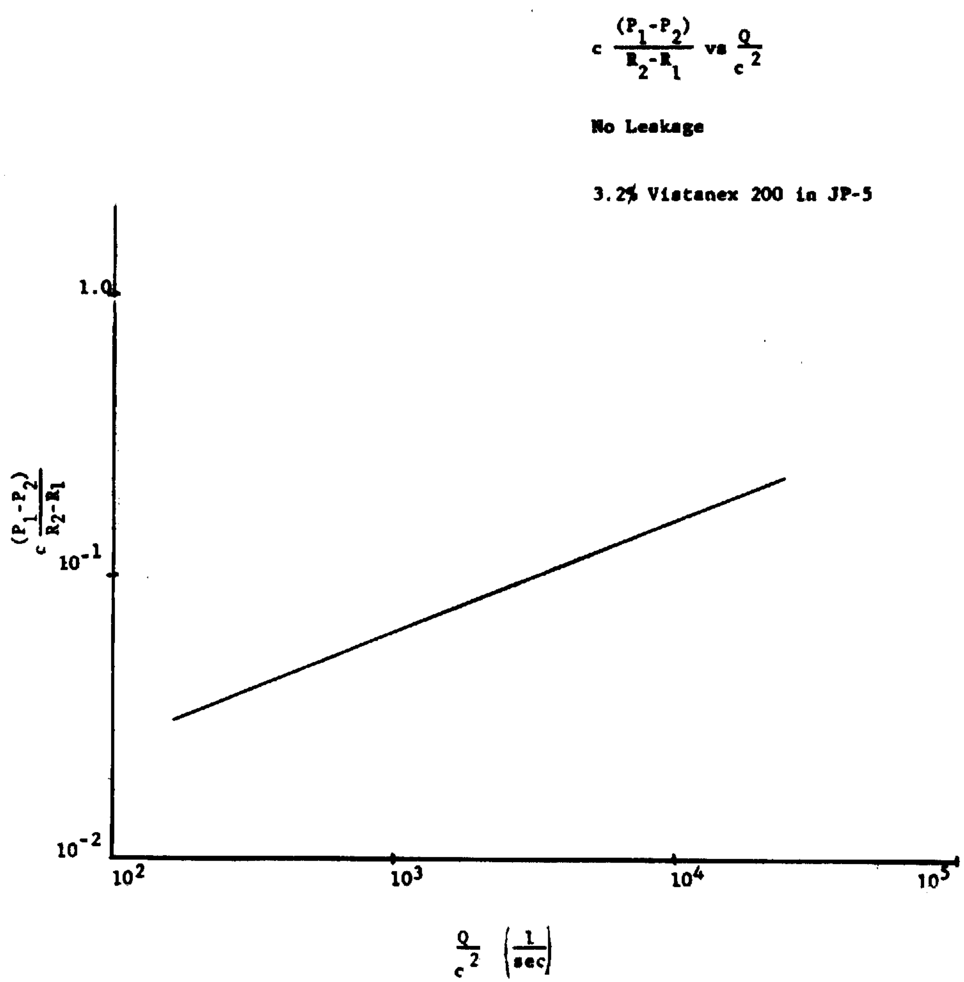
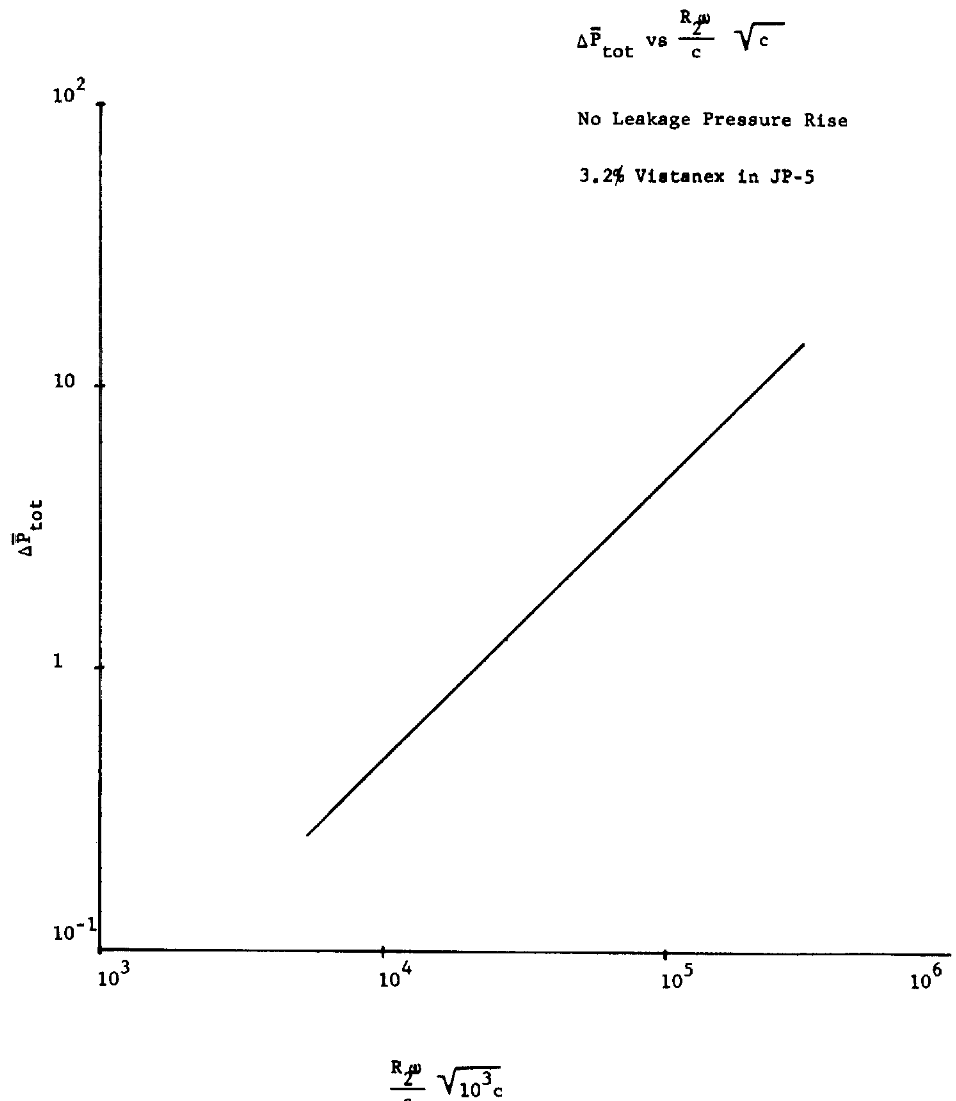


FIGURE 3



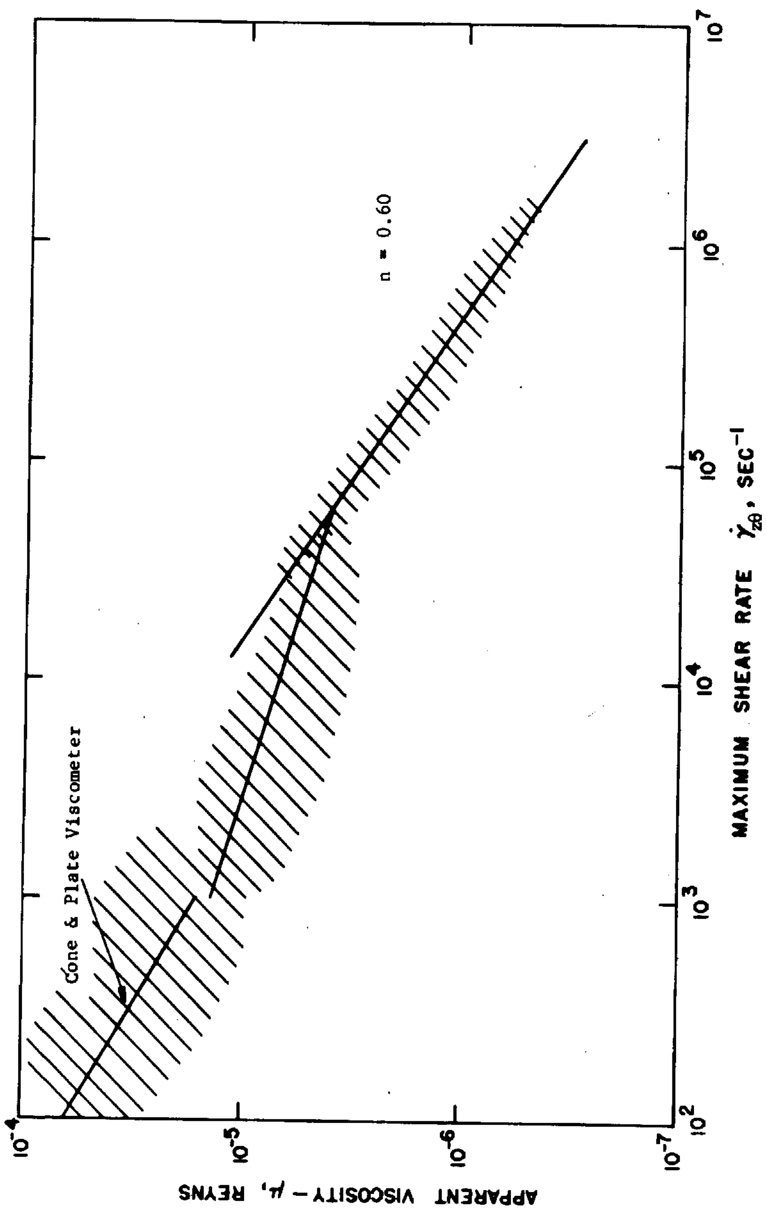


FIGURE 2
SHEAR-STRESS, SHEAR-RATE CORRELATION

NOMENCLATURE

c	= seal clearance
n	= invariant power-law index
p	= isotropic pressure
$\bar{P}$	= $p-P_{rr}$
P_{rr}	= radial component of the deviatoric normal stress
$P_{\theta\theta}$	= tangential component of the deviatoric normal stress
$\Delta\bar{P}_{tot}$	= $\bar{P}_1-\bar{P}_2 + \frac{3}{20}\rho\omega^2 (R_2^2-R_1^2)$
Q	= volumetric leakage-rate
r	= radial coordinate
R_1	= seal inner radius
R_2	= seal outer radius
T	= total seal torque
u	= radial component of the velocity
v	= tangential component of the velocity
z	= axial coordinate
$\dot{\gamma}$	= total shear-rate
$\dot{\gamma}_{z\theta}$	= tangential component of the shear-rate
μ	= viscosity
μ_o	= consistency index
ρ	= density
$\tau_{z\theta}$	= tangential component of shear stress
τ_{zr}	= radial component of shear stress
ω	= angular velocity of seal rotation

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CHAPTER VI

CRYOGENIC SEALING - R. B. Fleming

The sealing of cryogenic fluids has been considered a separate category in this study because in several respects, the problems associated with sealing cryogenic fluids differ greatly from those encountered at room temperatures and above. Furthermore, the sealing of cryogenic fluids is an important category in the general subject of dynamic sealing, as several of the presently-used rocket fuels and oxidizers are cryogenic fluids.

This chapter is divided into the following sections:

1. Introduction
2. Problems Associated With Cryogenic Sealing
3. Present Practice
4. Experimental Work on Dynamic Sealing of Cryogenic Fluids
5. Experimental Work on Friction and Wear
6. Areas in Which Further Knowledge is Needed
7. References

1. Introduction

Oxygen, hydrogen, and fluorine are the cryogenic fluids of greatest current interest as fuels and oxidizers for rocket engines. Also, liquid nitrogen and liquid helium are of interest for providing refrigeration in electronic devices and for other purposes.

Of the various types of dynamic seals, the positive contact seal is generally selected for cryogenic fluids. (See Present Practice below). Therefore, this chapter will be devoted primarily to the positive contact dynamic seal, in which a rotating sealing member slides in positive contact with a stationary seal. This type of seal commonly takes the form of a face seal, in which the sliding surface is perpendicular to the axis of rotation.

2. Problems Associated With Cryogenic Sealing

There are two major problem areas in the sealing of cryogenic liquids. The first is the friction and wear problem, and the second is the problem associated with sealing a liquid near saturation.

a) Friction and Wear

Most seals for use at cryogenic temperatures are of the positive contact type, and thus involve the rubbing of two solid surfaces. The problem of friction and wear results first from the fact that at cryogenic temperatures all common liquid lubricants become solid and therefore unsuitable as lubricants. In addition, many commonly used solid lubricants, such as carbon, are not suitable in certain cryogenic fluids, because these fluids do not possess the necessary lubricating properties. For reasonably low friction and wear, carbon and several other solid seal materials depend on a thin surface layer of dissolved or adsorbed gases or liquids. (References 20, 21, 22). A carbon seal in the presence of water vapor or oxygen operates satisfactorily because this lubricating surface film is replaced as fast as it wears off. However, inert or reducing cryogenic fluids (such as nitrogen or hydrogen) do not possess this filming property and are consequently more difficult to seal with conventional seal materials.

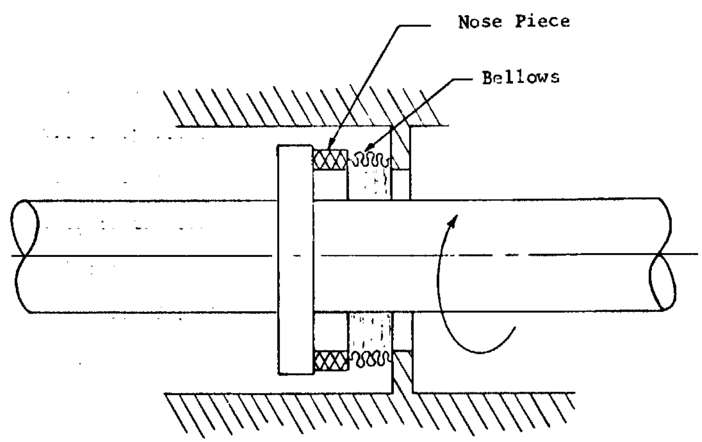
b) Sealing A Liquid Near Saturation

The second major problem involved in cryogenic sealing results from the fact that the fluid being sealed is close to its saturation temperature. Thus, any small addition of heat will either cause the fluid to boil or will produce an unstable liquid in which boiling will take place suddenly upon introduction of a nucleus for bubbles. It is this unstable condition which sometimes results in flashing of liquid to gas between the seal surfaces, which may cause the seal surfaces to separate periodically, with a consequent large increase in leak rates.

If the leak rate is low relative to the frictional heat generated between the seal surfaces, the liquid being sealed will vaporize either before entering the seal or within a short distance after entering the seal. The amount of heat transferred to the seal from warmer parts of the system also influences the evaporation rate of the liquid.

3. Present Practice

The most common type of dynamic seal for cryogenic fluids is the positive contact face seal. In this seal, the usual design is for a shoulder on the rotating shaft to slide against a nose piece of carbon-graphite composition which is held against the shoulder by means of a metal bellows. A sketch of this arrangement is shown on the next page.



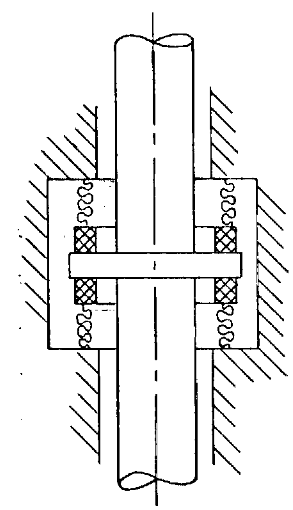
The metal bellows provides a "secondary seal" which maintains alignment of the sealing surfaces, compensates for wear of the seal materials, and provides a spring force to keep the seal surfaces in contact.

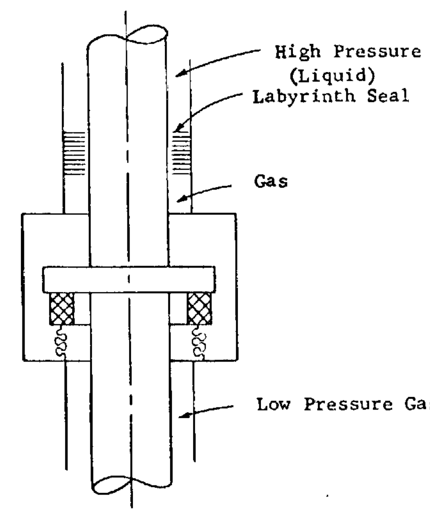
Carbon or graphite are generally considered satisfactory seal materials for oxygen, but they are not satisfactory for nitrogen, hydrogen, or helium. These inert or reducing fluids do not provide the necessary adsorbed film on the carbon surface for friction and wear reduction. For sealing these fluids, therefore, various impregnants must be added to the carbon to provide satisfactory lubricating properties. The impregnant used may vary depending upon the fluid to be sealed. A common impregnant for carbon seals is molybdenum disulfide. The exact compounding of most carbon seal materials is generally considered proprietary information of the carbon manufacturer. In addition to the carbon compounds, various polymers such as Teflon have been used as seal materials.

In addition to improving friction and wear capabilities of the carbon-graphite compositions, impregnants are also employed to reduce porosity of

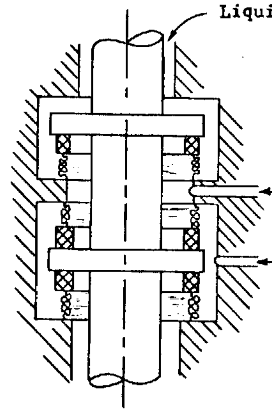
the material so that static losses of fluid are reduced.

Because of the problem, mentioned in the last section, of liquid boiling violently between the seal surfaces, two or more seals are sometimes employed in series. With this arrangement boiling and complete vaporization of the liquid takes place within the first seal, so that the second seal operates in an atmosphere of gas only. The first seal may be a labyrinth type of seal (Section 3G). Two possible arrangements are sketched below.





The sealing of fluorine liquid involves some unique problems. Because fluorine can react with carbon, even at fairly low temperatures, carbon seals are generally not used in fluorine systems, without the use of a bleed gas. One system which has been used to seal fluorine liquid is sketched on the next page.



(See p. 5-159 of Ref. 17)

Oxygen or an inert gas such as helium can be used as the purge or bleed gas. The purge gas is at a pressure higher than the fluorine with the result that leakage occurs from the purge gas into the fluorine system through one of the carbon seals. Therefore, the fluid being sealed is really the purge gas, not the fluorine. A system for sealing fluorine without the use of a purge gas would be desirable, but at present such a system is not known to exist.

4. Experimental Work on Dynamic Sealing of Cryogenic Fluids

Most of the work on dynamic sealing of cryogenic fluids has been done by firms engaged in liquid rocket engine work. In most of the effort these firms have applied, the problem of dynamic sealing has been a secondary problem in the design of pumps and turbines for cryogenic fluids. To our knowledge, no comprehensive work on seals for cryogenic fluids as such has been performed. Also, all investigations carried out thus far have been experimental, with very little analytical effort.

There are two government installations at which significant studies on dynamic sealing of cryogenic fluids are in effect. One of these is the Cryogenic Engineering Laboratory of the National Bureau of Standards, located at Boulder, Colorado. This program has recently commenced under USAF sponsorship. Their apparatus will be used to test bellows-loaded face seals. Two seals will be tested at once, with loading forces in opposite directions to eliminate the need for thrust bearings on the shaft.

Seals loading forces will be small - about 5.5 to 6 pounds. Maximum shaft speed will be 10,000 rpm, though it is felt that speeds of 40,000 to 60,000 rpm will be needed in the future. Liquid nitrogen and liquid hydrogen will be tested with pressures ranging up to 50 psi. It is expected that conventional type seals of various carbon compositions will be evaluated.

The other government installation engaged in this work to a significant extent, is the Lubrication Section at NASA Lewis Research Center, Cleveland, Ohio. This group has constructed two different sets of seal testing apparatus, one for cryogenic fuels and one for cryogenic oxidizers.

The Lewis dynamic seals apparatus for cryogenic oxidizers was designed for both liquid oxygen and liquid fluorine. Included in the device is a disposal system for waste fluorine gas. The designed maximum speed is 60,000 rpm. Provision has been made for measuring leak rates, torque, and wear.

The dynamic seals apparatus for cryogenic fuels at Lewis is designed for liquid nitrogen and liquid hydrogen.

The Lubrication Section of the Lewis Research Center has produced a considerable amount of experimental data on friction and wear characteristics of materials in cryogenic atmospheres. While not relating directly to sealing problems, their work on friction and wear gives significant support to the study of the problem of sealing. References to their work, and the work of others, are given in the next section.

We are not presently aware of any significant effort having been applied to friction and wear in fluorine or to the dynamic sealing of liquid fluorine. It is known (Ref. 17) that fluorine reacts with carbon, even at fairly low temperatures. This may preclude the use of conventional carbon seals for sealing liquid fluorine (without the use of a bleed gas). It is known also, that all metals react to some extent with fluorine, and in fact, the metals which are customarily used in fluorine service form an adherent film of the metal fluoride which prevents further chemical reaction between the fluorine and metal (Ref. 17 and 18). It is not known whether or not this fluorine film possesses lubricating properties. The dynamic sealing of fluorine is considered a very difficult problem (Ref. 18).

5. Experimental Work on Friction and Wear

It was pointed out in the second section of this Appendix that friction and wear are major problems in the dynamic sealing of cryogenic fluids. For this reason, data which has been obtained in cryogenic friction and wear testing is expected to be pertinent to the problem of dynamic sealing.

In addition to cryogenic friction and wear tests, tests obtained in a high vacuum are likely to be useful. The reason for this is that in a very high vacuum, adsorbed surface layers of contaminants are prevented from forming rapidly enough to be effective as lubricants (Ref. 16). Under the usual atmospheric conditions, so-called "dry friction" is not really dry at all, because of the presence of adsorbed films of water and other

contaminants, which may act as lubricants. Ref. 16 points out that even at the low pressures in which most "dry" friction tests have been made (10^{-6} to 1 mm mercury pressure) the residual gases in a system are still sufficient to cover a clean surface with an adsorbed film in a short time.

Because of the absence of lubricating films in most cryogenic fluids, and also under high vacuum conditions, it is possible that the friction and wear characteristics of materials under high vacuum conditions are similar to the same characteristics in say, liquid nitrogen or hydrogen. For materials which do not depend strongly upon adsorbed films for lubrication, it is possible that testing in the atmosphere is sufficient to determine friction and wear characteristics.

A number of materials have been tested in cryogenic friction and wear, and in dry friction and wear experiments. The table beginning on page 118 is an abbreviated list of these materials. The materials are divided into the following categories, which are indicative of the type of dry lubrication.

1. Layer-latticed solid lubricants as surface films, powders, or solid compositions.
2. Other solid lubricants.
3. Surface treatments of materials.
4. Untreated metals and alloys.
5. Polymers.

No attempt is made here to evaluate these various materials as cryogenic sealing materials, because the correlation between friction and wear data and the performance of the material as a cryogenic seal is not known. However, it is expected that a material which performs poorly in a dry friction and wear test is considered a necessary but not sufficient condition for satisfactory performance as a cryogenic contacting seal material. It is felt that many of the materials listed in the table would be good candidates for testing in a cryogenic seals experimental program.

6. Areas In Which Further Knowledge Is Needed

The subject of the positive contact dynamic seal operating in a cryogenic fluid is thus far relatively unexplored, both analytically and experimentally. The analytical approach is extremely complicated by the fact that so many phenomena must be considered. These phenomena include: (1) the fluid mechanics of flow between closely-spaced, moving boundaries, (2) heat transfer, (3) friction, (4) the effect of wear particles, (5) two-phase flow, if boiling is present, (6) the dynamics of the seal system, including the secondary seal (shaft run-out, axial vibrations, etc.), and (7) compatibility of materials (possible chemical or physical reactions with the fluid). These complexities have thus far discouraged significant attempts at analysis of the general problem.

It appears, then, that experimental data must be produced and compiled in order to guide the analytical effort. It would be valuable to know, for example, which of the above phenomena can be ignored in certain cases. At present, this knowledge is lacking because of the scarcity of experimental data. Our evaluation of the state of this art indicates that there has been no comprehensive experimental program on dynamic sealing of cryogenic fluids completed thus far. It is anticipated that some of the programs being conducted by governmental and industrial laboratories will begin producing the needed information in the near future.

In order to improve the techniques of dynamic sealing liquid fluorine, it would be desirable to have knowledge of the friction and wear or the sealing characteristics of various seal materials in a fluorine atmosphere.

SOLID LUBRICANTS FOR DRY FRICTION AND CRYOGENIC FRICTION TESTS

1. Layer-latticed solid lubricants, as surface films, powders, or solid compositions:

Lubricant	References	Testing Conditions, Etc.
MoS_2	1, 2, 6, 13, 15, 19, 23	Vacuum, liq. N_2
$CdCl_2$	2	Normal atmosphere
CdI_2	2, 10	Normal atmosphere, liq. N_2
PbI_2	2	Normal atmosphere
$CoCl_2$	2	Normal atmosphere
$AgSO_4$	2	Normal atmosphere
$CuBr_2$	2	Normal atmosphere
WS_2	2, 7	Normal atmosphere, vacuum
Graphite	2, 4, 8, 13, 19	Vacuum, liq. N_2
BN	5, 7	Normal atmosphere, vacuum
Sintered aluminum oxide	5	Normal atmosphere

2. Other solid lubricants

Lubricant	References	Testing Conditions, Etc.
Zinc Stearate	2	Normal atmosphere
AgI	2	Normal atmosphere
Fe_3O_4	2	Normal atmosphere
Mica	2, 19	Normal atmosphere, vacuum
Talc	2	Normal atmosphere
NiO	2	Normal atmosphere
CdO*	3	High temperature;
Sodium sulfate*	3	High temperature;
Cadmium sulfate*	3	High temperature
Carbon	5, 10, 13, 14, 23	Vacuum, Liq. N_2, Liq. H_2 (with various impregnants)

* Mixture with graphite

Lubricant	References	Testing Conditions, Etc.
CaF_2	6	Vacuum
Diamond	19	Vacuum
PbO	3	High temperature

3. Surface treatments of materials

Material	References	Testing Conditions, Etc.
Stainless steel with N_2, S, & Cr surface treatments	15	Liq. N_2

4. Untreated metals and alloys

Material	References	Testing Conditions, Etc.
Silver	6, 13	Vacuum, Liq. N_2
Tin	6	Vacuum
Gold	1, 6	Vacuum
Lead	6	Vacuum
Various steels	13	Liq. N_2

5. Polymers

Material	References	Testing Conditions, Etc.
PTFE (polytetrafluoroethylene)	5, 9, 10, 12, 13, 15	Normal atmosphere, Liq. N_2, Liq. H_2 (used with various impregnants)
PTFE, glass-fiber filled	9, 13	Liq. N_2, Liq. H_2
PTFCE (polytrifluorochloroethylene)	15	Liq. N_2
Nylon & MoS_2	5	Normal atmosphere
Nylon & Graphite	10	Liq. N_2 & Liq. H_2
Laminated phenolics	13	Liq. N_2
Melamine resin, glass fiber filled	13	Liq. N_2
Resin bonded fabrics	5	Normal atmosphere (with various impregnants)
PTFE in carbon	23	Vacuum

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CHAPTER VII

SEAL MATERIALS STUDY

SECTION 1

SECTION 2

SECTION 1

SEAL MATERIAL STUDY - R. E. Lee, Jr.

Introduction

The primary function of a seal is to prevent or control migration of gases or liquids across a joint or opening in a vessel or assembly. Sealing primarily falls into two classes: Static, where there is no relative motion between joined parts, and Dynamic, where some form of motion occurs between "rigid" members of an assembly. In dynamic sealing there are two predominate types; e.g. positive contact where there is rubbing contact between the rotating and stationary members, and controlled clearance where there is normally no rubbing contact between seal components, and limited but controlled leakage rates are permissible.

Seal Material Considerations

To establish applicable design criteria for rocket seal material selection, a complete awareness of rocket sealing requirements is first necessary. Table 1 presents a general outline of rocket engine seal functional requirements. This illustrates the various environmental and operational requirements which must be met by seal materials in both static and dynamic rocket seal applications. A better understanding of material properties as related to static and dynamic sealing operation is next in order. The predominant seal material properties are shown in Table 2 along with an indication of general magnitude of their relative importance as related to static and dynamic rocket seals.

Unfortunately, there is no single material that can meet the wide range of rocket seal requirements. Realizing this, a compromise between material type, design, and operating parameters is necessary. Table 3 consists of a compilation of some of the more important controllable and uncontrollable variables which will strongly influence the ultimate choice of specific materials for rocket engine seals. The uncontrollable variables are generally fixed requirements and have to be met, whereas the controllable variables are somewhat flexible and can be beneficially used towards optimizing rocket seal performance.

Classes of materials under consideration to meet the stringent requirements imposed by rocket seal operation are shown in Table 4. Considerable future work however is needed to determine the applicability of such materials for rocket seal use.

State of the Art

From the rapid technological advancement associated with present rocket propulsion systems and newer concepts arise numerous problem areas, some of which in themselves can be classified as new technologies. Seal materials and propellants for such systems fall well into the category of new technology, and from the limited information evidenced to date will have to be developed as such. True, some of the knowledge associated with conventional fluid sealing can, and will be applied to rocket seals. However, one is immediately made aware of the radical differences between the two sealing systems. Such a comparison is shown in Table 5 for dynamic and static contact sealing, where seal material and fluid requirements are compared for conventional and propellant sealing.

Table 5

CONVENTIONAL SEALING REQUIREMENTS VS. PROPELLANT SEALING FOR CONTACT SEALS

	CONVENTIONAL SEALING	PROPELLANT SEALING
1.	Low leakage is often acceptable	Leakage is critical
2.	Reasonable material wear is acceptable in many instances because of ease of seal material replacement and/or 1 above.	Wear is critical because of stringent leakage requirements, and inability for material replacement.
3.	Most sealing fluids are not damaging when leakage occurs, are not toxic or reactive in nature	Many of the propellants are highly reactive, thermally unstable and toxic in nature.
4.	The liquids being sealed often provide a load carrying film between contacting seal surfaces resulting in negligible wear.	The liquid propellants in general are poor (like water) in their ability to provide a load carrying fluid film between their sliding surfaces. A comparison of the viscosity of various propellants with water and oil can be seen in TABLE G-6.

	CONVENTIONAL SEALING	PROPELLANT SEALING
5.	Selection of seal material is generally not critical when hydrodynamic operating conditions prevail, but is moreso when boundary operating conditions prevail.	The low viscosities of the propellants indicate that one would expect boundary seal operating conditions most of the time, depending upon load and velocity. The wear factor is extremely important when boundary conditions prevail.
6.	In general, most of the seal materials and fluids associated with conventional sealing are not prone to react together.	The very character of the propellant makes the choice of seal material (irregardless of how compatible the materials are from the standpoint of friction and wear) extremely critical. Violent reactions have been known to occur.
7.	For extremely high loads where severe metal to metal contact is encountered, active chemical materials such as chlorine, sulfur and lead compounds are often added to the fluid for the purpose of reducing wear and/or surface damage. These additives react chemically to form a low-shear surface layer.	Chemical additives, like the seal materials per se will have to be carefully selected because of the high reactivity of many of the propellants. Inertness to the propellant is probably mandatory.

Having discussed the material-propellant needs from the broader operational viewpoint one may summarize these pertinent requirements:

A. A Contact seal material must possess an extremely low wear rate, and preferably low friction.

B. Compatibility between seal material, coating and additive is essential. A mismatch could result in:

1. Seal material going into solution (Solubility tendencies)
2. Alteration of seal material properties
3. Chemical Reaction (Corrosion)
4. Detonable mixture causing explosion

C. The mechanical and physical properties of the seal material must of course meet sealing design requirements. These pertinent material properties, along with controllable and uncontrolling variables were discussed previously.

It might be well at this point to re-emphasize the critical nature of (B) above. For example, it is known that fluorinated hydrocarbons from detonable mixtures with chlorine trifluoride, and titanium has been reported as being sensitive to impact, causing a spontaneous explosion in liquid fluorine. There is, unfortunately, no reliable analytical means at present for predicting fuel-material compatibility, so that we must continue to rely on experimental data as the basis of selection.

Friction and Wear

From our study of this subject thus far, it can be said that the state of the knowledge of seal materials operating in rocket propellant atmospheres is poor. Most of the seal friction and wear studies to date have been conducted in conventional fluids like oil, water, air, and some others of lesser extent in gases like helium and nitrogen. Limited efforts on material-propellant friction and wear studies have been conducted by Rocketdyne (Ref. 1). Some of the more successful materials and their respective lubricating propellants are tabulated in Table 7.

Efforts which may be applied to the problem of filling the void in seal material-propellant technology become apparent. Current recommendations for advances in this field are as follows:

1. Correlate present available seal test data with propellant-material requirements. The type of data more representative of propellant-seal material operating conditions would evolve from such investigations as past studies made in water (low viscosity), dry sliding studies, and gas environments such as Helium, Nitrogen, and Hydrogen. Examples of the latter can be seen in Tables 8 and 9.
2. Draw upon the knowledge accumulated in the structural materials-propellant compatibility studies for the consideration of potential

dynamic, and particularly static seal materials.

3. Devise a program based on conducting seal material tests in the various propellants.

New Concepts

In addition to the basic study discussed above, new concepts should be considered and explored to the extent they merit. Some concepts of potential value are as follows:

1. Resilient Composite Material

This is a recent development in which a metal-fiber skeleton is impregnated with a softer pliable material. The Former provides elasticity, and the latter provides plasticity. An example is a stainless stell or molybdenum skeleton impregnated with tin or silver.

2. Free Energy Relationships

Consideration should be given to the possibility of using free-energy relationships between propellant or gas, and seal materials, for the purpose of judging the probability of an existing chemical reaction between the lubricant (fuel or gas) and the seal substrate. In effect, one would be utilizing thermodynamic material properties to predict chemical reactivity. While it is realized that other variables will enter into the already complex situation, it is hoped that a trend might be effected. Subsequently, it would be determined if the chemisorbed film where limited chemical reactivity can exist, would prove to be a beneficial film from the standpoint of improving lubrication for positive contact seals. Some work has already been initiated in this area on rolling contact bearings by other researchers.

3. Lubrication Additives for Propellants

A brief review of potential additives that might prove suitable as lubricants in the presence of fuels and oxidizers should be made. Complete inertness

of the lubricant with the propellant would appear to be the major prerequisite. Work of this nature was conducted at ATL (Ref. 4) on normal propyl nitrate (NPN). Several additives proved to be effective in improving the lubricating characterisitcs of NPN for bearing purposes.

REFERENCES:

1. Propellant Lubrication Properties Investigation, M.F. Butner, Rocketdyne, WADD-TR-61-77, June, 1961.
2. Development of Seals for Rocket Engine Turbopumps, J.E. Wolfe, R. E. Connelly, ASLE Preprint No. 58LC-5, Oct. 1958.
3. High-Speed Rotating Shaft Seal for a Helium-Cooled Reactor Application, W. A. Glaeser, F. Orcutt, Battelle Memorial Institute, May, 1960.
4. Effects of Additives on the Lubrication Characteristics of Normal Propyl Nitrate, R. E. Lee, Jr., May 1955, On File at Advanced Technology Laboratory, G.E. Company.

Table 1

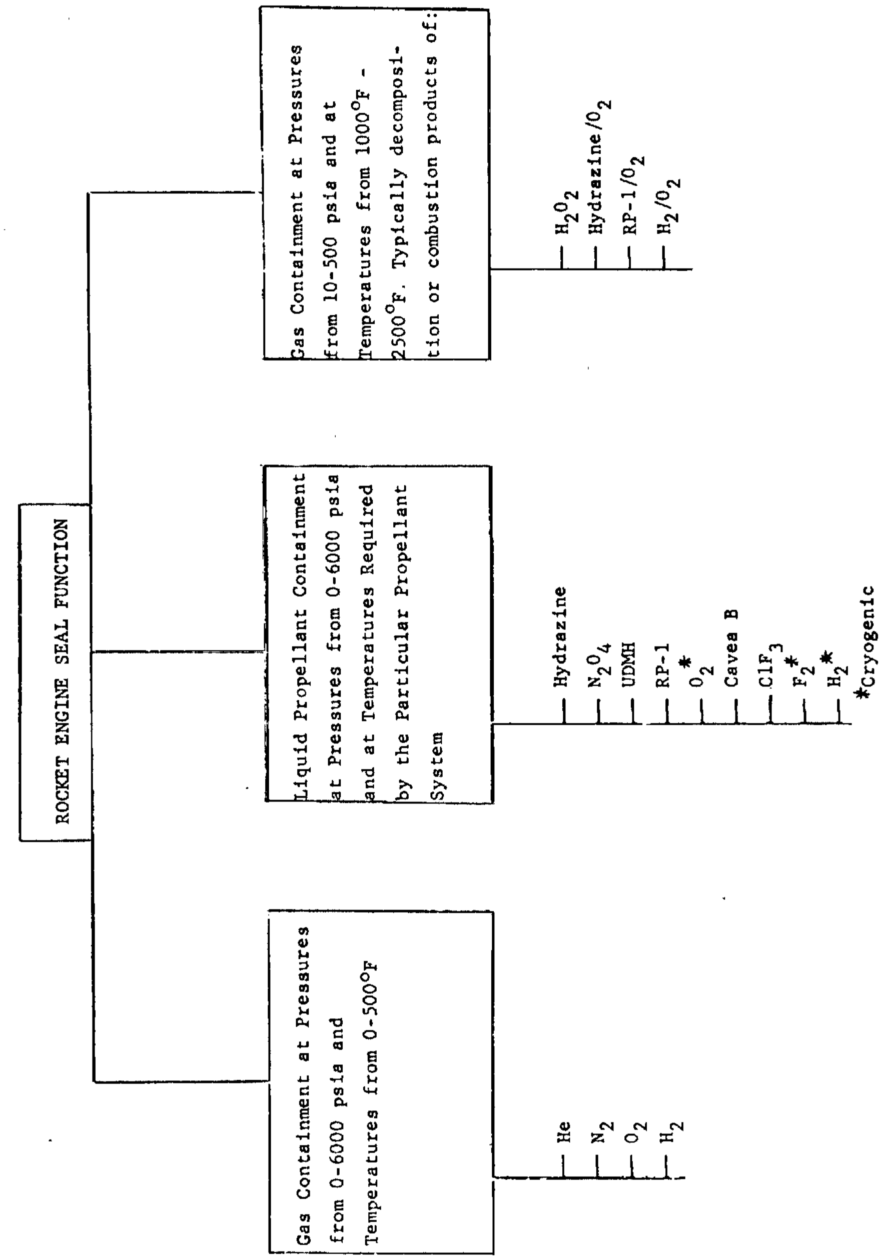


Table 2

MATERIAL PROPERTIES EFFECTING ROCKET SEAL PERFORMANCE

	STATIC	DYNAMIC Controlled Clearance	DYNAMIC Positive Contact
Dimensional Stability	2	1	1
Compressive Strength	1	2	1
Impact Strength	1	3	1
Thermal Shock Resistance	2	2	2
Hardness	2	3	1
Modulus of Elasticity	1	3	2
Thermal Conductivity	1	1	2
Permeability	1	2	1
Coefficient of Thermal Expansion	1	1	2
Wear Resistance	3	3	1
Corrosion Resistance	1	1	1
Homogenity	1	2	1

LEGEND

1 Highly Effective
2 Effective
3 Little to no Effect

Table 3

CONTROLLABLE AND UNCONTROLLABLE VARIABLES EFFECTING SEAL MATERIAL SPECIFICATION

Uncontrollable Variables (Operating Conditions)

- Temperature
  - a. Cryogenic
  - b. To 2000 F
- Containment Pressure
- Ambient Pressure
- Fluid
- Vibration Spectrum
- Radiation
- Space Availability
- Life requirements

Controllable Variables (Seal Design Parameters)

- Configuration
- Compatibility (friction and wear)
  - a. Solubility
  - b. Oxidation
- Surface Finish
  - a. Direction of contours
  - b. Class of finish
- Impermeability
  - a. To gases
  - b. To liquids
- Load
- Velocity

Table 4

CLASSES OF POTENTIAL ROCKET ENGINE SEAL MATERIALS PRESENTLY UNDER CONSIDERATION

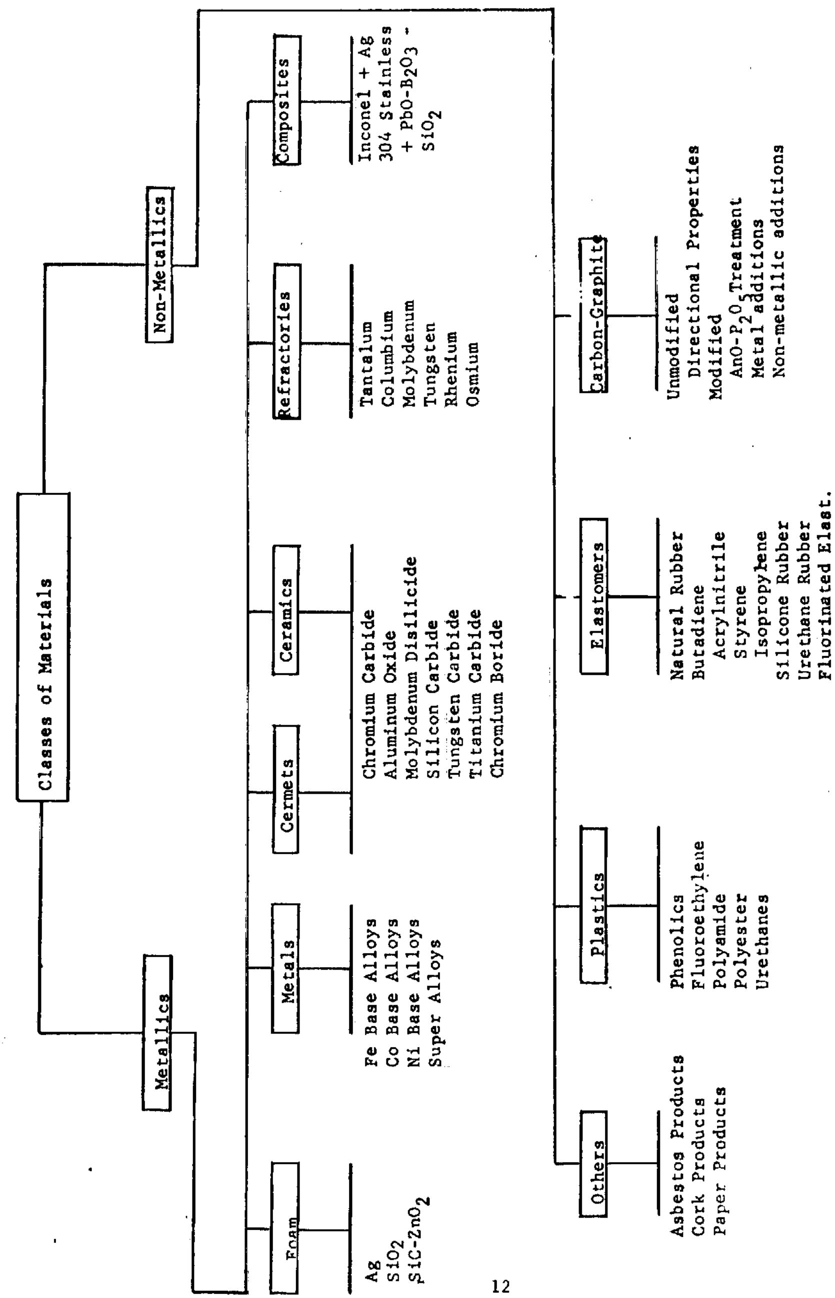


Table 6

COMPARISON OF ABSOLUTE VISCOSITIES FOR VARIOUS FLUIDS

FLUID	FORMULA	ABSOLUTE VISCOSITY (Centipoise)
WATER	H_2O	1.00 at 70°F
OIL SAE 10	Light Petroleum Oil	70 at 70°F
MONO PROPELLANTS		
Hydrazine	N_2H_4	0.90 at 77°F
UDMH	$(CH_3)_2NNH_2$	0.509 at 77°F
Hydrogen Peroxide	H_2O_2	1.156 at 77°F
FUELS		
Hydrogen (Liquid)	H_2	0.013 at -439°F
RP-1 Hydrocarbon	---	1.17 at 100°F
OXIDIZERS		
Nitrogen Tetroxide	N_2O_4	0.342 at 100°F
Fluorine (Liquid)	F_2	0.257 at -374°F
Chlorine Trifluoride	CF_3	0.412 at 77°F
Oxygen	O_2	0.15 at -280°F

Table 7

FRICTION AND WEAR TESTS (Ref. G-1)

Essentially six materials were run under sliding conditions in various propellants for the purpose of determining their friction and wear behavior. The more successful material combinations are listed under the respective propellant in which they were tested. The test specimens consisted of a 3/8 inch diameter rider loaded against a rotating disc. Loads were Hertzian in nature and sliding velocities from 1500 fpm to 9750 fpm. Further testing details and material limitations can be found in Reference 1.

PROPELLANT	MATERIAL COMBINATION
RP-1	Nitralloy 135 M vs. 9310 440C vs. 440C Nitralloy 135 M vs. Nitralloy 135 M 9310 vs. 440C or 52100 Glass supported Teflon vs. 52100 Carbon (U.S. Grade 2) vs. 52100 Phenolic linen (LLB grade) vs. 52100
Ethylene Diamine	Glass supported Teflon vs. 52100
Unsymmetrical Dimethylhydrazine	440C vs. 440C Glass supported Teflon vs. 52100
Hydrazine	Berylco 25 vs. 440C 440C vs. 440C Carbon (US Grade 2) vs. 52100
Liquid Hydrogen	Glass Supported Teflon vs. 52100 440C vs. 440C
Liquid Oxygen	440C vs. 440C Glass supported Teflon vs. 52100
Liquid Tetroxide	Carbon (US Grade 2) vs. 52100 440C vs. 440C
Inhibited Red Fuming Nitric Acid	440C vs. 440C Glass-supported Teflon vs. 52100

Table 8

FACE TYPE SEAL MATERIALS EVALUATION (Ref. 2)

Speed = 8000 ft/min Load = 200 psi, Test Fluid = Liquid N_2

Materials: Nose Piece	Materials: Mating Ring	Wear, in.: Nose	Wear, in.: Ring	Test Time min.	Comments
Carbon	Nitralloy	0.006	0.0002	5	Low leakage
Carbon	Chrome Plate	0.005	0.0001	5	Low leakage
Bronze	Nitralloy	0.001	0.0005	3	Severe scoring of bronze and excessive leakage
PTFCE	Nitralloy	0.042	NMW+	1	Seal failure
PTFE	Nitralloy	0.018	NMW	0.5	Seal failure
Carbon	Titanium-carbide composition	NMW	NMW	5	Low leakage
Aluminum (hard anodized)	Chrome plate	--	---	10 sec	Complete failure chrome ring galled anodized case worn through

+ No Measureable Wear

Table 9

HIGH SPEED ROTATING SHAFT SEAL STUDIES (Ref. G-3)

Environment = Helium Gas   Mean Face Load = 6.6 psi
Specimen Configuration = Ring Seal

Material Combination Rotor/Stator	Speed (rpm)	Wear Factor	Environment Temp. (°F)	Coeff. of Friction	Helium Leakage (lb/hr)	Bulk Stator Temp. °F	Comments
Flame-plated Cr_2C_3 / Metal Salt Impregnated Carbon Graphite	11,600	Negligible	273	0.12	0.001	150	Water Cooled, (Spray) 6-1/2 hrs. run.
	17,000	0.3	270	0.20	0.005	150	
	20,200		230	0.20	0.005	135	
"	11,600	Negligible	227	0.20	0.007	272	Dry Run 2-1/2 hrs
	17,600	0.8	267	0.29	0.008	317	
	20,400		290	0.35	0.008	345	
"	11,000	Negligible	280	0.15	0.0035	185	Spray cooled with polyalkylene glyc 3 hr. run
	18,000	Negligible	235	0.18	0.0065	200	
	20,400		230	0.20	0.0070	165	

SECTION 2
SEAL MATERIALS STUDY

J. M. Tessarzik

I. COMPATIBILITY OF MATERIALS OF CONSTRUCTION AND ROCKET PROPELLANTS

A. Static Compatibility

Two media are said to be statically compatible if they do not react chemically at all or only in a nondetrimental fashion.

In seals technology the media under consideration will usually be one material of construction and one fluid or gas, or two materials in the presence of a fluid or gas. Even in the "open air" there are only a few materials which will not react chemically at all, and also there are only a very few fluids or gases which do not react at all with any known material of construction.

Consequently the general statement that the chemical reaction should be limited to a "nondetrimental fashion" requires a quantitative definition. This definition must be application oriented. In this case, the seal designer has to determine from his application the nature and limits of chemical reactions in terms of oxide or other reaction films, corrosion and solubility rate, propellant decomposition, and sensitivity of the metal propellant combination to ignition under impact.

Reliable theoretical predictions are extremely difficult to make, particularly to the extent required for useful application. Preliminary selections of nonreacting materials will come from one of these two groups:

(a) Those for which no reaction at all is possible with the fluid under the operating conditions. An example is common base metals as components of seals for liquid helium.

(b) Those for which reaction is thermodynamically possible in that compounds between elements of the fluid compound and of the seal material can be formed with a large decrease in free energy. An example is Cu + F = CuF.

A knowledge of the free energy of the seal-propellant reactions is useful for predicting that no reaction is possible. As in (a) when reactions are possible, activation energies are usually required, making theoretical predictions as well as static compatibility tests unreliable. For purely static conditions, though, tests give a quick, relatively inexpensive answer, since no testing machines are required. The economics, expediency and certainty of testing over analysis are the motivating factors for not presenting here the fundamentals of reactions and corrosion of materials in propellants.

B. Dynamic Compatibility

There are three basic conditions under which static compatibility tests or analyses will not yield a satisfactory answer.

(1) Frictional energy is required to supply the necessary activation energy for the materials-propellant reaction. This could be the case when the reaction accomplished by a free energy drop requires a moderate activation energy.

(2) When a free energy drop takes place and the activation energy is low, so that reaction takes place, the compound which forms at the surface may be insoluble in the fluid, so that it shuts off reaction with a very thin film. This thin film may be beneficial as a lubricating film, and as long as it is not rubbed off faster than it can re-form, will also provide protection against further reaction of the metal with the propellant. However, if it is rubbed off faster than it can re-form, the reaction between the metal and the propellant may continue at a much faster rate than indicated in the static compatibility test.

(3) The frictional process, in a manner not yet understood, appears to activate reactions between components of the environment and

the surface which do not take place at all in a static system. This activation process does not appear to be related to frictional temperature effects, because it takes place at loads and rubbing speeds which, on the basis of present theory would predict asperity temperature rise of less than $1^{o}C$.

II. DESIGN INFORMATION - MATERIALS FOR ROCKET PROPELLANT SEALS

A. Static Compatibility

The elimination process of unsuitable materials should start with static compatibility check. This is nothing more than an experimentally-obtained assurance that the selected material will not react with the propellant in any dangerous way, exhibit an intolerable corrosion rate or promote propellant decomposition, and that it is free from impact sensitivity. There are various compatibility classifications in existence. Until a general standardization has been introduced, care must be taken whenever a compatibility table is used. One of the currently most complete tables* uses the following classifications:

Table 1

COMPATIBILITY CLASSIFICATIONS [(a)]

	Corrosion Resistance			
Class	Rating	Penetration Rate mils/year	Decomposition of propellant	Shock Sensitive
1	Excellent	1	No	No
2	Good	5	No	No
3	Fair	5 to 50	Some	No
4	Poor	50	Extensive	Yes

(a) The classification of a material is based on the lowest rating of any one of the three properties.

* Compatibility of Rocket Propellants with Materials of Construction, Defense Metals Information Center, By W. K. Boyd and E. L. White, Columbus 1, Ohio.

TABLE 2

LIQUID PROPELLANTS REPRESENTED IN THE CITED REFERENCES

	Ref. (1)	Ref. (2)	Ref. (3)	Ref. (4)
Nitrogen Tetroxide	x		x	
UDMH	x	x	x	x
Hydrazine	x	x	x	x
Pentaborane			x	x
LOX	x	x	x	x
Liquid Hydrogen	x	x	x	x
Fluorine (liquid and gas)	x			
Liquid Fluorine			x	x
RP-1		x		
NTO		x		
IRFNA		x		
Ethylene Diamine		x		
CTF	x		x	
BTF	x			
BPF	x		x	
IPF	x			
Chlorine Trifluoride				x
PF	x			
Hi CAL-3	x			
Ammonia	x		x	
Nitric Acid			x	
RFNA	x			
WFNA	x			
Concentrated FNA	x			
Hydrogen Peroxide	x		x	
Monoethyl Trifluoride			x	
Nitrogen Trifluoride			x	
Aerozine 50 (50% Hydrazine/ 50% UDMH)-See Note (5)			x	x

(1) Compatibility of Rocket Propellants with Materials of Construction (Defense Metal Information Center)
(2) Propellant Lubrication Properties Investigation (Rocketdyne) (Part I)
(3) Liquid Propellant Handbook (Liquid Propellant Agency)
(4) Materials Compatibility-Propellants (Space Technology Laboratories)
(5) See "Propellant Lubrication Properties Investigation" by Rocketdyne (No. WADD-TR-61-77 Part II) for bearing, gear, sliding and rolling tests in this propellant.

B. Compatibility of Solid Film Lubricants with Rocket Propellants

Solid film lubricants consist in many cases of a mixture of materials which vary widely in their reaction to rocket propellants. Since each of the components may affect compatibility, and this to varying degrees depending upon its quantitative ratio to the whole, one will probably have to rely on tests for each solid film lubricant. Even one basic solid lubricant mixture may behave quite differently if used with different types of binder, as the following specification demonstrates.

The "Martin Materials Specification MMS N 375, Rev. 1," dated 6/12/62 lists three lamellar-type solid lubricants and their compatibilities with rocket propellants (Ref. 6).

Type I Solid Film Lubricant:

This is a MoS_2-graphite mixture which is rubbed or sprayed on. It provides no corrosion resistance. It should be noted here that plain MoS_2 acts as a catalyst when placed in direct contact with UDMH, and also causes deterioration of some fuels.

Type II

This material is a resin bonded MoS_2-graphite lubricant. It provides corrosion protection under normal (atmospheric) conditions only and will withstand fuel splash. It may not be immersed in UDMH (acts as a catalyst) or N_2O_4, (softens up and becomes useless).

Type III

The lubricant consists basically of MoS_2 and graphite in a sodium silicate carrier. It appears not to be affected by any hazardous materials as N_2O_4, UDMH, 70% solution nitric acid, ethylene diamine.

It appears that none of the commonly used solid lubricants provides good corrosion protection. The best one can hope for is that the lubricant itself will not react with the propellant.

C. Compatibility of Grease-type Lubricants with Rocket Propellants

The great difficulties which accompany the design of "unlubricated" (or solid film lubricated) sliding systems for liquid-fuel powered rockets has led to a vigorous search for grease-type lubricants useable in rocket propellant atmospheres. It is claimed (Ref. 7, 8, 9, 10) that a physically stable lubricant of greaselike consistency may be prepared by thickening mixed perfluoroalkylamines (alkyl = C_4 to C_6) with polytetrafluoroethylenes of relatively low molecular weight and that the product is inert with a wide variety of rocket motor liquid fuels and oxidizers--i.e. JP-4, ethyl alcohol, unsymdimethylhydrazine, diethylenetriamine, 1 to 1 UDMH and hydrazine, 90% hydrogen peroxide, inhibited red fuming nitric acid, liquid oxygen, and nitrogen tetroxide (with the possible exception of N_2O_4 in which the base fluid is somewhat soluble). The lubricant is also inert to most elastomers used in missile and rocket motor systems and has antiwear and extreme pressure properties that are approximately the same as those of conventional petroleum and synthetic greases. Further testing in operating equipment will be required for such applications where sliding speed, provisions for lubricant pockets and general design make it feasible to consider grease-type lubrication.

III. DESIGN INFORMATION - FRICTION AND WEAR

The second step in elimination of unsuitable materials of construction for a seal used on a particular rocket propellant consists of application of the criteria of friction and wear.

Friction and wear are of prime importance to those seal elements which are in rubbing contact with other parts. This elimination step applies therefore only to contacting dynamic seals.

The obviously easiest way of finding a construction material for a sealing element would be a table of experimental results similar to the compatibility data, referring to the exact geometric form desired, and tested under identical operating conditions. Until such information becomes available, an approximation-by-steps-of-diminishing-reliability approach must suffice.

Table 8 shows a ranking of the available data:

Table 8

FRICTION AND WEAR TEST DATA

Sealing Elements in Rocket Propellants	Sealing Elements in Air, Gases, Low Visc. Fl.	Sealing Elements in Vacuum
A	B'	C''
Pin & Disk Tests in Rocket Propellants	Pin & Disk Tests in Air, Gases, Low Visc. Fl.	Pin & Disk Tests in Vacuum
B	C'	D'
Brg. & Gear Tests in Rocket Propellants	Brg. & Gear Tests in Air, Gases, Low Visc. Fl.	Brg. & Gear Tests in Vacuum
C	D	E

In the upper left-hand corner of Table 8 the most valuable and directly applicable data is listed: the friction and wear characteristics of sealing elements in the particular and/or other rocket propellants. Ideally, this would be the only data the seal designer need have, but since it is almost nonexistent, results from pin and disk tests and bearing and gear tests in rocket propellants, free atmosphere or vacuum and of seal elements in other atmospheres must be used for preliminary design purposes.

If the choice of materials found by this method appear too limited due to lack of previous data taking and not due to the nature of the propellant and the severity of the operating conditions, then it will become necessary to supplement it with new, and specific, experimental data.

The theoretical considerations found under "Friction and Wear" should be a helpful guide to the most efficient and successful selection of candidate materials.

Friction and Wear Test Data - Group A

Face type seal tests in Liquid N_2

Ref. (115) "Development of Seals for Rocket Engine Turbopumps", J. E. Wolfe, R. E. Connelley, ASLE Preprint No. 58LC-5-October, 1958.

Speed: 8000 ft/min.

Load: 200 psi

Materials		Wear, in.		Test Time	
Nose Piece	Mating Ring	Nose	Ring	Min.	Comments
Carbon	Nitralloy	0.006	0.0002	5	Low Leakage
Carbon	Chrome Plate	0.005	0.0001	5	Low Leakage
Bronze	Nitralloy	0.001	0.0005	3	Severe scoring of bronze and excessive leakage
PTFCE	Nitralloy	0.042	NMW*	1	Seal failure
PTFE	Nitralloy	0.018	NMW	0.5	Seal failure
Carbon	Titanium-carbide composition	NMW	NMW	5	Low leakage
Aluminum (hard anodized)	Chrome plate	--	--	10 sec.	Complete failure chrome ring gal[illegible] anodized case worn through

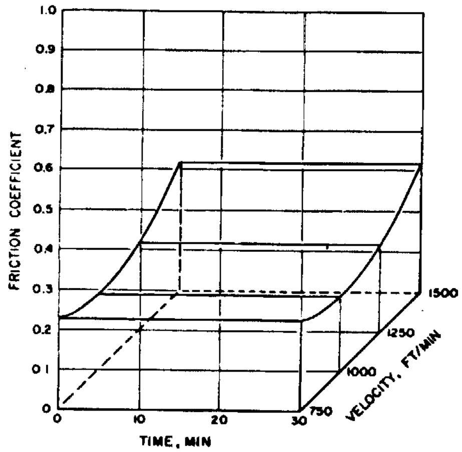
*No measureable wear

FRICTION AND WEAR TEST DATA - GROUP B

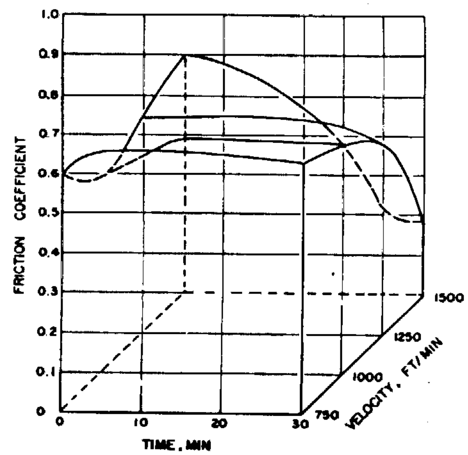
Sliding Tests Performed in 50-50 UDMH/N_2H_4

Reference 111· "Propellant Lubrication Properties Investigation" WADD-TR-61-77 Part II

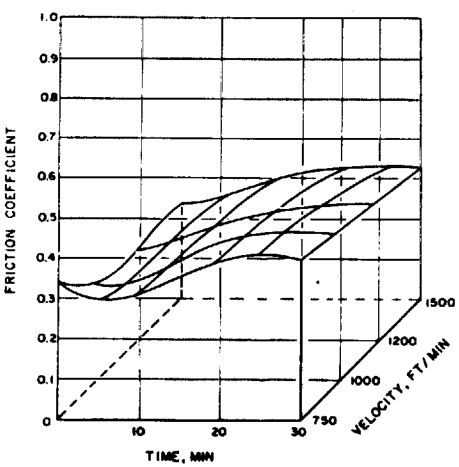
Test Equipment: Pin-On-Disk (3/8" Dia. Pin)



B25 vs B25

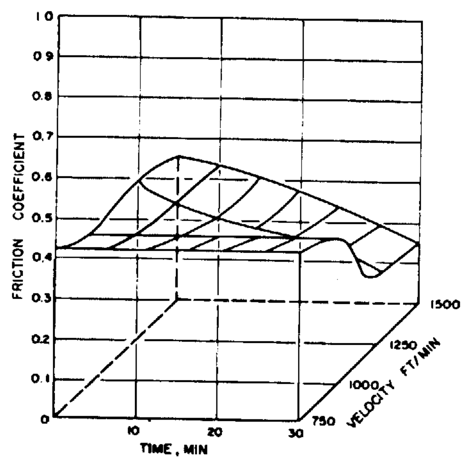


440C vs 440C



303 vs 310

(Continued)



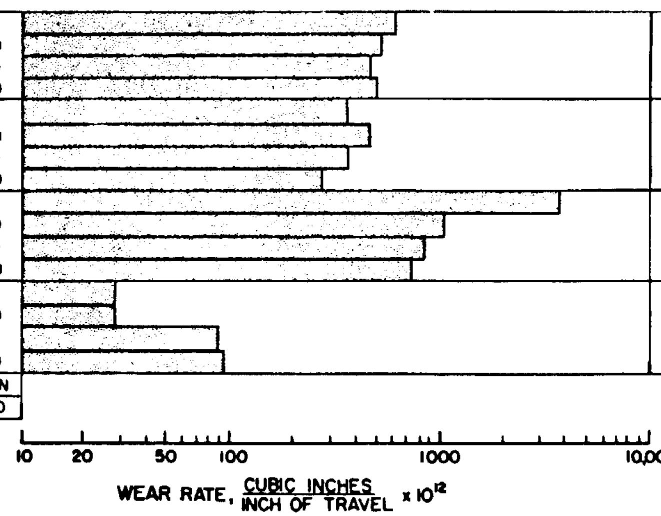
410N vs 410N

Notes:

(1) B-25 - Berylco 25 (Beryllium Copper)
(2) 440 - 440-C Corrosion-Resistant Steel
(3) 410N - 410 " " " , nitrided, 0.0005 to 0.010
(4) 303, 310 - 300 Series Stainless Steel

WEAR RATES

RIDER MATERIAL	PLATE MATERIAL	SPEED FT/MIN	WEAR RATE	AVG
B-25	B-25	750	603	526
		1000	539	
		1250	466	
		1500	496	
440-C	440-C	750	359	365
		1000	464	
		1250	367	
		1500	270	
303	310	750	3680	1567
		1000	1010	
		1250	852	
		1500	728	
410N	410N	750	28	59
		1000	28	
		1250	87	
		1500	93	



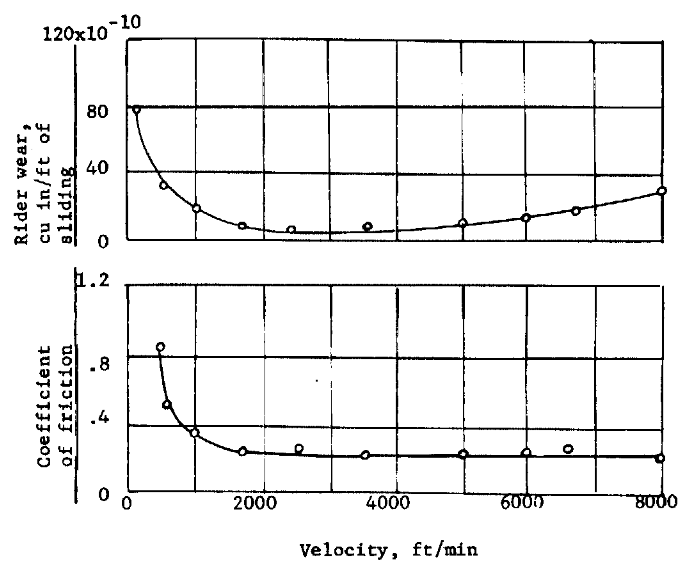
Friction and Wear Test Data - Group B

Sliding tests performed in Liquid Oxygen.

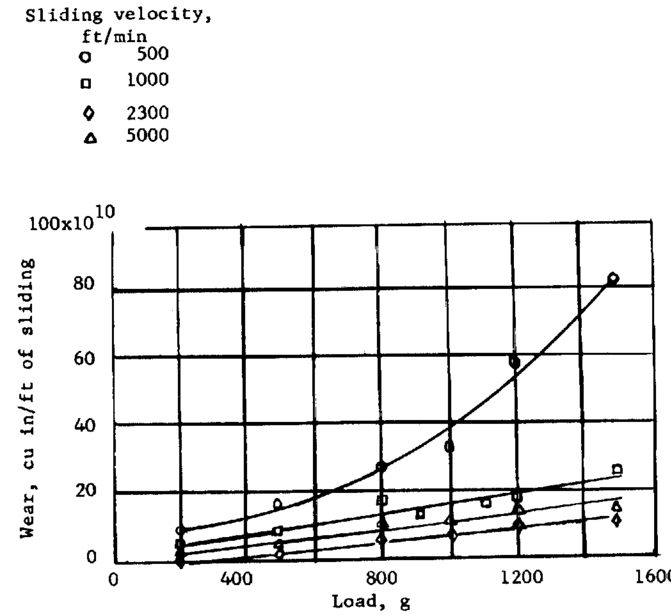
Ref: (116) "Boundary Lubrication Characteristics of a Typical Bearing Steel in Liquid Oxygen", W. F. Hady, G. P. Allen, and R. L. Johnson; NASA T N-D - 1580, February, 1963.

Test Equipment: pin-on-disk (3/10" diam. pin).

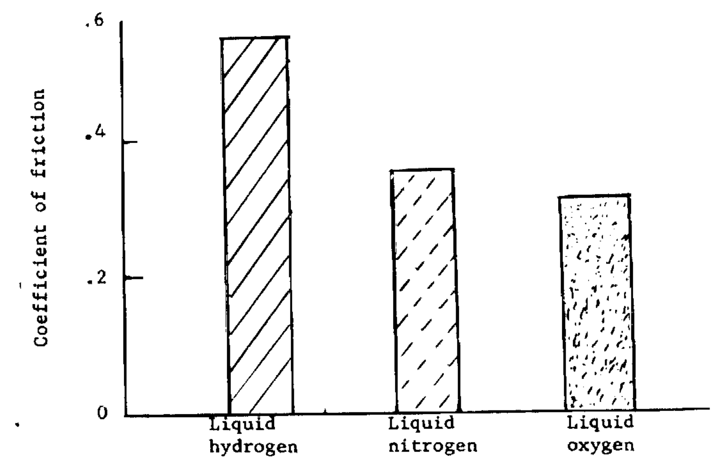
Materials: AISI 440-C chromium martensitic stainless steel (R_C 52-54) (rider and disc).



Variation of rider wear and coefficient of friction with velocity for 440-C stainless steel sliding on 440-C stainless steel in liquid oxygen. Load, 1000 grams; duration, 1 hour.



Wear of 440-C stainless steel sliding on 440-C stainless steel in liquid oxygen at various loads and sliding velocities. Duration, 1 hour.



Coefficient of friction of 440-C stainless steel sliding on 440-C stainless steel in reducing, inert, and oxidizing cryogenic liquids. Sliding velocity, 2300 feet per minute; load; 1000 grams; duration, 1 hour.

FRICTION AND WEAR TEST DATA - GROUP B

Sliding Tests Performed in RP-1 (Mil-F-25576, Liquid Hydrogen (LH_2), Ethylene Diamene (EDA), Unsymmetrical Dimethylhydrazine (UDMH), Hydrazine (N_2H_4), Liquid Oxygen (LOX), Nitrogen Tetroxide (N_2O_4), Inhibited Red Fuming Nitric Acid (IRFNA).

Reference 112: Propellant Lubrication Properties Investigation, WADD-TR61-77, Part I, by M. F. Butner (Rocketdyne).

Test Equipment: Pin-on-Disk (3/8" Dia. Pin), sprayed or submerged in propellant.

Specimens			Load, lb	Total Distance Travelled, in. x 10^6	Button Wear by Volume, cu in. x 10^6	Button Wear Rate, cu in./in. x 10^{12}	Friction Coefficient	
Plate	Button							
Material	Material	Velocity					Initial	Stabilized
			PROPELLANT RP-1					
440-C	440-C	1570	0.65	0.565	3.30	7.26	0.185	0.185
	135M	2100	0.65	0.754	2.50	2.79	0.15	0.15
	B-25	2620	0.65	0.942	7.00	7.43	0.05	0.05
	440-C	3140	0.65	1.13	1.00	1.86	0.15	0.15
B-25	B-25	1570	0.6984	0.54	7.70	13.5	0.41	0.06
	135M	2100	0.698	0.72	0.30	0.331	0.36	0.017
	440-C	2620	0.698	0.9	0.52	0.530	0.31	0.04
	B-25	3140	0.581	1.08	100.0	97.4	0.51	0.08
9310	135M	1570	0.698	0.54	0.93	1.71	0.42	0.05
	440-C	2100	0.698	0.72	3.80	5.56	0.365	0.04
	B-25	2620	0.698	0.90	28.00	31.1	0.56	0.035
	52100	3140	0.698	1.08	6.10	5.56	0.38	0.04
135M	Al	1570	0.77	0.54	120.00	220.0	0.16	0.05
	Al_2O_3	2100	0.77	0.72	0.28	0.389	0.30	0.08
	52100	2620	0.77	0.90	8.00	8.9	0.52	0.07
	135M	3140	0.77	1.08	5.20	4.82	0.26	0.05
440-C	B-25	4750	0.77	1.71	-	-	1.0	0.54
	Bakelite	6330	0.77	2.28	-	-	0.58	0.44
	Carbon	7910	0.77	2.85	-	-	0.73	0.52
	GST	9500	0.77	3.42	-	-	0.69	0.59
440-C	B-25	1570	0.77	-	-	-	0.11	0.04
	Bakelite	2100	0.77	-	-	-	0.12	0.07
	Carbon	2620	0.77	-	-	-	0.165	0.12
	GST	3140	0.77	-	-	-	0.185	0.085
440-C	B-25	1570	0.77	1.832	140.0	76.3	0.09	0.09
	Bakelite	2100	0.77	2.443	6.5	2.65	0.04	0.04
	Carbon	2620	0.77	3.054	8.0	2.62	0.06	0.06
	GST	3140	0.77	3.664	8.5	2.32	0.06	0.06

(Continued)

Specimens			Load, lb	Hertz Stress, psi	Total Distance Travelled, in. x 10^6	Button Wear by Volume, cu in. x 10^6	Button Wear Rate, cu in./in. x 10^{12}	Friction Coefficient	
Plate	Button								
Material	Material	Velocity, fpm						Initial	Stabilized
			PROPELLANT EDA						
440-C	440-C	1570	0.77	105,000	0.622	9.0	14.45	0.52	0.05
	B-25	2100	0.77	90,000	0.829	88.0	106.0	0.38	0.03
	B-25	2620	0.77	90,000	1.040	70.0	67.3	0.53	0.48
	440-C	3140	0.77	105,000	1.242	10.0	8.05	0.43	0.20
440-C	Carbon	4720	0.77	-	1.855	72.0	38.8	0.17	0.17
	GST	6290	0.77	-	2.475	80.0	32.3	0.082	0.082
	Carbon	7860	0.77	-	3.090	80.0	25.9	0.082	0.082
	GST	9440	0.77	-	3.71	105.0	28.3	0.11	0.11
			PROPELLANT UDMH						
440-C	Carbon	4870	0.77	-	1.96	11.1	5.66	0.10	0.10
	GST	6500	0.77	-	2.61	10.7	4.1	0.075	0.075
	Carbon	8120	0.77	-	3.26	48.0	14.71	0.10	0.10
	GST	9740	0.77	-	3.92	10.8	2.76	0.051	0.051
440-C	Al	1610	0.77	72,600	0.628	420.0	669.0	0.41	0.14
	52100	2150	0.77	105,000	0.84	70.0	83.0	0.89	0.08
	B-25	2680	0.77	90,000	1.05	280.0	266.0	0.35	0.12
	440-C	3220	0.77	105,000	1.26	9.5	7.54	0.20	0.15
			PROPELLANT N_2H_4						
440-C	Al	1635	0.77	72,600	0.648	1050.0	1620.0	0.20	0.20
	440-C	2180	0.77	105,000	0.864	25.0	28.9	0.30	0.06
	B-25	2725	0.77	90,000	1.08	2.9	2.68	0.03	0.04
	440-C	3270	0.77	105,000	1.295	2.0	1.54	0.15	0.075
440-C	Carbon	4875	0.77	-	1.755	61.0	28.8	0.155	0.155
	GST	6500	0.77	-	2.34	511.0*	218.0	0.18	0.18
	GST	8125	0.77	-	2.925	125.0*	42.5	0.18	0.18
	GST	9750	0.77	-	3.55	125.0*	36.2	0.29	0.29
			PROPELLANT LOX						
440-C	Carbon	5190	0.77	-	1.87	80.0	42.7	0.34	0.16
	GST	6920	0.77	-	2.49	22.0	8.8	0.036	0.036
	Carbon	8650	0.77	-	3.115	220.0	70.7	0.35	0.17
	GST	10,380	0.77	-	3.72	52.0	14.0	0.065	0.065

(Continued)

Specimens Plate Material	Button Material	Button Velocity, fpm	Load, lb	Hertz Stress, psi	Total Distance Travelled, in. x 10^6	Button Wear by Volume, cu in. x 10^6	Button Wear Rate, cu in. in. x 10^{12}	Friction Coefficient Initial	Friction Coefficient Stabilized
PROPELLANT LOX									
440-C	440-C	1572	0.77	105,000	0.566	6.0	10.6	0.20	0.11
	440-C	2095	0.77	105,000	0.755	4.5	5.95	0.21	0.08
	B-25	2620	0.77	90,000	0.925	40.0	43.2	0.42	0.30
	440-C	3140	0.77	105,000	1.132	16.0	14.12	0.33	0.15
PROPELLANT LH_2									
440-C	440-C	1572	0.77	105,000	0.566	2.3	4.06	0.09	0.09
	B-25	2095	0.77	90,000	0.755	30.0	39.7	0.075	0.45
	440-C	2620	0.77	105,000	0.925	8.4	9.09	0.14	0.14
	440-C	3140	0.77	105,000	1.132	8.8	7.76	0.13	0.13
440-C	B-25	5190	0.77	90,000	1.742	22.0	12.6	0.15	0.15
	Bakelite	6920	0.77	-	2.32	90.0	38.8	0.13	0.13
	Carbon	8650	0.77	-	2.90	24.0	8.3	0.10	0.10
	GST	10,380	0.77	-	3.49	10.0	2.87	0.11	0.11
PROPELLANT N_2O_4									
440-C	Carbon	4875	0.77	-	1.755	1.5	0.854	0.02	0.03
	GST	6500	0.77	-	2.34	8.4	3.59	0.025	0.025
	Carbon	8125	0.77	-	2.92	5.0	1.71	0.03	0.04
	GST	9750	0.77	-	3.51	23.0	6.55	6.55	0.120
440-C	440-C	1572	0.77	105,000	0.566	0.69	1.22	0.3	0.011
	440-C	2345	0.77	105,000	0.756	0.75	0.991	0.27	0.015
	440-C	2940	0.77	105,000	0.443	0.9	0.955	0.29	1.01
	440-C	3520	0.77	105,000	1.132	1.7	1.50	0.18	0.011
PROPELLANT IRFNA									
440-C	440-C	1760	0.77	105,000	0.633	1.4	2.21	0.23	0.03
	440-C	2345	0.77	105,000	0.844	1.9	2.25	0.18	0.025
	440-C	2940	0.77	105,000	1.055	3.8	3.60	0.11	0.03
	440-C	3520	0.77	105,000	1.267	4.0	3.16	0.075	0.045
440-C	Carbon	5100	0.77	-	1.835	740.0*	403.0	0.05	0.05
	GST	6800	0.77	-	2.450	23.0	9.35	0.025	0.025
	Carbon	8500	0.77	-	3.060	1350.0*	441.0	0.09	0.09
	GST	10,200	0.77	-	3.672	30.0	8.18	0.07	0.07

*in part by rider erosion

Friction and Wear Test Data - Group B[1]

Shaft Seal (ring seal) tests performed in Helium gas

Ref. (117) "High Speed Rotating Shaft Seal for a Helium-Cooled Reactor Application," W.A. Glaeser, F. Orcutt; Battelle Memorial Inst., May, 1960.

Face Load: 6.6 psi

Materials: rotor - flame plated Cr_2C_3

stator - metal salt impregnated carbon graphite

Test	Speed RPM	Environ. Temp(°F)	Bulk Stator Temperature	Coeff. of Friction	Helium Leakage	Wear Factor
1	11,600	273	150	0.12	0.001	Negligible
2	17,000	270	150	0.20	0.005	0.3
3	20,200	230	135	0.20	0.005	
4	11,600	227	272	0.20	0.007	Negligible
5	17,600	267	317	0.29	0.008	0.8
6	20,400	290	345	0.35	0.008	
7	11,000	280	185	0.15	0.0035	Negligible
8	18,000	235	200	0.18	0.0065	Negligible
9	20,400	230	165	0.20	0.0070	

Note: Test 1 - 3: Water Cooled (spray) 6.5 hours run

4 - 6: Dry-run 2.5 hours

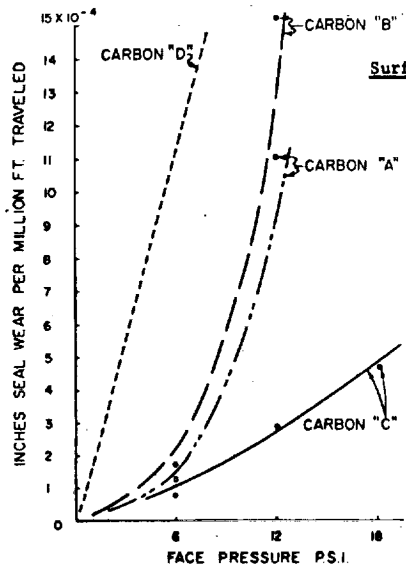
7 - 9: Spray cooled with polyalkylene glycol 3 hour run

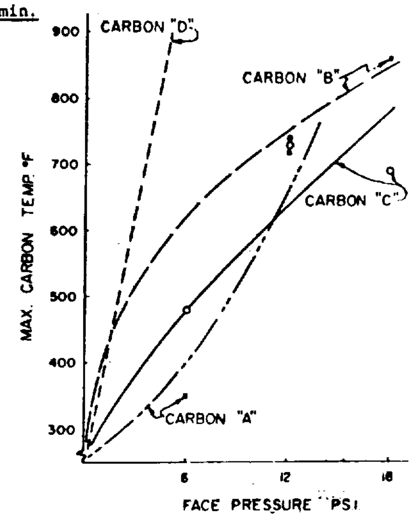
Sliding Seal Test Data in Air

Reference 113: "Testing High Speed Seal Carbons" by R. R Payton and W. R. Shobert, ASLE Trans. 5, No. 2, p. 308-314, (1962).

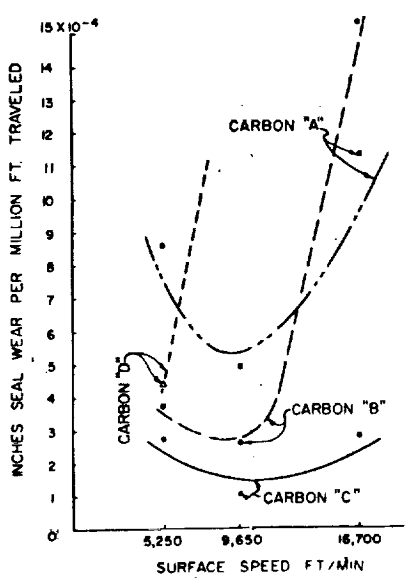
Test Equipment: Simulated seal application, 4" dia. carbon face seals.
Room Temperature
Disk Material: AISI 4190 Steel, Chrome Plated
Test Duration: 6 hrs.

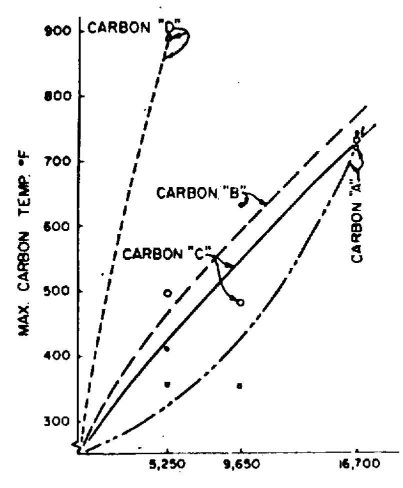
Surface Speed = 16,700 ft/min.





Face Pressure = 12 psi





PHYSICAL PROPERTIES

Carbon-graphite	A (resin impreg.)	B (org./inorg. impreg.)	C (anti-oxid impreg.)	D (premium graphite)
Scler. hardness	80-95	40-60	70-90	50-70
App. density, gm/cc	1.70-1.80	1.65-1.80	1.75-1.95	1.55-1.70
Trans. Str.#	8.5-11.5	4-7	6-10	4-7
Temp. limit, °F	500	1000	1150	800

Thousands of psi.

FRICTION AND WEAR TEST DATA - GROUP B[1]

Sliding Tests of Refractory Metal Type Seals in Air

Reference 66 : "An Investigation of Seal Materials for High-Temperature Application" by R. H. Baskey, ASLE Trans. Vol. 3, No. 1, p. 116-123.

Test Equipment: Pin-on-disk machine (rectangular pin)
Velocity: 29,000 ft/min.
Load: 14 lb/in^2

Summary of Wear Data

Rectangular Pins Against Tool Steel

Pin material	Pin wear (in hr) Ambient temp. 75 F	600 F	1050 F	Temp. of pin (°F)	Pin rating	Remarks: P-pin, D-disk
Refractory Hard Metals						
Pure MoB	0.006	0.003	0.006	-	Good	P-oxidized; D-fine scratches
Pure CrB_2	0.192	-	-	-	N.G.	P-fractured
Pure WSi_2	0.026	0.022	0.012	1200	Fair	P-mild scoring
Pure VB_2	0.0096	0.012	0.024	1800	Fair	P-transferring to disk
Pure CrN	-	-	-	-	N.G.	P-fractured
Pure VSi_2	0.162	0.186	0.22	1200	N.G.	P-galled-pulled out
Pure $CrSi_2$	0.264	-	-	-	N.G.	P-fractured; D-grooved
Pure Ti_5Si_3	0.054	-	-	-	N.G.	P-eroded
Pure VN	0.018	0.078	0.090	1300+	N.G.	P-edge fracturing
Pure WB	-	-	-	-	N.G.	P-shattered 15 sec.
Pure TiB_2	0.012	-	-	-	N.G.	P-fractured
Binary Alloys						
70% VSi_2-30% Ni	0.018	-	-	-	N.G.	P-fractured
60% MoB-40% Ni	0.066	-	-	1800	N.G.	P-fractured; D-extreme pickup
60% CrB_2-40% Ni	0.70	-	-	1800+	N.G.	P-poor thermal cond.
80% WSi_2-20% Ni	0.090	0.090	0.948	1500+	N.G.	P-partially scored
40% CrN-60% Ni	0.168	0.384	0.432	1500+	N.G.	D-scored
65% WB-35% Ni	-	-	-	-	N.G.	P-fractured
Ternary Alloys						
60% MoB-20% Ni-20% Ag	0.018	0.042	0.102	1300	Poor	P-erosion; D-pickup
90% WB-10% Ni-Ag Inf.	0.003	0.003	0.003	1300	Excel.	Mild even wear—both parts
60% CrB_2-30% Ni-10% Ag	0.090	0.124	0.272	1300+	N.G.	P-microscopic erosion
65% VSi_2-20% Ni-15% Ag	0.018	0.048	0.114	1300+	Poor	P-pickup from disk
60% WSi_2-20% Ni-20% Ag	0.716	-	-	-	N.G.	D-excessive pickup
60% WB-20% Ni-20% Ag	0.003	0.003	0.002	1300	Excel.	Mild even wear—both parts
68% VSi_2-12% Ni-20% Ag	0.054	0.03	0.18	1300	N.G.	P-edge cracking; D-pulled out
45% VB-35% Ni-20% Ag	0.114	0.084	0.153	1500	N.G.	P-erosion; D-grooved
60% CrN-20% Ni-20% Ag	0	0.006	0.003	1300	Very good	P-mild wear

Note: pin rating N.G., no good.

Summary of Wear Data

Rectangular Pins Against a Ni-Cr-Fe Alloy

Pin material	Pin wear (in hr) Ambient temp. 75 F	600 F	1250 F	Temp. of pin (°F)	Pin rating	Remarks: P-pin, D-disk
Refractory Hard Metals						
Pure MoB	0.045	0.072	0.072	1800	Poor	P-fractured
Pure CrB_2	0.096	-	-	-	N.G.	P-fractured
Pure WSi_2	-	-	0.024	1350+	Fair	P-scored
Pure VB_2	0.024	0.024	0.018	-	Fair	P-pickup; D-OK
Pure CrN	0.366	-	-	-	N.G.	P-fractured
Pure VSi_2	0.015	0.063	0.050	1330	Poor	P-pickup; D-grooved
Pure $CrSi_2$	0.41	0.132	-	-	N.G.	D-galled
Pure Ti_5Si_3	-	-	-	-	N.G.	P-fractured
Pure VN	0.018	0.006	-	1500	N.G.	P-fractured
Pure WB	0.60	-	-	-	N.G.	P-fractured
Pure TiB_2	0.003	0.027	0.066	1400+	Poor	D-seizing
Binary Alloys						
70% VSi_2-30% Ni	0.030	0.16	0.72	1800+	N.G.	P-fractured
60% MoB-40% Ni	0.072	-	-	1600	N.G.	P-fractured
60% CrB_2-40% Ni	0.216	-	-	1600	N.G.	P-excess. chattering
80% WSi_2-20% Ni	0.012	0.66	-	-	Fair	P-smeared, D-scratched
40% CrN-60% Ni	0.426	1.08	-	1500	N.G.	P and D-scored
Ternary Alloys						
60% MoB-20% Ni-20% Ag	0.066	0.078	0.054	1300	Poor	P and D-pitted
90% WB-10% Ni-Ag Inf.	0.009	0.024	0.006	1400	Good	P-slightly scored
60% CrB_2-30% Ni-10% Ag	0.864	3.85	-	1800+	N.G.	P-erodes; D-grooved
65% VSi_2-20% Ni-15% Ag	0.672	-	-	1800+	N.G.	P-fractured, D-galled
60% WSi_2-20% Ni-20% Ag	0.018	0.390	-	-	N.G.	P-scored
60% WB-20% Ni-20% Ag	0.090	0.276	3.77	1600	N.G.	P and D-scored
68% VSi_2-12% Ni-20% Ag	0.018	0.078	0.072	1300	Fair	P-even wear, D-seizing
45% VB-35% Ni-20% Ag	0.048	0.120	0.054	1500+	Poor	P-mild scoring, D-pitted
60% CrN-20% Ni-20% Ag	0.025	0.053	0.018	1400	Fair	P-edge fractured; D-pickup
60% CrN-30% Ni-10% Ag	0.048	0.024	0.006	1500+	Good	P-mild even wear

Note: pin rating N.G., no good.

FRICTION AND WEAR TEST DATA - GROUP C

Bearing Tests (Rolling Element) Performed in Liquid Oxygen, Liquid Hydrogen, Nitrogen Tetroxide.

Reference 107: "Lubrication of Bearings with Rocket Propellants" by M. F. Butner and J. C. Rosenberg, Lubrication Engineering, Vol. 18, No. I, p. 17 (January 1962).

Types of Bearings Tested

TABLE 2.—Bearing Design Information

BRG. NO.	BRG. SIZE	BRG. TYPE (a)	CAGE TYPE*	RINGS AND BALL MATERIAL	CAGE MATERIAL	NO. OF BALLS AND SIZE	OUTER RACE CURV.	INNER RACE CURV.	CONTACT ANGLE, DEG.	DIAM. CLEAR. INCHES	AXIAL END-PLAY, INCHES
1.	109	AC	OLR	440-C	M-2	15-11/32	0.53	0.52	25	0.0030- 0.0034	0.0130 0.0160
2.	109	AC	OLR	440-C	Chrome-Plated M-2	15-11/32	0.53	0.52	25	0.0030- 0.0034	0.0130 0.0160
3.	109	AC	OLR	440-C	15% Fiberglass-Filled PTFE	15-11/32	0.53	0.52	25	0.0030- 0.0034	0.0130 0.0160
4.	109	AC	OLR	440-C	Glass-Supported PTFE	15-11/32	0.53	0.52	25	0.0030- 0.0034	0.0130 0.0160
5.	109	AC	OLR	440-C	Anodized 248T Aluminum	15-11/32	0.53	0.52	25	0.0030- 0.0034	0.0130 0.0160
6.	109	AC	OLR	440-C	None	15-11/32	0.53	0.52	25	0.0030- 0.0034	0.0130 0.0160
7.	109	C	OLR	440-C	MoS, Coated Phenolic	12-3/8	0.52	0.52	16	0.0012	0.0083
8.	108	C	BR	440-C	410 Cres Ribbon Steel	13-11/32	0.53	0.516	10	0.0001 0.0005	0.0055
9.	107	C	ILR	440-C	Phosphor Bronze	11-5/16	0.53	0.516	12	0.0005 0.0009	0.0060
10.	107	C	ILR	52100	Phosphor Bronze	11-5/16	0.53	0.516	12	0.0005 0.0009	0.0060
11.	207	C	OLR	M-50-Tool Steel	Navy 46N7 Nickel Alloy, Class B	9-7/16	0.53	0.516	18	0.0015 0.0020	0.0125
12.	207	C	OLR	52100	Silver-Plated Silicon-Iron-Bronze	9-7/16	0.53	0.516	18	0.0015 0.0020	0.0125
13.	207	C	BR	52100	0.010-Thick PTFE Coated Ribbon Steel	9-7/16	0.53	0.516	18	0.0015 0.0020	0.0125
14.	208	C	OLR	52100	Phenolic Linen	9-15/32	0.52	0.52	14	0.0015 0.0020	0.0090
15.	210	C	BR	52100	Ribbon Steel	11-15/32	0.53	0.516	20	0.00013 0.00057	0.0150

*OLR—Outer Land Riding
ILR—Inner Land Riding
BR—Ball Riding

(a) AC = Angular Contact
C = Conrad

(Continued)

Bearing Tests (Rolling Element) Performed in Liquid Oxygen, Liquid Hydrogen, Nitrogen Tetroxide

TABLE 3.—Results of Tests in Liquid Hydrogen

TEST REF. NO.	BRG. REF. NO.	BRG. SIZE	SPEED, RPM	DURATION	DN x 10^{-6}	THRUST, LB.	MAX. HERTZ STRESS, 10^3 psi	CAGE-BALL RUBBING SPEED, FPM	SUCCESS	FAILURE	COMMENTS
1.	15	210	6,000	8 min	0.3	0		2100	X		Ribbon Steel Cage—Light galling in cage pockets.
2.	15	210	12,000	7 min	0.6	0		4200		X	Ribbon Steel Cage—Galling in cage pockets and cage breakage.
3.	11	207	15,000	7 min	0.525	300	209	3980	X		Navy 46N7 Nickel Alloy*—Slight wear and surface deterioration in cage pockets.
4.	11	207	20,000	10 sec	0.7	300	214	5300	X		Navy 46N7 Nickel Alloy*
5.	11	207	15,000	3 min	0.525	300	209	3980	X		Navy 46N7 Nickel Alloy*
6.	11	207	20,000	20,000	0.7	300	214	5300	X		Navy 46N7 Nickel Alloy*
7.	11	207	15,000	1 min	0.525	300	209	3980	X		Navy 46N7 Nickel Alloy*
8.	11	207	20,000	2 min	0.7	300	214	5300		X	Navy 46N7 Nickel Alloy*—Rough rotation because of cage-ball galling.
9.	11	207	20,000	3 min	0.7	300	214	5300		X	Navy 46N7 Nickel Alloy*—Rough rotation because of cage-ball galling (Fig. 1).
10.	12	207	15,000	2 min	0.525	300	209	3980		X	Silver-Plated Silicon-Irone Bronze Cage, Bronze was plated out on the balls.
11.	10	107	15,000	3 min	0.525	50	145	3630	X		Phosphor Bronze Cage
12.	10	107	20,000	3 min	0.7	50	151	4850		X	Phosphor Bronze Cage—Cage material plated out on the balls.
13.	9	107	15,000	3 min	0.525	50	145	3630	X		Phosphor Bronze Cage
14.	9	107	20,000	3 min	0.7	50	151	4850		X	Phosphor Bronze Cage—Cage material plated out on the balls.
15.	14	208	15,000	8 min	0.6	100	158	4460	X		Phenolic Cage—Bearing in good condition.
16.	14	208	20,000	3 min	0.8	100	158	5950	X		Phenolic Cage
17.	14	208	20,000	10 min	0.8	100	158	5950	X		Phenolic Cage
18.	14	208	20,000	13 min	0.8	100	158	5950	X		Phenolic Cage
19.	7	109	25,000	6 min	1.125	50	124	7550	X		MoS_2-Coated Phenolic Cage.
20.	7	109	25,000	14 min	1.125	50	124	7550	X		MoS_2-Coated Phenolic Cage. MoS_2 coating worn off.
21	7	109	25,000	51 min	1.125	330	232	7550		X	MoS_2-Coated Phenolic Cage, High friction caused cage breakage (Fig. 2).
22.	4	109	25,000	9¾ hr	1.125	330	213	7750	X		Glass-Supported PTFE Cage. Some cage pocket wear, otherwise in good condition.

*Class B

(Continued)

TABLE 4.—Results of Tests in Liquid Oxygen

Test Ref. No.	Fluid	Brg. Ref. No.	Brg. Size	Speed, RPM	Duration	DN x 10^{-6}	Thrust Load, Lb.	Max Hertz Stress, 10^3 psi	Cage-Ball Rubbing Speed, FPM	Success	Failure	Comments
1	LO_2	1	109	4980	22 min	0.224	60	113	1540		X	Steel Cage, Metal removed from cage pockets and welded to race track.
2.	LO_2	2	109	4980	7 min	0.224	60	113	1540		X	Chromium Plated Steel Cage, Metal removed from cage pockets.
3.	LO_2	3	109	4980	1 hr	0.224	60	113	1540	X		
4.	LO_2	3	109	4980	8 hr	0.224	260	182	1540	X		Fiberglass-filled PTFE Cage, Bearing in excellent condition, some cage pocket wear (Fig. 3).
5.	LO_2	4	109	4980	1 hr	0.224	60	113	1540	X		
6.	LO_2	4	109	6400	9 hr	0.288	345	203	1980	X		Glass-Supported PTFE Cage, Bearing in excellent condition.
7.	LO_2	5	109	6400	6 min	0.288	50	105	1980		X	AluminumCage, Heavy cage pocket and land galling (Fig. 4).
8.	LO_2	6	109	6400	1 hr	0.288	50	105	3960		X	Full Complement Bearing, No Cage, Heavy metal removal at point of contact between balls.
9.	LO_2	7	109	6400	1 hr	0.288	60	132	1930	X		MoS_2-Coated Phenolic Cage, MoS_2 removed from land and cage pockets.
10.	LO_2	8	108	6400	3 min	0.288	440	296	1730		X	Ball Riding Steel Cage, Rapid, cage destruction.
11.	LO_2	13	207	6400	1 hr	0.288	300	209	1700	X		PTFE-Coated Ball Riding Cage, (Fig. 5) Bearing in excellent condition.

TABLE 5.—Results of Tests in Nitrogen Tetroxide

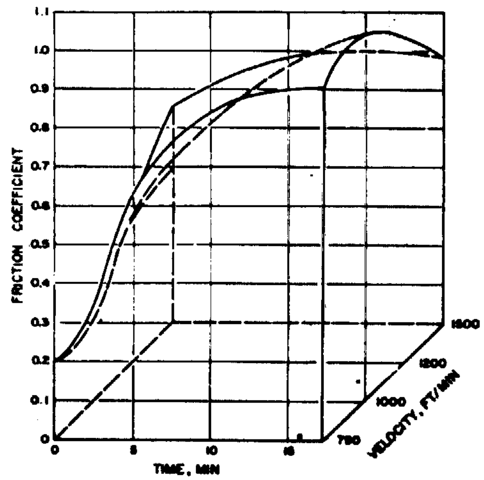
Test Ref. No.	Fluid	Brg. Ref. No.	Brg. Size	Speed, RPM	Duration	DN x 10^{-6}	Thrust Load, Lb.	Max Hertz Stress, 10^3 psi	Cage-Ball Rubbing Speed, FPM	Success	Failure	Comments
12.	N_2O_4	1	109	6400	60 min	0.288	330	198	1980		X	Steel Cage, Cage pocket wear and ball discoloration due to heating.
13.	N_2O_4	3	109	6400	156 min	0.288	330	198	1980	X		Fiberglass-Filled PTFE Cage, Bearing in excellent condition.
14.	N_2O_4	4	109	6400	280 min	0.288	330	198	1980	X		Glass-Supported Cage, Some cage bleaching, otherwise bearing in excellent condition (Fig. 6).
15.	N_2O_4	5	109	6400	60 min	0.288	330	198	1980		X	AluminumCage, Heavy cage pocket and land galling.
16.	N_2O_4	6	109	6400	60 min	0.288	330	198	1980		X	Full-Complement Bearing; No Cage, Ball discoloration due to heat.
17.	N_2O_4	8	108	6400	30 min	0.288	330	271	1730		X	Ball Riding Steel Cage, Heavy ball pocket galling.

FRICTION AND WEAR TEST DATA - GROUP C

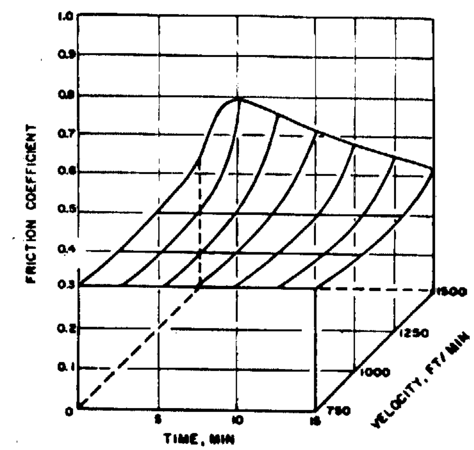
Sliding Tests Performed in Air

Reference 111: "Propellant Lubrication Properties Investigation" WADD-TR-Gl-77 Part II

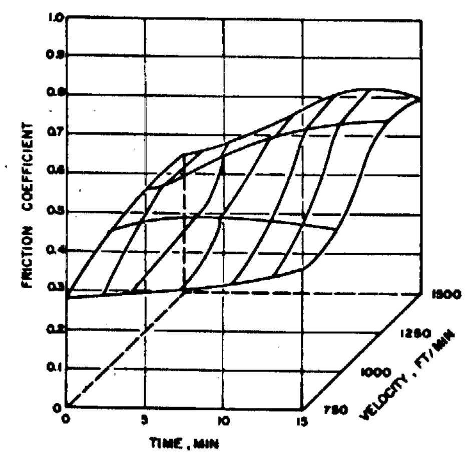
Test Equipment: Pin-On-Disk (3/8" Dia. Pin)



B25 vs. B25



440-C vs. 440-C



410N vs. 410N

Notes:

(1) B-25 - Berylco 25 (Beryllium Copper)
(2) 440 - 440-C Corrosion-resistant Steel
(3) 410N - 410 " " " , nitrided, 0.0005 to 0.010
(4) 303, 310 - 300 Series Stainless Steel

FRICTION AND WEAR TEST DATA - GROUP C

Bearing Tests (Rolling Elements) performed in RP-1, Ethelene Diamene (EDA), Unsymmetrical Dimethylhydrazine (UDMH), Hydrazine (N_2H_4), Liquid Oxygen (LOX), Nitrogen Tetroxide (N_2O_4), Inhibited Red Fuming Acid (IRFNA).

Reference 112: Propellant Lubrication Properties Investigation, WADD-TR-61-77, Part I, by M. F. Butner (Rocketdyne).

Type of Bearing Tested: Ball bearing, inner-land riding, glass-supported Teflon cage.

Speed: 24,600 RPM

Load: 330 lbs., axial.

Duration min	Total Revolutions $\times 10^6$	Wear, milligrams Cage	Balls	Inner Race	Outer Race	Comments
		Propellant: RP-1				
60	1.48	10.0	6.1	0.3	2.8	Success
60	1.48	-	-	-	-	"
-	1.48	54.7	1.1	376.0	3.8	"
		Propellant: EDA				
5	0.12	-	-	-	-	Spalling Failure
60	1.48	15.1	861.0	110.6	25.4	Frosting of balls and races in load zone
60	1.48	33.5	22.4	1.1	0.9	"
		Propellant: UDMH				
4	0.098	-	-	-	-	Spalling Failure
2.5	0.062	-	-	-	-	"
5	0.12	-	-	-	-	"
		Propellant: N_2H_4				
0.5	0.012	-	-	-	-	Frosting of balls and races in load zone.
20	0.49	-	-	-	-	Spalling Failure
		Propellant: LOX				
60	1.48	0.2061	0.5366	0.0910	0.0311	Success
60	1.48	0.1241	0.2035	0.0137	0.0084	Success
60	1.48	-	-	-	-	Success
		Propellant: NTO				
60	1.48	0.3089	0.0132	0.0036	0.0282	Success
60	1.48	0.0783	0.0809	0.0084	+0.0062	Success
		Propellant: IRFNA				
60	1.48	0.201	0.1036	0.0086	0.0856	Success
60	1.48	+0.3885	0.3579	0.5736	0.073	Success

FRICTION AND WEAR TEST DATA - GROUP C

Rolling Contact Tests Performed in 50-50 UDMH/N_2H_4

Reference 111: "Propellant Lubrication Properties Investigation", WADD-TR-61-77, Part II, by M. F. Butner and R. A. Byron (Rocketdyne).

Test Equipment: Converted Shell Four-Ball Machine (1/2" Dia. Test Balls)
Propellant Flow 0.0026 gal/min.
Maximum temperature of Test Balls: 120°F
Speed: 1800 RPM

TABLE OF MATERIAL COMBINATIONS TESTED, IN DESCENDING ORDER OF PERFORMANCE

1. Tungsten carbide (WC) (Kennametal K-96) balls and race.
2. Tungsten carbide balls, 440-C race.
3. Stellite No. 19 balls, Stellite No. 3 race.
4. 440-C balls and race.
5. Stellite No. 19 balls, 440-C race.
6. 440-C balls and races with diamond powder breakin.
7. Haynes 25 balls, 440-C races.
8. Haynes 25 balls and races.

(Continued)

ROLLING CONTACT TEST RUN SUMMARY

Test No.	Drive Ball Mat'l	Test Ball Mat'l	Test Race Mat'l	Normal Load, pounds	Test Duration, minutes	Weight Change* Drive Ball, milligrams	Weight Change* Test Balls, milligrams	Weight Change* Test Race, milligrams	Photo, Fig.	Micro Photo, Fig.	Results
1A	440	440	440	44	5	..	..	..	..	..	No surface change
1B	440	440	440	66	5	..	..	..	..	..	No surface change
1C	440	440	440	88	5	..	..	..	..	..	No surface change
1D	440	440	440	132	5	..	..	..	..	..	No surface change
1E	440	440	440	176	5	-0.2	-0.4	-0.6	..	..	Wear track evident; small "blisters"
1F	440	440	440	88	5	..	..	..	..	..	No surface change
1G	440	440	440	110	5	..	..	..	..	..	Microscopically visual wear track
1H	440	440	440	121	5	..	..	..	..	..	No change from above
1I	440	440	440	132	5	..	..	..	..	..	No change from above
1J	440	440	440	143	5	..	..	..	..	..	No change from above
1K	440	440	440	154	5	..	..	..	..	..	Wear bands distinctly visible
1L	440	440	440	154	30	0	-0.6	-0.3	..	..	
1M	440	440	440	154	5	..	..	..			Run with gaseous nitrogen blanket
1N	440	440	440	154	5	..	..	..	..	..	Run without gaseous nitrogen blanket; no change
1P	440	440	440	154	50	0	-0.3	0	29	31	Run without gaseous nitrogen blanket; distinct wear tracks
2A	WC	WC	WC	154	60	-0.3	-0.6	-0.8	32	..	No visual evidence of wear
2B	WC	WC	WC	165	5	..	..	..	..	..	No visual evidence of wear
2C	WC	WC	WC	176	5	..	..	..	..	..	No visual evidence of wear; however, three small fatigue pits appeared on the drive ball
2D	WC	WC	WC	187	5	..	..	..	..	..	No visual evidence of wear; however, three small fatigue pits appeared on the drive ball
2E	WC	WC	WC	187	5	..	..	..	..	..	Raceway spall appeared
2F	WC	WC	WC	176	20	+0.7	+0.1	0	..	33	Noisy operation; halted run
2G	WC	WC	WC	154	60	+0.7	+0.2	-0.4	..	34, 35	Drive ball had large pit. Raceway and test balls showed no surface damage.
3A	Stel	Stel	Stel	154	60	+0.6	+0.6	+0.3	36	37, 38	Small polished band on drive ball; no visual surface damage on test race. Slight noise at start, smoothing out
3B	Stel	Stel	Stel	154	60	-0.3	-0.2	-0.1	..	..	Same as above
3C	Stel	Stel	Stel	154	60	-0.4	-1.0	-0.7	..	..	Same as above
4A	H-25	H-25	H-25	154	5	-8.4	+3.0	+1.0	..	..	Heavy surface damage; run started noisy, became louder
4B	H-25	H-25	H-25	154	5	-8.3	-5.5	-1.0	..	..	Same as above
4C	H-25	H-25	H-25	154	5	-7.0	-7.0	0	39	40	Same as above

*Specimens weighed at conclusion of each test series

(Continued)

Test No.	Drive Ball Mat'l	Test Ball Mat'l	Test Race Mat'l	Normal Load, pounds	Test Duration, minutes	Weight Change* Drive Ball, milligrams	Weight Change* Test Balls, milligrams	Weight Change* Test Race, milligrams	Photo, Fig.	Micro Photo, Fig.	Results
5A	440	H-25	440	154	5	-0.3	+0.4	+2.0	..	..	Heavy surface damage. Became noisy at 3-1/2 min and grew louder
5B	440	H-25	440	154	5	-0.9	-5.3	+0.8	..	..	Same as above
5C	440	H-25	440	154	5	-1.0	-1.3	-1.5	43	44	Same as above
6A	440	WC	440	154	5	..	..	..	..	..	Polished tracks in race; no marks on WC test balls
6B	440	WC	440	154	55	-0.3	-0.2	0	..	..	Same as above
6C	440	WC	440	154	60	-2.6	-0.1	0	..	..	Same as above
6D	440	WC	440	154	60	-0.5	-0.3	-0.4	41	42	
7A	440	Stel	440	154	5						
7B	440	Stel	440	154	55	-0.6	-0.3	+1.2	45	46	Run started noisy, then quieted down. Drive ball slightly pitted. Visible wear track on race test balls show scratches
7C	440	Stel	440	154	60	0	0	0	..	..	Started noisy, quieted as run progressed. Race had surface damage.
7D	440	Stel	440	154	60	-0.6	+0.1	+0.4	..	..	Started and ran quietly. Surface damage visible on race and test balls
8A	440	440	440	22	5	..	..	..	..	..	Breakin with 6 to 10μ diamond powder. Balls and race had sand-blasted appearance
8B	440	440	440	154	5	..	..	..	..	..	Breakin with 6 to 10μ diamond powder. Balls and race had sand-blasted appearance
8C	440	440	440	154	55	-4.0	-14.6	-2.8	49	50	Race had wide solid wear track. Drive ball had deep wear groove. Test balls had polished surface. Run started noisy, became progressively smoother
8D	440	440	440	154	5	..	..	..	..	..	Breakin with 6 to 10μ diamond powder. Balls and race had sand-blasted appearance
					55	..	..	..	..	..	Race had wide solid wear track. Drive ball had deep wear groove. Test balls had polished surface. Run started noisy, became progressively smoother
8E	440	440	440	154	5	-13.0	-72.3	-2.8	47	48	Breakin with 6 to 10μ diamond powder. Balls and race had sand-blasted appearance

*Specimens weighed at conclusion of each test series

FRICTION AND WEAR TEST DATA - GROUP C[1]

Sliding Tests Performed in Dry N_2

Reference 109: "Dry Lubricated Bearings for Operation in a Vacuum" by P. H. Bowen, ASLE Trans. 5, No. 2, p. 315-326 (1962).

Test Equipment: Falex-type tester.
Velocity: 230 ft/min.
Disk Material: M-10 Tool Steel
Load: 100-300 psi

Test Materials — Plastics	86 F Avg. coef. friction	86 F Wear (mm/hr)	160 F Avg. coef. friction	160 F Wear (mm/hr)
Nylon	E*	Fail	E	Fail
Nylon—40% MoS_2 filler	.20	0.7	.20	0.7
PTFE (Polytetrafluoroethylene)	E	Fail	—	—
PTFE—mica filler	.11	3.0	.17	3.6
PTFE—glass fiber—MoS_2 filler	.03	1.5	.03	1.6
PTFE—glass cloth impregnated	.50	3.0	.45	3.5
PTFCE (Polychlorotrifluoroethylene)	.56	Fail	—	—
Polypropylene	> .60	Fail	> .60	Fail
Chlorinated polyester	> .60	Fail	> .60	Fail

Composites and alloys	Temperature (°F)	Avg. coef. friction	Wear (mm hr)
65Fe-35C	1000	.21	2.8
40Fe-60C (Heat treated)	1000	.18	3.7
30Fe-70C (Heat treated)	1000	.28	4.0
40Ni-60C (Carburized)	1000	.18	9.4[b]
30Ni-70C (Carburized)	1000	.07	6.5
56Co-30Cr-W12 + Sb_2S_3	1000	.15	2.1
Mo-V AISI 440C SS	1000	.28	1.2
AISI 304 SS	1000	.85	7.7
55 Bronze-27PTFE-18MoS_2	160	.21	1.9
70Ag-20PTFE-10WSe_2	160	.25	2.2

* Erratic values obtained.
[b] Polished area.

FRICTION AND WEAR TEST DATA - GROUP C[1]

Sliding Tests Performed in Nitrogen

Reference 108: "Lubrication of Bearings and Gears in Aerospace Environmental Facilities", by P. H. Bowen, AEDC-TDR-63-166.

Test Equipment: Modified Hohman Model A4 (Falex) wear and friction tester.
Load: 3 lb. (equivalent to 150 psi for a 2 mm scar width)
Disk Material: M-10 Tool Steel
Test Time: I hr.

Test Material	Tests at 86°F Slide Vel. ft/min	Tests at 86°F Coef. of Friction μ	Tests at 86°F Wear mm	Tests at 160°F Slide Vel. ft/min	Tests at 160°F Coef. of Friction μ	Tests at 160°F Wear mm
Nylon	460	>0.60	Failed	460	>0.60	Failed
Nylon	230	>0.60	Failed	230	>0.60	Failed
Nylon - 40% MoS_2 filler	460	0.17	0.7	460	0.17	0.7
Nylon - 40% MoS_2 filler	230	0.20	0.7	230	0.20	0.7
Nylon - 20% C filler	460	0.08	0.7	460	0.09	0.7
Nylon - 20% C filler	230	0.08	0.7	230	0.08	0.7
PTFE	460	0.12	7.2	460	0.14	Failed
PTFE	230	0.13	6.9	230	0.16	Failed
PTFE* - mica filler	460	0.14	3.2	460	0.17	4.6
PTFE - mica filler	230	0.11	3.0	230	0.17	3.6
PTFE - ceramic filler	460	0.20	1.7	460	0.14	1.8
PTFE - ceramic filler	230	0.19	1.3	230	0.21	2.0
PTFE - glass fiber and MoS_2 filler	460	0.02	1.6	460	0.03	1.5
PTFE - glass fiber and MoS_2 filler	230	0.03	1.5	230	0.03	1.6
PTFE - glass cloth filler	460	0.39	2.9	460	0.40	3.4
PTFE - glass cloth filler	230	0.50	3.0	230	0.45	3.5
PTFCE**	460	--	--	460	0.60	8.0
PTFCE	230	0.56	7.3	230	--	--
Polypropylene	460	>0.28	Failed	460	>0.24	Failed
Polypropylene	230	>0.28	Failed	230	>0.22	Failed
Chlorinated Polyether	460	>0.40	Failed	460	>0.40	Failed
Chlorinated Polyether	230	>0.40	Failed	230	>0.40	Failed
Carbon (hard) - PTFE impregnate	460	0.03	1.0	460	0.02	0.5
Carbon (hard) - PTFE impregnate	230	0.02	0.7	230	0.02	0.9
Carbon (med. soft) - PTFE impregnate	460	0.03	1.3	460	0.02	0.7
Carbon (med. soft) - PTFE impregnate	230	0.02	1.0	230	0.02	0.9

* Polytetrafluoroethylene
** Polychlorotrifluoroethylene

FRICTION AND WEAR TEST DATA - GROUP C[1]

Sliding Tests of Solid Lubricant Materials in Air

Reference 114:	"Development of Design Criteria for a Dry Film Lubricated Bearing System", by M. E. Campbell and J. W. Van Wyk, ASD-TDR-62-1057, March 1963.
Test Equipment:	Lubricant composite specimens were pressed against a rotating KL62B titanium carbide ring.
Load:	5 lbs.
Velocity:	7000 ft/min.
Time:	10 minutes
	Room temperature.

Material	Specimen Weight (Grams) Before Test	Loss	Coefficient of Friction Initial	High	Low	Remarks
90% MoS_2 + 10% (50% Zr + 50% Ni)	3.4160	0.2813	0.14	0.14	0.02	Low F_c at 4 minutes.
90% MoS_2 + 10% Cr	3.2520	0.3315	0.08	0.08	0.03	F_c erratic, gradual increase last 5 minutes.
90% MoS_2 + 10% Metco 43C	4.2784	0.2714	0.09	0.09	0.03	Gradual F_c decrease during test; pellet wear scar is very shiny.
90% MoS_2 + 10% (60% Fe + 4096 Cr)	3.9348	0.3281	0.11	0.11	0.07	F_c showed gradual decrease during run.
90% MoS_2 + 10% (80% Fe + 20% Pd)	3.7304	0.4879	0.09	0.09	0.01	F_c dropped to 0.01 at 3 minutes and remained constant.
90% MoS_2 + 10% (50% Pd + 50% Cr)	4.0051	0.1975	0.10	0.14	0.08	Low F_c at end of run.
ATJ Graphite	1.7190	0.0020	0.23	Constant		Test run at 1 pound.
ATJ Graphite	1.7347	0.0022	0.076	Constant		Test run at 3 pounds.
ATJ Graphite	1.6158	0.0029	0.11	0.40	0.11	Gradual rise of F_c during test.
60% TiO_2 + 40% Ni	5.1250	0.0748	0.14	0.165	0.14	F_c nearly constant throughout test.
AMP 10220-500-1B	1.7585	0.0041	0.064	0.122	0.064	Gradual rise of F_c during test.
48% TiO_2 + 6% NiO + 6% MoS_2 + 40% Ni	3.8616	0.1916	0.11	0.11	0.11	Fairly constant F_c throughout test.
32% TiO_2 + 4% NiO + 4% MoS_2 + 60% Ni	6.2098	0.2002	0.14	0.15	0.13	F_c erratic between high and low values.
80% MoS_2 + 20% (80% Fe + 20% Pd)	5.4294	0.1145	0.19	0.20	0.19	F_c fairly constant.
40% MoS_2 + 60% (80% Fe + 20% Pd)	6.1725	0.2369	0.23	0.30	0.20	Chattered during first 2 minutes of test.
60% MoS_2 + 40% (80% Fe + 20% Pt)	5.8777	0.4142	0.38	0.38	0.23	F_c erratic.
40% MoS_2 + 60% (80% Fe + 20% Pt)	5.8194	0.2830	0.28	0.28	0.20	Gradual decrease of F_c during test.
No. 8 White Solid Lubricant with Ceramic Binder.	2.8366	0.1593	0.26	0.33	0.23	Tended to chatter and developed considerable heat.
40% MoS_2 + 60% (60% Fe + 40% Cu)	5.1056	0.1393	0.31	0.31	0.11	Tended to chatter and erratic F_c.
60% TiO_2 + 20% NiO + 20% Ni	3.6684	0.5057	0.22	0.26	0.19	Shoulder chipped off.
45% TiO_2 + 15% NiO + 40% Ni	4.3954	0.0998	0.15	Nearly Constant		Corners chipped.
ATJ Graphite + B_2O_3	1.5364	0.0038	0.04	Constant		Little difference between treated and untreated

(Continued)

FRICTION AND WEAR TEST DATA - GROUP C[1]

Sliding Tests of Solid Lubricant Materials in Air (Continued)

Material	Specimen Weight (Grams) Before Test	Loss	Coefficient of Friction Initial	High	Low	Remarks
ATJ Graphite Not Treated	Not Weighed		0.09	Constant		
AMP P/N 10220-500-1A(Heat Treated)	1.7476	0.0079	0.10	0.10	0.08	F_c nearly constant.
MY 3F	2.7164	0.0027	0.06	0.06	0.055	Constant at 0.055 after 1 minute.
MY 9D	2.4565	0.0039	0.05	0.05	0.05	Some vibration.
Pyro. No. 1	2.1019	0.0069	0.09	0.09	0.05	F_c constant at 0.05 after 2 minutes.
Pyro. No. 2	2.1990	0.0128	0.14	0.18	0.10	F_c fluctuated, showed general rising trend.
EY 9106	1.5845	0.0048	0.09	0.11	0.08	
$MoSi_2$ + Graphite	1.7519	0.0067	0.07	0.14	0.07	Gradual rise in F_c; considerable vibration; pellet contained lumps or crystals of hard abrasive material.
TiB_2 + Graphite	1.6129	0.0033	0.13	0.13	0.08	Considerable vibration; smooth wear track.
90% MoS_2 + 10% Metco 43C	3.3635	0.0211	0.11	0.13	0.11	Vibration and tended to grab
80% TiO_2 + 20% (60% Fe-40% Cu)	3.8778	0.0104	0.13	Constant, ± 0.01		Some vibration; F_c somewhat erratic.
20% TiO_2 + 80%(60% Fe-40% Cu)	2.0688	0.0177	0.12	0.14	0.12	F_c showed slight gradual increase.
10% MoS_2 + 90% K-163B	4.2780	0.0476	0.14	Constant		Vibration during first half of test.
90% MoS_2 + 10% J-162B	3.2108	0.1760	0.15	0.15	0.08	Gradual decrease in F_c.
80% MoS_2 + 20% K-162B	2.8551	0.0773	0.17	0.17	0.11	Gradual decrease in F_c.
70% MoS_2 + 30% K-162B	3.9504	0.0439	0.09	Constant		
20% MoS_2 + 80% K-162B	4.5250	0.1915	0.25	0.50	0.16	Chattered during first two minutes of test.
Unknown	2.5060	0.1079	0.18	0.18	0.13	Fairly constant F_c after 1 minute at 0.1
24% TiO_2 + 18% MoS_2 + 40% Ni	4.1258	0.5073	0.15	0.22	0.15	F_c showed gradual increase.
16% TiO_2 + 12% NiO + 12% MoS_2 + 60%Ni	5.6201	0.2180	0.17	0.17	0.108	Low F_c at 3 min., gradual increase to $F_c \cong 0.16$ at 10 minutes.
64% TiO_2 + 8% NiO + 8% MoS_2 + 20% Ni[2]	4.7666	0.2302	0.13	0.26	0.13	F_c gradual rise during test.
20% MoS_2 + 80% (60% Fe + 40% Cu)	4.7867	0.0501	0.14	0.17	0.14	F_c nearly constant throughout run.
75% MoS_2 + 25% (80% Fe + 20% Pd)	3.3836	0.2821	0.11	0.33	0.11	Considerable chatter. F_c rose gradually; unsatisfactory.
75% MoS_2 + 25% (80% Fe + 20% Pd)	3.5470	0.0421	0.08	0.08	0.05	F_c constant.
80% MoS_2 + 20% (20% Pd + 80% Fe)	4.4470	0.5464	0.09	0.28	0.09	Mirror finish on pellet; gradual increase in F_c (F_c = 0.05 at 1800 rpm).
70% MoS_2 + 30% (65% Fe + 35% Mo)	1.2459	0.0172	0.12	0.116	0.056	
40% MoS_2 + 40% TiO_2 + 20% Ni	1.8963	0.2772	0.16	0.25	0.16	F_c showed steady, gradual increase.
90% MoS_2 + 10% (20% Cr + 80% Ni)	3.4830	0.3068	0.17	Constant over full 10 minutes.		High wear.

FRICTION AND WEAR TEST DATA - GROUP C[1]

Sliding Tests of Lead-Lubricated Steel in Dry Gaseous Helium

Reference 110: "Lubrication Research and Test Method Development For Aerospace Propulsion Systems", by B. B. Baber, R. A. Burton, F. Chang, P. M. Ku, J. A. Russell, Southwest Research Institute Report No. RS-380, July 1963.

Test Equipment: Pin-on-disk
Disk Material: 52100 Tool steel, lead plated
Pin Material: 52100 Tool steel

Test No.	Load, gm	Temp. °F	f_s	f_k	Track Width, mm	Track Depth, mm × 10^3	Film Thickness, A
1	200	75	0.80	0.80	-	1.60	16,000
		-70	1.00	0.95	-	1.60	16,000
		-320	1.00	0.70	0.60	1.60	16,000
2	50	75	1.24	1.00	-	1.60	16,000
		-70	1.38	1.10	-	1.60	16,000
		-320	1.38	1.10	0.35	1.60	16,000
3	200	-75	0.95	0.95	0.25	-	30,000
		-70	1.15	1.10	0.20	3.00	30,000
		-320	1.10	0.90	0.20	-	30,000
4	208	75	1.25	1.25	0.35	4.00	40,000
		-70	1.15	1.10	0.35	4.00	40,000
		-320	1.10	1.05	0.35	4.00	40,000
5	50	75	1.30	1.16	0.15	1.60	16,000
	100	75	1.20	1.10	0.28	1.60	16,000
	1,000	75	0.66	0.60	0.50	1.60	16,000
6	50	-320	1.14	1.14	0.13	0.20	20,000
	100	-320	1.00	0.90	0.20	0.40	20,000
	1,000	-320	0.77	0.64	0.25	2.00	20,000
7	50	75	1.00	1.00	0.10	1.20	20,000
	100	75	1.20	1.10	0.20	1.20	20,000
	1,000	75	0.75	0.73	0.90	2.00	20,000
8	50	-320	1.14	1.00	0.15	1.20	12,000
	100	-320	1.00	0.90	0.15	1.20	12,000
	1,000	-320	0.79	0.70	0.25	1.20	12,000
9	50	-320	1.20	1.00	0.13	1.40	20,000
	100	-320	1.20	1.00	0.15	1.60	20,000
	1,000	-320	0.80	0.77	0.25	2.00	20,000
10	200	75	0.44	0.44	0.40	0.80	8,000
		-70	0.50	0.47	0.38	0.80	8,000
		-320	0.40	0.37	0.55	0.80	8,000
11	200	75	0.45	0.43	0.40	0.80	8,000
		-70	0.49	0.47	0.35	0.40	8,000
		-320	0.48	0.45	0.45	0.60	8,000
12	200	75	0.63	0.60	0.35	0.80	8,000
		-70	0.73	0.70	0.45	0.80	8,000
		-320	0.50	0.47	0.40	0.80	8,000
13	50	75	1.00	0.90	0.10	0.40	4,000
		-70	1.10	0.96	0.15	0.40	4,000
		-320	1.00	0.90	0.15	0.40	4,000
14	1,000	75	0.22	0.22	0.25	0.20	2,000
		-70	0.35	0.33	0.33	0.20	2,000
		-320	0.48	0.44	0.33	0.20	2,000
15	50	75	1.00	1.00	0.25	0.40	4,000
	100	75	0.77	0.77	0.30	0.40	4,000
	1,000	75	0.23	0.27	0.43	0.40	4,000
16	50	-320	1.00	0.95	0.08	0.40	4,000
	100	-320	0.77	0.75	-	-	4,000
	1,000	-320	0.60	0.57	0.50	0.40	4,000
17	50	-320	0.80	0.80	0.10	0.40	4,000
	100	-320	0.80	0.70	0.15	0.40	4,000
	1,000	-320	0.52	0.50	0.20	0.40	4,000

FRICTION AND WEAR TEST DATA - GROUP C[1]

Sliding Test Performed in Helium and Air

Reference 110: "Lubrication Research and Test Method Development For Aerospace Propulsion Systems", by B. B. Baber, R. A. Burton, F. Chang, P. M. Ku, J. A. Russell, Southwest Research Institute Report No. RS-380, July 1963.

Test Equipment: Pin-on-Disk
Velocity: 7.2 ft/min.
Room Temperature: (Copper specimens pre-oxidized at 400°F in air)
Disk Material: Copper, oxidized Pin Material: Copper, oxidized

Test No.	Oxide Thickness, A	% Cu_2O	% CuO	μ_s	μ_k	μ_s/μ_k	Test Atmosphere
2	2400	87	13	1.26	1.10	1.15	He
3	2200	60	40	1.26	1.18	1.07	He
4	2350	100	0	1.03	0.93	1.11	He
5	2350	100	0	0.82	0.73	1.12	He
6	1900	100	0	1.08	0.98	1.10	He
7	2500	80	20	1.24	1.17	1.06	He
8	2900	78	22	1.19	1.09	1.09	He
9	2650	75	25	1.40	1.30	1.08	He
10	2300	81	19	1.19	1.11	1.07	He
11	1900	81	19	1.21	0.99	1.22	He
13	2500	92	8	1.55	1.45	1.07	He
14a	3000	67	33	1.30	1.21	1.07	He
14b	3000	67	33	1.19	1.09	1.09	Air
15a	1800	100	0	1.51	1.42	1.06	He
15b	1800	100	0	1.04	0.93	1.12	Air
16a	3300	51	49	1.39	1.31	1.06	He
	3300	51	49	1.91	1.84	1.04	He
16b	3300	51	49	0.50	0.47	1.06	Air
17a	2800	68	32	0.71	0.66	1.08	He
	2800	68	32	1.04	0.94	1.11	He
17b	2800	68	32	0.86	0.80	1.08	Air
18a	3500	83	17	0.88	0.81	1.09	He
	3500	83	17	1.09	1.03	1.06	He
18b	3500	83	17	0.82	0.75	1.09	Air
19a	3400	48	52	1.21	1.11	1.09	He
	3400	48	52	1.47	1.36	1.08	He
19b	3400	48	52	0.74	0.66	1.12	Air
19c	3400	48	52	0.91	0.85	1.07	He
20a	4000	81	19	0.76	0.66	1.15	He
	4000	81	19	0.92	0.87	1.06	He
20b	4000	81	19	0.63	0.58	1.07	Air
20c	4000	81	19	0.92	0.86	1.07	He
21a	3500	55	45	1.54	1.44	1.07	He
	3500	55	45	1.97	1.92	1.03	He
21b	3500	55	45	0.63	0.57	1.11	Air
21c	3500	55	45	0.90	0.85	1.06	He

FRICTION AND WEAR TEST DATA - GROUP D

Bearing tests (ball bearings in vacuum (10^{-7} torr.)

Reference 106: "Bearings for Vacuum Operation - Retainer Material and Design"

By: H. E. Evans and T. W. Flatley
ASME Trans. Vol. 85, No.2,
May 1963.

Speed: 10,000 rpm (size R2-5 Brgs.)

Load: None

Room Temperature:

Test No.	Retainer (Machined)	Bearing Materials: Balls	Bearing Materials: Races	Test Duration (Hours)
1	Tool Steel Circle C Silver Plated	Gold Plated A ##	Gold Plated A	1049 #
2	"	" A	" A	1660
3	"	" B ###	" B	332
4	"	" B	" A	136
5	"	" B	" A	33
6	S - Inconel	" A	" A	873
7	"	" A	" A	639
8	S - Inconel	" B	" B	178
9	"	" B	" B	148
10	Silnic Bronze Gold Plated	" A	" A	369
11	"	" A	" A	279
12	"	" B	" B	39
13	"	" B	" B	32
14	Silnic Bronze	" A	" A	159
15	"	" A	" A	80
16	"	" B	" B	156
17	2.5% Beryllium Copper	" A	" A	329
18		" A	" A	12
19	"	" B	" B	71
20	"	" B	" B	4
21	"	" B	" A	68
22	"	" B	" A	6

\# Removed before failure.
\## Plating A; 23+ carat gold with additives (nickel, indium, cobalt), to improve adherence hardness and brightness.
\### Plating B: 24 carat gold.

FRICTION AND WEAR TEST DATA - GROUP D[1]

Sliding Tests Performed in Vacuum

Reference 112: "Lubrication of Bearings and Gears in Aerospace Environmental Facilities", by P. H. Bowen, AEDC-TDR-63-166.
Test Equipment: Pin-on-disk, (1/8" Dia. Pin) 0.012 sq.in. rubbing area)
Velocity = 52 ft/min.
Room Temperature
Disk Material: 440-C Stainless Steel

Identification	RH Hardness	Slider Material	Vacuum Press. torr	Coefficient of Friction At 4 Slider loads (gms): 1000	2000	3000	4000	Wear (gms) at At 2 Slider loads/gm: 1-3000/gm for 45 min.	4000/gm 60 min.
A	75	(X)* Ag-(1) *PTFE-WSe_2	6×10^{-8}	.10	.09	.10	.12	0.040	0.071
B	37	(Y) Ag-(1) PTFE-WSe_2	1×10^{-7}	.16	.17	.16	.18	0.016	0.066
C	73	(XL) Ag-(3) PTFE-WSe_2	1×10^{-7}	.14	.15	.17	.17	0.014	0.040
D	73	(XL) Ag-(3) PTFE-WSe_2**	1×10^{-7}	.20	.14	.15	---	0.124	---
E	--	Sintered Ag-PTFE-MoS_2	2×10^{-7}	.28	.34	.33	.28	0.073	0.264
F	64	Cu-Ag-PTFE-USe_2	5×10^{-7}	.29	.33	.34	.29	0.008	0.046
G	36	(Z) Ag-(3) PTFE-WSe_2	2×10^{-7}	.26	.27	.25	.20	0.004	0.038
H	72	Cu-Bronze-(3) PTFE-WSe_2	3×10^{-7}	.43	.32	.32	.31	0.004	0.016
J	73	(X) Ag-(3) PTFE-WSe_2***	4×10^{-7}	.18	.18	.19	.19	0.014	0.055
K	--	PTFE-WSe_2 (7%)	6×10^{-7}	.20	.21	.20	.20	0.017	0.054
L	--	-WSe_2 (3%)	6×10^{-7}	.04	.05	.05	.06	<0.001	<0.001
M	--	-graphite	1×10^{-7}	---	.13	.18	.21	0.005	---

* ( ) PTFE or ( ) Ag Various Particle Size
** Rider was Ga coated
*** Rider was nitrited

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CHAPTER VIII

Analysis of Seal Categories

A. J. Bialous

Guide to Tabular Data

The system which has been devised for the orderly entry of data selected from the literature into the tables is shown in Table A-1 (Seal Categorization System). It serves as a numerical key for both the entry and retrieval of the tabulated data.

This manner of data presentation is considered to be "user" oriented rather than the "seal-type" orientation as is normally done. The user-oriented approach permits a search for a sealing method to be concentrated on the sealing category of interest and ignores other capabilities of a particular sealing device. For example: In the case where information is desired regarding useful methods for fluid sealing of a reciprocating mechanism, Table A-1 indicates that it will be found in Section I.C. of the data tables. Now, if the sealing mechanism of interest is a ring-seal, the section can be pin-pointed to I.C.1.a.(4). It makes it unnecessary for a designer to review all of the dynamic sealing devices and methods and allows him to concentrate directly on the actual design.

The categorization number for each data table is given in the upper right corner. Seal types listed and definitions used, conform with those given in the above mentioned reports N63-19596 and N63-19597.

SEAL CATEGORIZATION SYSTEM

I. DYNAMIC SEALS

A. Rotary Motion
1. Selection & Control of Geometry
   a. Positive Contact
      (1) Face
      (2) Packings
      (3) Lip
      (4) Rings
   b. Controlled Clearance
      (1) Labyrinth
      (2) Bushing
2. Control of Fluid Properties
   a. Freeze
   b. Magnetic
3. Control of Fluid Forces
   a. Centrifugal
   b. Screw
   c. Magnetic

B. Oscillatory Motion
1. Selection & Control of Geometry
   a. Positive Contact
      (1) Face
      (2) Packings
      (3) Lip
      (4) Rings
      (5) Diaphragms
   b. Controlled Clearance
      (1) Labyrinth
      (2) Bushing
2. Control of Fluid Properties
   a. Freeze
   b. Magnetic
3. Control of Fluid Forces

C. Reciprocating Motion
1. Selection & Control of Geometry
   a. Positive Contact
      (1) Diaphragm
      (2) Packings
      (3) Lip
      (4) Rings
   b. Controlled Clearance
      (1) Labyrinth
      (2) Bushing
2. Control of Fluid Properties
   a. Freeze
   b. Magnetic
3. Control of Fluid Forces

II. STATIC SEALS

A. Selection & Control of Geometry
1. Positive Contact
   a. Elastic Deformation
      (1) Gaskets
      (2) Rings
      (3) Diaphragms
   b. Plastic Deformation
      (1) Gaskets
      (2) Rings
      (3) Packing
   c. Hermetic Sealing
      (1) Welding
      (2) Brazing
      (3) Soldering
      (4) Cementing
      (5) Mold in place seals (metal injection etc.,) luting, caulking, depositing, plating
   d. Other - Lapped
2. Controlled Clearance
   (1) Labyrinth
   (2) Bushing

B. Control of Fluid Properties
1. Freeze
2. Magnetic

C. Control of Fluid Forces

TABLE A-1

Description of Recommended General References

A number of the literature references reviewed contained useful information on the subject of fluid sealing but did not comply with the requirements for tabular data which have been described earlier. These references are identified in this report as "General References." They are located among the data tables, in the areas which pertain to their specific sealing category as determined from chart in Table A-1. The degree of generalized coverage varies from the most general case as for example, "Sealing Materials" to a much narrower general category such as "Dynamic Seals - Rotary Motion - Ring Type". [Category I.A.1.a.(4)] Information categorized under the latter heading could be a general discussion on the subject of sealing rings as rotary motion seals with no information regarding any specific performance data.

Recommended references which apply to sealing in general are presented in section 9 just before the tabulated data.

Tabulated Data

DYNAMIC SEALS	- ROTARY MOTION	- Face Type	- I.A.1.a.(1)
DYNAMIC SEALS	- ROTARY MOTION	- Packing Type	- I.A.1.a.(2)
DYNAMIC SEALS	- ROTARY MOTION	- Lip Type	- I.A.1.a.(3)
DYNAMIC SEALS	- ROTARY MOTION	- Ring Type	- I.A.1.a.(4)
DYNAMIC SEALS	- ROTARY MOTION	- Labyrinth Type	- I.A.1.b.(1)
DYNAMIC SEALS	- ROTARY MOTION	- Bushing Type	- I.A.1.b.(2)
DYNAMIC SEALS	- ROTARY MOTION	- Centrifugal Type	- I.A.3.a.
DYNAMIC SEALS	- OSCILLATORY MOTION	- Ring Type	- I.B.1.a.(4)
DYNAMIC SEALS	- RECIPROCATING MOTION	- Diaphragm Type	- I.C.1.a.(1)
DYNAMIC SEALS	- RECIPROCATING MOTION	- Packing Type	- I.C.1.a.(2)
DYNAMIC SEALS	- RECIPROCATING MOTION	- Lip Type	- I.C.1.a.(3)
DYNAMIC SEALS	- RECIPROCATING MOTION	- Ring Type	- I.C.1.a.(4)
STATIC SEALS	- ELASTIC DEFORMATION	- Gasket Type	- II.A.1.a.(1)
STATIC SEALS	- ELASTIC DEFORMATION	- Ring Type	- II.A.1.a.(2)
STATIC SEALS	- ELASTIC DEFORMATION	- Diaphragm Type	- II.A.1.a.(3)
STATIC SEALS	- PLASTIC DEFORMATION	- Gasket Type	- II.A.1.b.(1)
STATIC SEALS	- PLASTIC DEFORMATION	- Ring Type	- II.A.1.b.(2)
STATIC SEALS	- PLASTIC DEFORMATION	- Packing Type	- II.A.1.b.(3)
STATIC SEALS	- HERMETIC SEALING	- Welded Type	- II.A.1.c.(1)
STATIC SEALS	- HERMETIC SEALING	- Brazed Type	- II.A.1.c.(2)
STATIC SEALS	- HERMETIC SEALING	- Soldered Type	- II.A.1.c.(3)
STATIC SEALS	- HERMETIC SEALING	- Cemented Type	- II.A.1.c.(4)
STATIC SEALS	- HERMETIC SEALING	- Molded Type	- II.A.1.c.(5)
STATIC SEALS	- HERMETIC SEALING	- Other Types	- II.A.1.d.

LIST OF RECOMMENDED GENERAL REFERENCES

SEALING IN GENERAL

GENERAL REFERENCES

SEALING IN GENERAL

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TABULATED DATA

I

DYNAMIC SEALS

GENERAL REFERENCES

DYNAMIC SEALS

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I.A.

DYNAMIC SEAL

ROTARY MOTION

GENERAL REFERENCES

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CATEGORY I.A

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DYNAMIC SEALS -- ROTARY MOTION

CATEGORY -- I.A.1.a.(1)

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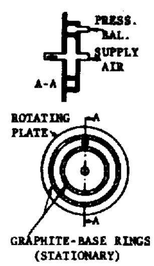
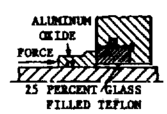
SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTATING
SEAL TYPE --- FACE

REFERENCE NO.	B.L. Johnson June 1958 AD201739	A.N. Curren et al Jan. 1957 AD119437	H.O. Pedersen et al Feb. 1959 AD225213 →			
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE			61-64	62-68	44-91	40-153
PRESSURE PSIA MAX. RANGE	600	38.3	95-155	15	25-155	15-25
VELOCITY FPM	1700 to 2100 RPM	10,000	9000-15000 RPM →		9000-18000 RPM →	
MATERIAL SEAL	Carbon Sleeve Stellite Face with Synthetic Rubber "O"-Rings	Carbon-Base Rings	Upstream Inlet Seal Glass filled teflon; SS 321 →	Downstream Outlet Seal	Upstream Inlet Seal	Downstream Outlet Seal
MATE	Steel	Chromium Plated Metal	SS 329 →			
FLUID	Military Gasoline	Air	90% Hydrogen Peroxide →			
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm. cc/sec)	No leakage after run-in	Zero	8-10 cc/min.	Negligible	5-80 cc/min.	Negligible to 76 cc/min.
TEST DURATION	1000 Hrs.		13 1/2 Minutes →		131 1/3 Minutes →	
LIFE	More than 1000 Hrs.	Wear .0005"/Hr. (Ring)				
REMARKS	Pump Shaft Sleeve Type Seal. Mfg. by J. Crane Co.		Test time accumulated in test runs of 3 minutes (max.). → See report for information regarding tests with catalyst bed temperatures of 1300°F adjacent to the seals and bearings being tested.			

DESIGN





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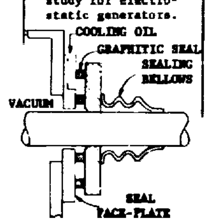
I.A.1.a.(1)
Chart 1

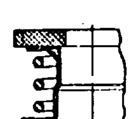
SEAL ANALYSIS

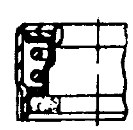
DYNAMIC SEAL
APPLICATION -- ROTATING
SEAL TYPE --- FACE

REFERENCE NO.	A.S. Damholm et al Nov. 1961 Supp. 013	M.M. Sackman et al March 1957 AD156901	P.A. Scott Sept. 1960 S-039 →		R.S. Rainey May 1950 R-006 →	
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE		-67 to +185	480	480-530	300	400
PRESSURE PSIA MAX. RANGE	14.7 (Vacuum System)	0 to 60 (Internal Pressure)	1065	1065-1090		
VELOCITY FPM	10,000 RPM	2100 RPM	1800 RPM	1800 to 2200 RPM	4000-5000 and up →	
MATERIAL SEAL	Graphitic	National Carbon Co. (Karbate #22 Tube Material)	Rotating Face: 30% ± 1% Chromium; 12% ± 1% Tungsten; 2.2% ± 0.5% Carbon; balance is Cobalt.		Synthetic Material →	
MATE		Stainless Steel 303*	Stationary Face: 25-30% Lead and Aluminum; 5% Calcium, Sodium, Copper, Antimony, Iron, Silicone and Tin; balance is Carbon with traces of other metals. Impregnated with a phenolic resin.			
FLUID	Air and Oil	Nitrogen	Heavy Water (D_2O) →			
SURFACE FINISH (μ in. CLA)		*Lapped to 5-10 RMS				
LEAK RATE (Atm. cc/sec)	Vacuum held 2 x 10^{-5} to 10^{-7} Torr	.05 cu.ft. per Hr.	0.001 to 0.05 gal./hr.	0.001 to 0.3 gal./hr.		
TEST DURATION	65 Hrs.		1579 Hrs.	3685 Hrs.		
LIFE		Approx. 10,000 Hrs. +				
REMARKS	Work is in connection with a feasibility and design study for electrostatic generators.	This is a waveguide rotary joint. Max. temp. rise = 30°F. Running torque equals 40 oz.-in. or less. Lubricants: Lehigh Chem. Co. #L-793; Signal Corp. PD-603. Static sealing portions were "Buna N" O-Rings (Shore A = 40-60).	Thin film of the fluid is used as a lubricant between the faces of the seal.	First 2000 hrs. was run with Helium saturated in the water.	Boot-type seal. Has long service life. Used in water pumps and other types of pumps where friction must be an absolute minimum.	Case-type Seal. Has long service life. Used in similar applications as boot-type seal, but the seal is entirely unitary and must be pressed into position.

DESIGN







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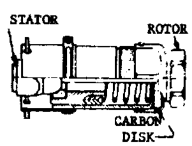
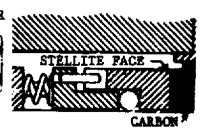
I.A.1.a.(1)
Chart 2

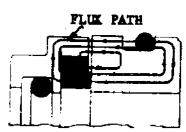
SEAL ANALYSIS

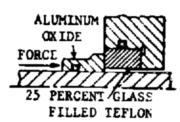
DYNAMIC SEAL
APPLICATION -- ROTATING
SEAL TYPE --- FACE

REFERENCE / NO.	Anonymous Dec. 1950 Supp. 006	A.L. Decker Sept. 1957 D-015	S. Elonka Dec. 1958 E-022	H.O. Pedersen et al Feb. 1959 AD225213 →	
INFORMATION BASED ON TESTS / RECOMMENDATION	X	X	X	In fixture →	
TEMPERATURE °F MAX.			550		
RANGE	-40 to 450			47 to 91	40 to 140
PRESSURE PSIA MAX.	165 for steam. 315 for hydraulic oil and chemicals.	315	5000		
RANGE				25 to 160	15 to 25
VELOCITY FPM	Up to 1000 RPM	500	20,000	Approx. 16,000 RPM →	
MATERIAL SEAL	Non-porous carbon disk	Carbon	A suitable magnetic ring (usually 400 series St. Stl.) & carbon ring.	Aluminum oxide →	
MATE	Rigid pipe balls	Stellite Face	Alnico-5 Ring	25% glass filled teflon →	
FLUID	Steam, Air, Oil, Gas or Water	Gasoline		90% H_2O_2 one side N_2 on other	90% H_2O_2 one side Air on other
SURFACE FINISH (μ in. CLA)					
LEAK RATE (Atm. cc/sec)				Inlet seal 0.08 to 1.33	Outlet seal 0.075 to 1.26
TEST DURATION				3 to 6 min. runs, 120 min. total. →	
LIFE		15,000 hrs.			
REMARKS	Rotating joint. Joints come in 1/2", 3/4" and 1" pipe sizes. (Rotherm Engineering Co., Inc.)		Magnetic seal for shafts up to 14" dia.	Rocket engine, turbo-pump, fluid cooled bearing seal remained excellent but recommend KEL "F" O-rings be replaced with metal.	
DESIGN					









SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTATING
SEAL TYPE --- FACE

REFERENCE / NO.	H.E. Tilbe February 1958 T028			
INFORMATION BASED ON TESTS / RECOMMENDATION	X	X	X	X
TEMPERATURE °F MAX. / RANGE	140	140	140	140
PRESSURE MAX. (at pump) / RANGE	1000 PSIG	1050 PSIG	400 PSIG	1050 PSIG
VELOCITY	1760 RPM →			
MATERIAL SEAL	Stellite →			
MATE	Graphitar →			
FLUID	De-ionized water →			
SURFACE FINISH (μ in. CLA)				
LEAK RATE (Atm. cc/sec)	Nil	1.5 cc/min	0.3 cc/min	3 cc/min
TEST DURATION / LIFE	720 Hours	423 Hours	104 Hours	1053 Hours
REMARKS	Pump temperature - 500°F →			

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTARY
SEAL TYPE --- FACE

REFERENCE	D. Summers - Smith April 1961				
NO.	S132				
INFORMATION BASED ON TESTS / RECOMMENDATION	X	X	X	X	X
TEMPERATURE MAX. RANGE	20°C →				
PRESSURE PSI MAX. RANGE	16 PSIA (Fluid Pressure) →				
VELOCITY FPM	500 RPM →				
MATERIAL					
SEAL	Metal filled Carbon Graphite →				
MATE	Ni-Resist →				
FLUID	Water →				
SURFACE FINISH (μ in. CLA)	2 - 6 →				
LEAK RATE (Atm. cc/sec)	7400 ml/hr	500 ml/hr	125 ml/hr	50 ml/hr	0 ml/hr
TEST DURATION LIFE					
REMARKS (Contact Pressure)	4 PSIG Seal area - 0.953 sq. in. →	9 PSIG	11 PSIG	19 PSIG	32 PSIG

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTARY
SEAL TYPE --- FACE

REFERENCE	Marsh, H. G. April 1962				
NO.	M014				
INFORMATION BASED ON TESTS / RECOMMENDATION	X	X	X	X	X
TEMPERATURE °F MAX.	40°C (Oil) →				
PRESSURE PSIG RANGE Oil	6	16	36	60	76
PRESSURE PSIG RANGE Gas	0	10	30	45	60
VELOCITY	9600 RPM →				
MATERIAL					
SEAL	(See information list under Remarks)				
MATE					
FLUID	Turbo "27" and Air				
SURFACE FINISH (μ in. CLA)					
LEAK RATE (Atm. cc/sec)	Complete gas sealing with oil down to one PSI above gas pressure				
TEST DURATION LIFE	Continuous service for two years				
REMARKS	Metering holes, 1/16 inch dia. Shaft dia. 4 inches In this type of seal the clearance is maintained by means of a fixed pad thrust bearing concentric with a sealing land and running against the same plane rotor.				

I.A.1.a.(1)
Chart 7

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTARY
SEAL TYPE --- FACE

REFERENCE	Marsh, M. G. April 1962			
NO.	M014			
INFORMATION BASED ON TESTS	X	X	X	X
RECOMMENDATION				
TEMPERATURE MAX.	50^oC	200^oC	200^oC	415^oC
PRESSURE RANGE	40 inches of water →			
VELOCITY	3000 RPM →			
MATERIAL SEAL	Carbon →			
MATE	Carbon →			
FLUID	Nitrogen →			
SURFACE FINISH (μ in. CLA)				
LEAK RATE cu ft/Hr @ STP	1.48	0.75	0.89	0.20
TEST DURATION LIFE	Operation at stated conditions for over one year →			
REMARKS	Air lubricating pressure, 7 to 15 PSIG. Clearance of sealing land 0.0002 to 0.0006 inches. Flatness within 0.0001 inch on 6 inch diameter. Seal #1 - 20^o to 200^oC; Seal #2 - 170^o to 415^oC. Data from Zenith Reactor Application.			

I.A.1.a.(1)
Chart 8

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTARY
SEAL TYPE --- FACE

REFERENCE	Dabney, M. J. January 1951
NO.	D001
INFORMATION BASED ON TESTS	X
RECOMMENDATION	
TEMPERATURE ^oF MAX. RANGE	Ambient
PRESSURE PSI MAX. RANGE	850 - 1000 PSI
VELOCITY	3600 RPM
MATERIAL SEAL	Stationary face - bronze filled carbon
MATE	Rotating face - nitralloy
FLUID	Liquid propane
SURFACE FINISH	Seal faces - within 3 light bands. Stationary gaskets - 20 microinches RMS max.
LEAK RATE	Normal leakage 12 to 15 drops/minute
TEST DURATION	500 Hours
REMARKS	Shaft diameter 2-1/2 inches. Several seals exceeded 10000 hrs.

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTATIONAL
SEAL TYPE --- FACE

REFERENCE	R. E. Connelly 1959		
NO.	C054		
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X
TEMPERATURE °F MAX. RANGE	(See Test Fluid Below) →		
PRESSURE PSI MAX. RANGE	200 PSI	200 PSI	200 PSI
VELOCITY FPM	8000	8000	8000
MATERIAL			
SEAL	Carbon	Carbon	Carbon
MATE	Nitralloy	Chrome Plate	Titanium
FLUID	LN_2	LN_2	LN_2
SURFACE FINISH (μ in. CLA)			
LEAK RATE (Atm. cc/sec)	Low	Low	Low
TEST DURATION LIFE	5 Minutes	5 Minutes	5 Minutes
REMARKS	Face Loading 100 PSI →		

Note:
Additional Tests: Bronze vs Nitralloy; PTFCE vs Nitralloy; PTFE vs Nitralloy; Aluminum vs Chrome Plate - (Considered failures since operational life was very short.)

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTATING
SEAL TYPE --- FACE

REFERENCE	R. P. Cochran (Refer to AD 119437) January 9, 1957		
NO.	C046		
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X
TEMPERATURE °F	1090 (Outer) 940 (Inner)	260	440
PRESSURE PSI MAX. RANGE	38.3 Blade Cooling Air →		
VELOCITY FPM	10,000	6,940	10,000
MATERIAL			
SEAL	Petroleum Coke with 20-35% Graphite (Rings) →		
MATE	Inconel X with Chrome Plate (Disk) →		
FLUID	Air →		
SURFACE FINISH (μ in. CLA)	2 μ in rms →		
LEAK RATE (Atm. cc/sec)	None →		
TEST DURATION LIFE	13.5 Hours →		
REMARKS	No blade cooling →	Constant cooling →	
	Balanced pressure conditions →		

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTARY
SEAL TYPE --- FACE

REFERENCE	Paxton, B. R. November 1962				
NO.	Supp 071				
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	Ambient →				
PRESSURE PSIG RANGE	0 - 5.5 PSIG →				
VELOCITY	9600 FPM →				
MATERIAL SEAL	Carbon B (org./ inorg. impreg.) →				
MATE	AISI - C10108	Al oxide on 4140	Carbon B	Chrome on 4140	4140
FLUID	Air →				
SURFACE FINISH (μ in. CLA)					
LEAK RATE	Less than 0.12 CFM →				
TEST DURATION	3.5 Hours	3.5 Hours	3.5 Hours	3.5 Hours	3.5 Hours
CARBON WEAR (in/10^6 ft travel)	11×10^{-5}	13×10^{-5}	14.5×10^{-5}	16×10^{-5}	19×10^{-5}
CARBON TEMP.	670°F	560°F	550°F	550°F	630°F
REMARKS	1/2 Hour test time at each face load pressures of 0, 3, 6, 9, 12, and 18 PSI (total 3.5 hours) Seal air pressure 2.5 PSIG equals 6 PSI face pressure; Seal air pressure 5.5 PSIG equals 18 PSI face pressure				

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTARY
SEAL TYPE --- FACE

REFERENCE	Paxton, B. R. November 1962
NO.	Supp 071
INFORMATION BASED ON TESTS RECOMMENDATION	X
TEMPERATURE °F MAX. RANGE	Room Ambient
PRESSURE PSIG MAX. RANGE	4 PSIG
VELOCITY FPM	16700
MATERIAL SEAL	Carbon "C" (Anti-oxid. impreg.)
MATE	Chrome plated AISI 4140
FLUID	Air
SURFACE FINISH (μ in. CLA)	Carbon 2 - 4 light bands; Chrome plate 4 - 6 RMS
LEAK RATE (Atm. cc/sec)	Satisfactory (See remarks below)
TEST DURATION LIFE	6 Hours
REMARKS	Max. allowable leakage 0.12 CFM; Carbon avg. temp. 600°F; Face pressure - 12 PSI; Mean diameter - 4 inches

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTARY
SEAL TYPE --- FACE - FLOATING

REFERENCE NO.	Schweiger, F. A, June 1963 Supp 072
INFORMATION BASED ON TESTS RECOMMENDATION	X(Phase I)
TEMPERATURE °F	400°F - Lub. Temp. 1050°F - Amb. Temp. 600°F - Seal press. Air Temp.
PRESSURE PSIG MAX.	63 PSIG Seal pressurization
VELOCITY FPM	11700 FPM
MATERIAL	
SEAL	2490 Carbon and 2573 Carbon
MATE	8740 Steel - flame plated Linde LA2 Aluminum Oxide
FLUID	Heyden Newport TP 653B oil
SURFACE FINISH	
LEAK RATE	Seal performance good for 30 hours
TEST DURATION LIFE	151 Hours 29 Minutes
REMARKS	The floating face seal employs a hydrostatic air bearing face and tandem secondaries. Allowable oil leakage - 1.3 SCFM (oil side), 3.2 SCFM (air side). This work performed as part of the supersonic transport component development program.

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTARY
SEAL TYPE --- FACE - FLOATING

REFERENCE NO.	Schweiger, F. A. June 1963 Supp 072
INFORMATION BASED ON TESTS RECOMMENDATION	X (Phase II)
TEMPERATURE °F	600°F Seal Press. air temp. 325°F Oil chamber
PRESSURE PSIG MAX.	63 PSIG - Seal pressurization 10 PSIG - Air chamber
VELOCITY	7000 RPM
MATERIAL	
SEAL	2480 Carbon
MATE	8740 Steel - plated with Linde LA-2 Aluminum Oxide
FLUID	Esso Turbo 15 (MIL-L-7808)
SURFACE FINISH	Seal lapped within two light bands
LEAK RATE (Atm. cc/sec)	N.A.
TEST DURATION LIFE	16 Hours 20 Minutes at conditions stated
REMARKS	Satisfactory performance

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTARY
SEAL TYPE --- FACE

REFERENCE	Allen, C. M. August 1959		
NO.	A010		
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X
TEMPERATURE °F MAX. (Stator) RANGE	90	98	95
PRESSURE PSIG MAX.	5	5	5
VELOCITY RPM	11400	195000	22300
VELOCITY FPM	10890	18610	21280
MATERIAL	(Stator) CCP72 (Carbon Graphite) →		
	(Rotor) Flame-plated WC on steel →		
FLUID	Helium →		
SURFACE FINISH	Flatness, one light band: finish - one microinch RMS		
LEAK RATE (Lb/Hr)	0.001	0.0008	0.0018
TEST DURATION LIFE (Hours)	7	7	7
REMARKS	Unit face load 5.8 PSI →		
	Water cooled (0.04 GPM) →		
	Above is sample of test program. Eleven materials were chosen for evaluation as primary seal elements.		

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTARY
SEAL TYPE --- FACE

REFERENCE	Allen, C. M. November 1959			
NO.	A010			
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X
TEMPERATURE °F (Environment)	315	300	320	345
(Stator)	300	290	295	290
PRESSURE PSIG MAX. RANGE	10	10	10	10
VELOCITY RPM	20200	20200	20000	20400
MATERIAL SEAL	(Stator) CCP72 (Carbon Graphite) →			
MATE	(Rotor) Flame-plated Cr_2C_3 on steel →			
FLUID	Air	$N_2 + 1\% O_2$	N_2	He
SURFACE FINISH	Flatness - one light band; finish - one microinch rms →			
LEAK RATE (Lb/Hr)	0.004	0.06	0.0016	0.008
TEST DURATION LIFE (Hours)	1.5	1.5	1.5	1.25
REMARKS	Face Load 6.6 PSI - without liquid cooling. Above data is sample of test program. Two primary seal material combinations were evaluated with other gaseous environments.			

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTARY
SEAL TYPE --- FACE

REFERENCE	Allen, C. M. August 1959		
NO.	A010		
INFORMATION BASED ON **TESTS** **RECOMMENDATION**	X	X	X
TEMPERATURE °F **MAX.** (Stator) **RANGE**	90	98	95
PRESSURE PSIG **MAX.**	5	5	5
VELOCITY RPM FPM	11400 10890	195000 18610	22300 21280
MATERIAL	(Stator) CCP72 (Carbon Graphite) →		
	(Rotor) Flame-plated WC on steel →		
FLUID	Helium →		
SURFACE FINISH	Flatness, one light band: finish - one microinch RMS		
LEAK RATE (Lb/Hr)	0.001	0.0008	0.0018
TEST DURATION **LIFE** (Hours)	7	7	7
REMARKS	Unit face load 5.8 PSI →		
	Water cooled (0.04 GPM) →		
	Above is sample of test program. Eleven materials were chosen for evaluation as primary seal elements.		

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTARY
SEAL TYPE --- FACE

REFERENCE	Allen, C. M. November 1959			
NO.	A010			
INFORMATION BASED ON **TESTS** **RECOMMENDATION**	X	X	X	X
TEMPERATURE °F (Environment) (Stator)	315 300	300 290	320 295	345 290
PRESSURE PSIG **MAX.** **RANGE**	10	10	10	10
VELOCITY RPM	20200	20200	20000	20400
MATERIAL				
SEAL	(Stator) CCP72 (Carbon Graphite) →			
MATE	(Rotor) Flame-plated Cr_2C_3 on steel →			
FLUID	Air	$N_2 + 1\%O_2$	N_2	He
SURFACE FINISH	Flatness - one light band; finish - one microinch rms →			
LEAK RATE (Lb/Hr)	0.004	0.06	0.0016	0.008
TEST DURATION **LIFE** (Hours)	1.5	1.5	1.5	1.25
REMARKS	Face Load 6.6 PSI - without liquid cooling. Above data is sample of test program. Two primary seal material combinations were evaluated with other gaseous environments.			

GENERAL REFERENCES

DYNAMIC SEALS -- ROTARY MOTION

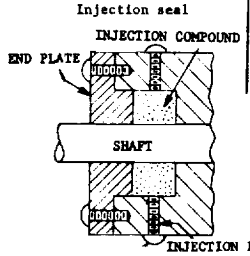
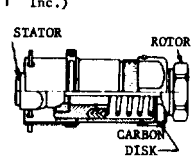
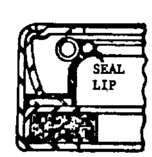
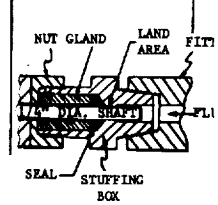
CATEGORY -- I.A.1.a.(2)

1. Elonka, S., Mechanical Packings, Power, V. 99, pp. 107-130, March 1955, (E-019)
2. Syracuse University Research Institute, Development of Rotary and Translatory Seals--Final Report, Aug. 1954, D.D.C. Report #AD 48377
3. Anonymous, The Design of O-Ring Seals for Rockets, U.S. Naval Ordnance Test Station Technical Memorandum #1046, March 1952, D.D.C. Report No. AD 82306
4. Dawton, R.H.V.M., High Vacuum Shaft Seals, Flanged Joints, and The Gassing and the Permeability of Rubber-like Material, July 1956, D.D.C. Report #AD 116938

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTATING
SEAL TYPE --- PACKING

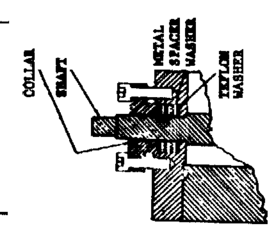
I.A.1.a.(2)
Chart 1

REFERENCE NO.	C.S. Grove Jr. Aug. 1954 AD48377		Anonymous Feb. 1930 A-041	Anonymous Dec. 1950 Supp. 006	R.S. Rainey May 1950 R-006	R.E. Headrick Oct. 1962 H-028
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX.	75				200	750
RANGE			338-464	-40 to 450		
PRESSURE PSIA MAX.	30" of Hg			165 for steam. 315 for hydraulic oil and chemicals.		1015
RANGE			60-200			
VELOCITY FPM	60 RPM		8700 (3000 RPM)	Up to 1000 RPM	2000	
MATERIAL SEAL	American Felt Seal No. 7544: felt and hycar - oil impregnated.	FRL-3	Huhn's packing (segmented carbon rings; corrugated spring made of nickel plated band steel).	Non-porous carbon disk.	Synthetic material with auxiliary of leather, felt, or any other suitable material.	40% dense stainless steel impregnated with silver.
MATE	1/4" dia. hardened, polished steel shaft.		11 inch dia. shaft	Rigid pipe balls		Shaft was flame plated with chromium carbide.
FLUID	Nitrogen		Steam	Steam, air, oil, gas or water		Hydraulic
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm. cc/sec)	0.0937-0.1140 cu. in. per (yr.)(psig)	0.1 cu.in. per (yr.) (psig)	30 to 86 liters/hr.			Less than 1 drop/min.
TEST DURATION	2840 hrs.	100-1200 hrs.				625,000 rev. in 125 min.
LIFE	2840 hrs.	100-1200 hrs.				
REMARKS		Injection seal		Rotating joint. Joints come in 1/2", 3/4" and 1" pipe sizes. (Rotherm Engineering Co., Inc.)	Dual-type spring-loaded seal. Used in dusty atmospheres or where abrasive particles must be sealed from oil, also outer axle seals, front end of engines, unprotected pinions and transmissions.	
DESIGN						

DYNAMIC SEAL
APPLICATION -- ROTATING
SEAL TYPE --- PACKING

SEAL ANALYSIS

REFERENCE	E.A. Billett et al, Feb. 1958
NO.	B-054
INFORMATION BASED ON: TEST	X
INFORMATION BASED ON: RECOMMENDATION	
TEMPERATURE °F: MAX.	
TEMPERATURE °F: MIN.	
TEMPERATURE °F: RANGE	70-392
PRESSURE PSIA: MAX.	
PRESSURE PSIA: MIN.	
PRESSURE PSIA: RANGE	Vacuum
VELOCITY RPM	
MATERIAL: SEAL	.062" thick teflon washers and .062" thick polished steel spacer washers.
MATERIAL: MATE	
FLUID	Helium
SURFACE FINISH (μ in. CLA)	
LEAK RATE (Atm. cc/sec)	11 x 10^{-5} μ/sec.
TEST DURATION	
LIFE	
REMARKS	
DESIGN	



GENERAL REFERENCES

DYNAMIC SEALS -- ROTARY MOTION

CATEGORY I.A.1.a.(3)

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2. Dega, R.L., and Symons, J.D., Seal Testing to Establish Quality Control Specifications Can Reduce Leakage, SAE Trans., V. 68, pp. 440-447 - 1960, (D-017)

3. Gillespie, L. H., How to Design with Teflon Packings, Applied Hydraulics and Pneumatics, V. 12, pp. 121-124- July 1959, (G-022)

4. Jagger, E.T., Rotary Shaft Seals -- Some Effects of Shaft Surface Finish, Int. Conf. on Fluid Sealing, Paper Al, 14 pp., April 17-19, 1961, British Hydromechanics Research Assn., Harlow, Essex, England, (J-011)

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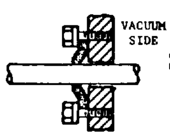
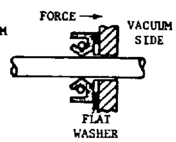
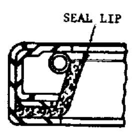
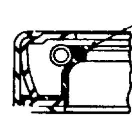
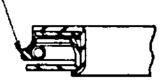
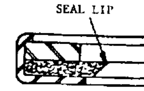
SEAL ANALYSIS

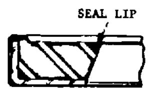
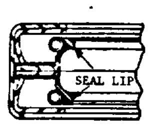
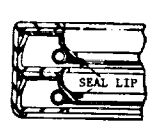
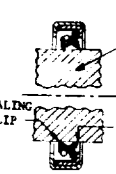
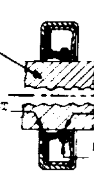
DYNAMIC SEAL
APPLICATION -- ROTATING
SEAL TYPE --- LIP

REFERENCE NO.	R.H.V.M. Dawton Oct. 1957 D-014		R.S. Rainey May 1950 R-006			
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	248	248	200	300	300-synthetic, 200-leather	180-200
PRESSURE PSIA MAX. RANGE	0 to 14.7					
VELOCITY FPM	5.9 (1 rev./sec. of 3/8" Dia. Shaft)		2000	3600-4000		1000
MATERIAL SEAL	Neoprene		Leather	Synthetic material	Leather or synthetic material	Leather
MATE	Steel, Glass, Brass					
FLUID	Air					
SURFACE FINISH (μ in. CLA)				Well finished		
LEAK RATE (Atm. cc/sec)	5×10^{-6}					
TEST DURATION		200 Hrs.				
LIFE	2 days					
REMARKS	Lubrication: Apiezon "B" Oil		Spring loaded seal. Cannot be used when shaft runout is high. General purpose uses; automobile, tractor, truck wheels and farm implements.	Can be used with shaft runouts as high as 0.030". Not used without lubrication. Uses: engines, transmissions, pinions, and in general applications where the shaft speed is above 2000 RPM.	External expansion spring-loaded seal. The seal rotates and can be useful in such applications as wheel spindles on vehicles.	Quite effective against heavy lubricants when speed and temp. are low. Effective as dust and dirt seal. It is simple and inexpensive.
DESIGN						

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTATING
SEAL TYPE --- LIP

I.A.1.a.(3)
Chart 2

REFERENCE NO.	R.S. Rainey May 1950 R-006				S.M. Lillis March 1960 L-031	
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	250-300	Usually not over 180	180-200			
PRESSURE PSIA MAX. RANGE	30-40	85-115				
VELOCITY FPM	3000-3600	Usual Max. = 800-1000	1800-2000	1000	Normal: 2100 RPM. Max: 23 RPM	
MATERIAL SEAL	Synthetic material		Leather or synthetic material.		Polyacrylic rubber	Silicone rubber
MATE					Hub: 45-55 Rockwell C.	Hub: 50-55 Rockwell C.
FLUID					High grade engine oil	
SURFACE FINISH (μ in. CLA)					Hub: 16 (Max.)	Hub: 12
LEAK RATE (Atm. cc/sec)						
TEST DURATION LIFE					2000 hrs.	500 hrs.
REMARKS	Steel supported seal. Should not be used where there is excessive runout or whip. Applications: inner automobile axle seals, low speed gear boxes, & farm equipment.	High pressure seal. Used in pumps & on extremely small shafts where the relation of ID to OD does not allow space for relief.	Combination spring-loaded seal. Must be lubricated from both sides. Seal is used for the separation & retention of of two fluids.	Tandem spring-loaded seal. Both members must be lubricated. For heavy duty applications or under a head of oil. Used sometimes on track roller applications, disc plows, etc.	Diesel truck engine front crankshaft oil seal.	Diesel engine rear crankshaft oil seal.
DESIGN						

DYNAMIC SEAL
APPLICATION -- ROTARY
SEAL TYPE --- LIP

SEAL ANALYSIS

REFERENCE	L. R. Sitney
NO.	1952 S058
INFORMATION BASED ON: TESTS	X
INFORMATION BASED ON: RECOMMENDATION	
TEMPERATURE °F MAX. RANGE	Ambient
PRESSURE MAX. RANGE	* 8×10^{-6} mm/Hg
VELOCITY	6600 RPM
MATERIAL: SEAL	Rubber
MATERIAL: MATE	Stainless Steel
FLUID	
SURFACE FINISH (μ in. CLA)	Shaft polished with crocus cloth
LEAK RATE (Atm. cc/sec)	(Operation satisfactory)
TEST DURATION LIFE	150 Hours
REMARKS	* Limited vacuum of system 2 - Klozure seals (Garlock Model 63) Seal lubricated with DC-702 Silicone oil and DC-33 Silicone Grease Guard vacuum between seals

GENERAL REFERENCES

DYNAMIC SEALS -- ROTARY MOTION

CATEGORY -- I.A.1.a.(4)

1. Anonymous, O-Ring Packings Simplify Lightweight Hydraulic Equipment, Prod. Engrg., V.15, pp. 581-585, Sept. 144, (A-091)

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5. Hoblin, L.E., et al, Kinetic and Static Friction of the O-Ring Type Gasket, March, 1952, D.D.C. Report AD 103928

6. Tipton, F.W., Design Data for O-Rings and Similar Elastic Seals, Nov. 1956, D.D.C. Report Supp. AD 110598

7. Trepus, G.E., Design Data for O-Rings and Similar Elastic Seals, Sept. 1957, D.D.C. Report AD 131094

8. Pedersen, H.O., et al, Evaluation of Bearing and Seals in 90% Hydrogen Peroxide, Feb. 1959, D.D.C. Report AD 225213

9. Tipton, F. W., Design Handbook for O-Rings and Similar Elastic Seals, Oct. 1959, D.D.C. Report Supp. AD 230639

10. Krivitsky, A.S., et al, Development of Self-lubricating and Non-adhering Elastomeric Materials, Oct. 1960, D.D.C. Report AD 254806

11. Walker, W.R., Design Handbook for O-Rings and Similar Elastic Seals, April 1961, D.D.C. Report AD 265443

I.A.1.a.(4)
Chart 1

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTATING
SEAL TYPE --- RINGS

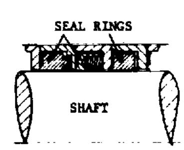
REFERENCE / NO.	L.E. Van Hise et al March 1955 AD68495	→	→	→	→	→
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	70-120	70-104	70-95	72-90	70-97	75-91
PRESSURE PSIA MAX. RANGE	500	445	445	500	500	200
VELOCITY FPM	1500 (4000 rpm)	1200 (3200 rpm)	790 (2100 rpm)	1500 (4000 rpm)	→	1200 (3200 rpm)
MATERIAL SEAL	Buna N O-Ring - 1.437"I.D., 0.139" Thick	→	→	→	→	→
	Durometer 80	→	Durometer 60-70	Durometer 80	→	→
MATE	Chrome Plate on Steel Shaft	→	→	Naval Brass Shaft	Bronze Shaft	Beryllium Copper Shaft
FLUID	Salt Water	→	→	→	→	→
SURFACE FINISH (μ in. CLA)	4.5 to 5.25	→	→			
LEAK RATE (Atm. cc/sec)	Good Run	→	→	→	→	→
TEST DURATION	10 Min.	→	→	→	→	→
LIFE						
REMARKS	"Spirolox" Type Retaining Ring Used	→	→	→	→	→
*% Squeeze	1	6	6	1	0.4	4
Lubrication	RPM 5	→	D.C. 4	→	→	→
DESIGN						

$$\text{*\% Squeeze} = \frac{\text{Shaft Diam.} - \text{O-Ring I.D.}}{2} \div \text{O-Ring Thickness} \times 100$$

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTATING
SEAL TYPE --- RINGS

REFERENCE / NO.	L.E. Van Hise et al March 1955 AD68495	B.L. Johnson June 1958 AD201739	A. Brkich et al Nov.-Dec. 1959 B-103
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X
TEMPERATURE °F MAX. RANGE	80-114		570
PRESSURE PSIA MAX. RANGE	200	600	6500
VELOCITY FPM	1200 (3200 rpm)	1700 to 2100 rpm	3580-5180 RPM
MATERIAL SEAL	Buna N O-Ring - 1.437" I.D., 0.139" Thick. Durometer 80	Carbon Sleeve Stellite Face with Synthetic Rubber O-Rings	Floating Rings
MATE	Shaft is BTR Tool Steel Chrome Plate	Steel	4 to 8 inch dia. shafts.
FLUID	Salt Water	Military Gasoline	Water
SURFACE FINISH (μ in. CLA)	10		
LEAK RATE (Atm. cc/sec)	Good Run	No Leakage after Run-In	Satisfactory*
TEST DURATION	10 Min.	1000 Hrs.	2800 hrs.
LIFE		More than 1000 Hrs.	
REMARKS	"Spirolox" Type Retaining Ring Used 2% Squeeze* D.C. 4 Lubricant	Pump Shaft Seal. Sleeve Type Mechanical Seal. Mfg. by J. Crane Co.	Boiler feed pump shaft seals. *Controlled leakage.
DESIGN	* See I.A.1.a.(4) Chart 1 for Definition		

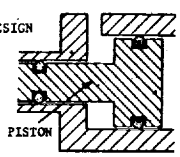


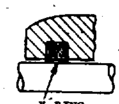
I.A.1.a.(4)
Chart 2

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTATING
SEAL TYPE --- RINGS

REFERENCE NO.	M.H. Everett Jan. 1961 E-034 →					R.L. Hayman March 1945 H-024
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	-20 to 350	-80 to 250	-65 to 225	-100 to 350	600 -40 to 450	
PRESSURE PSIA MAX. RANGE	See Remarks →					Pulsating 0 to 3000*
VELOCITY FPM	350-600 (Depending on Shaft Diameter) →					
MATERIAL SEAL	Polyacrylate - Acrylic Acid Ester Polymer	Butyl Rubber	Polysulfide Polymer	Fluorosilicone Polymers	Hexafluoropropylene and Vinylidene Fluoride Copolymers	Two "O" Rings in Single Groove
MATE	Metal →					
FLUID	Oil, Ozone	Hyd. Fl. of Phosphate Ester Gases, Ozone, Veg. Oils	Ozone, Pet. Base Fuels & Oils, Aromatic Solv.	Pet. Based Fuel & Lub., Ester Type Lub., Hydrocarbons (Many)	Great Many Fluids and Chemicals	
SURFACE FINISH (μ in. CLA)	32 to 125 on Groove Sides; 16 on Diameter over which the O-Ring must slide →					
LEAK RATE (Atm. cc/sec)						Satisfactory
TEST DURATION						20,000 Cycles
LIFE						
REMARKS	5% Peripheral (Compression) Loading → "O"-Rings are good up to a pressure of 5000 PSI (Varies with the hardness of the ring material and the clearances.) → Ring I.D. must be larger than the Shaft Diameter →					Swing Joint Seal. *30 Times/Min. at a Rate of Pressure Increase of 200,000 PSI/Sec.
DESIGN						

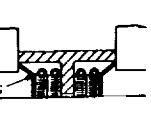




SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTATING
SEAL TYPE --- RINGS

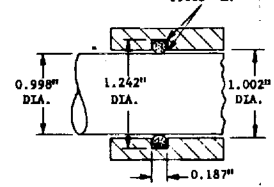
REFERENCE NO.	H.O. Pedersen et al Feb. 1959 AD225213 →					
INFORMATION BASED ON TESTS RECOMMENDATION	In Fixture →					
TEMPERATURE °F MAX. RANGE	Gradient 34 to 75	Gradient 25 to 1310	Gradient 18 to 75	Gradient 28 to 1355	47 to 91	40 to 140
PRESSURE PSIA MAX. RANGE	ΔP = 125	ΔP = 250 →		ΔP = 300	25 to 160	15 to 25
VELOCITY FPM	16,400 to 17,300 RPM	10,000 to 16,000 RPM	14,000 to 17,800 RPM	12,000 to 16,000 RPM	Approx. 16,000 RPM →	
MATERIAL SEAL	"S" Monel (Segmented Design) →		Teflon Glass (Segmented) →		Aluminum Oxide →	
MATE	NI-Resist. Sleeve →		Electrolized #347SS →		25% Glass Filled Teflon →	
FLUID	90% H_2O_2 One Side → Air on Other	Decomposed Peroxide Gas on Other	Air on Other	Decomposed Peroxide Gas on Other	Nitrogen on Other	Air on Other
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm. cc/sec)	107		30		Inlet Seal 0.08 to 1.33	Outlet Seal 0.075 to 1.26
TEST DURATION	5 Min.	30 Min. →		18 Min.	3 to 6 Min. Runs, 120 Min. Total →	
LIFE						
REMARKS	Rocket Engine, Turbo-pump System Fluid Cooled Bearing → No Galling or Pick-up →			Decomposed Peroxide Gas Deteriorated Glass-Teflon Seal	Rocket Engine, Turbo-pump, Fluid Cooled Bearing Seal remained excellent but recommend KEL "F" O-rings be replaced with metal.	
DESIGN						



SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTATING
SEAL TYPE --- RINGS

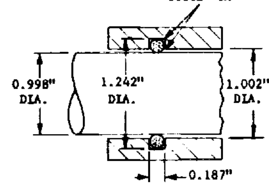
REFERENCE	G.E. Trepus Sept. 1957					
NO.	AD 131094					
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	75					150
PRESSURE PSIA MAX. RANGE	5000					
VELOCITY FPM	10					
MATERIAL	"O"-Rings					
SEALS	11 Different Nitrile Rubbers (Buna N) and Leather back-up Rings.		Philprene VP-A Shore A = 95 No back-up Rings	Hycar 1072X3 Shore A = 77	Philprene VP-A Durometer (Shore A) = 95	
MATE						
FLUID	Velocite - K					
SURFACE FINISH (μ in. CLA)	20 (Shaft)					
LEAK RATE (Atm. cc/sec)	Failure Indicated by Wetting of Talc Dust at Shaft End					
TEST DURATION	Rotations To Failure:					
LIFE	Max.-3331 Ave.-1707	Max.-3436 Ave.-1200	16,800	16,000	6,790	20,790
REMARKS	"O"-Ring Size AN6227-19 (0.984"IDx0.139" Wide) Rings Lubricated With Petrolatum	No Lubrication	Ring Lubricated with Petrolatum		No Lubrication Physical Properties Listed in Test Data Tables Are Not Necessarily Taken At The Test Conditions.	
DESIGN			Best of 14 Different Materials.		Best of 18 Different Materials.	Best of 17 Different Materials.



SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTATING
SEAL TYPE --- RINGS

REFERENCE	G.E. Trepus Sept. 1957					Anonymous Feb. 1930
NO.	AD 131094					A-041
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	200	75	150	200	75	338-464
PRESSURE PSIA MAX. RANGE	5000				1500	60-200
VELOCITY FPM	10					8700 (3000 RPM)
MATERIAL	"O"-Rings Philprene VP-A	9 Different Nitrile Rubbers			11 Different Nitrile Rubbers	Huhn's Packing: Segmented Carbon Rings; Corrugated Spring Made of Nickel Plated Band Steel.
SEAL	Durometer (Shore A) = 95. No back-up Rings					
MATE						11 inch dia. shaft
FLUID	Velocite - K					Steam
SURFACE FINISH (μ in. CLA)	20 (Shaft)					
LEAK RATE (Atm. cc/sec)	Failure Indicated by Wetting of Talc Dust at Shaft End					30 to 86 Liters/hr.
TEST DURATION	Rotations To Failure:					
LIFE	14,780	Max.-1,580 Ave.- 810	Max.-1,140 Ave.- 447	Max.-580 Ave.-221	Max.-7,057 Ave.-3,981	
REMARKS	"O"-Ring Size AN6227-19 (0.984"IDx0.139" Wide) No Lubrication Physical Properties Listed in Test Data Tables Are Not Necessarily Taken At The Test Conditions				Ring Lubricated With Petrolatum	
DESIGN	Best of 15 Different Materials					



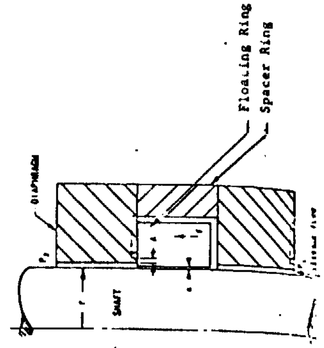
I.A.1.a.(4)
Chart 7

DYNAMIC SEAL
APPLICATION -- ROTATING
SEAL TYPE --- RING (Face - Labyrinth)

SEAL ANALYSIS

REFERENCE	Smardyk, A. April 1954		
NO.	S063		
INFORMATION BASED ON TESTS	X	X	X
RECOMMENDATION			
TEMPERATURE °F MAX. RANGE	200	200-450	125
PRESSURE PSI MAX. RANGE	2000 (12 rings)	2000 (12 rings)	2000 (12 rings)
VELOCITY RPM	385	116-496	3500
MATERIAL SHAFT	Armco 17-4 ph-Chrome Plate	Type 440C-Chrome Plate	Colmonoy 6 on Inconel
SEAL RING	Stellite 3	Stellite 3	Titanium Carbide
DIAPHRAGM	K-Monel	K-Monel	Titanium Carbide
FLUID	Distilled Water	Distilled Water	Distilled Water
SURFACE FINISH (μ in. CLA)	Lapped	Lapped	Lapped
LEAK RATE (lb/hr)	150/200 Initial/Final	90/275 Initial/Final	350
TEST DURATION LIFE (Hours)	13878	11600	2767
REMARKS	Shaft clean but rings had brown oxide coating. Seal Shaft Diameter - 1-1/4 inches (All tests)	Test discontinued.	Shaft worn 0.001 inch. Rings in excellent condition.

DESIGN - See Sketch



GENERAL REFERENCES

DYNAMIC SEALS -- ROTARY MOTION

CATEGORY -- I.A.1.b.(1)

1. Anonymous, Flow Through Labyrinth Packings, Mech. Engrg., V. 56, pp. 678-680, Nov. 1934, (A-059)

2. Heffner, F.E., A General Method For Correlating Labyrinth Seal Leak Rate Data, A.S.M.E. Paper No. 59 - Lub - 7, 8 pages, Oct. 1959, See also ASME Trans., Journal of Basic Engineering, V. 82, No. 2, pp. 265-275, June, 1960, (H-033)

3. Kuchler, T.C., and Pearson, W., Clearance Seals, Machine Design, The Seals Book, pp. 20-23, Jan. 19, 1961, (K-044)

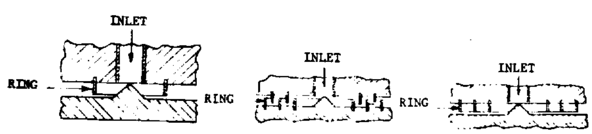
I.A.1.b.(1)
Chart 1

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTATING
SEAL TYPE --- LABYRINTH

REFERENCE	W.J. Kearton →		
NO.	Oct. 1954 AD85318		
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X
TEMPERATURE °F MAX RANGE	100		
PRESSURE PSIA MAX. RANGE	16.2 to 114.7	15 to 95	
VELOCITY FPM			
MATERIAL SEAL	Steel (1/32" thick)	Rolled brass (1/32" thick)	Brass (1/64" thick)
MATE			
FLUID	Air →		
SURFACE FINISH (μ in. CLA)			Lapped
LEAK RATE (Atm. cc/sec)		Discharge coefficients were lower than those in a single ring gland.	Discharge coefficients greater than unity were measured.
TEST DURATION LIFE			
REMARKS	Tests concerning the flow of air through radial labyrinth glands.	The observed pressure distribution agreed well with the theoretical values.	
DESIGN	Single ring gland	20 ring gland (staggered type).	10 ring gland (unstaggered type).

Coefficient of discharge equals the ratio of the actual mass flow to the mass flow based on isentropic expansion and the full area of construction. The report contains numerous equations and curves relative to design.



I.A.1.b.(1)
Chart 2

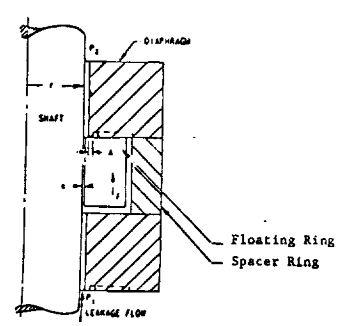
SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTATING
SEAL TYPE --- LABYRINTH (Ring Type - Face)

REFERENCE	Smaardyk, A.		
NO.	April 1954 S063		
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X
TEMPERATURE °F MAX. RANGE	200	200-450	125
PRESSURE PSI MAX. RANGE	2000 (12 rings)	2000 (12 rings)	2000 (12 rings)
VELOCITY RPM	385	116-496	3500
MATERIAL SHAFT SEAL RING DIAPHRAGM	Armco 17-4 Ph-Chrome Plate Stellite 3 K-Monel	Type 440C-Chrome Plate Stellite 3 K-Monel	Colmonoy 6 on Inconel Titanium Carbide Titanium Carbide
FLUID	Distilled Water	Distilled Water	Distilled Water
SURFACE FINISH (μ in. CLA)	Lapped	Lapped	Lapped
LEAK RATE (lb/hr)	150/200 Initial/Final	90/275 Initial/Final	350
TEST DURATION LIFE (Hours)	13878	11600	2767
REMARKS	Shaft clean but rings had brown oxide coating.	Test discontinued.	Shaft worn 0.001 inch. Rings in excellent condition.

Seal Shaft Diameter - 1-1/4 inches →

DESIGN - See Sketch



Typical Floating Ring

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II.A.1.b.(1)
Chart 3

SEAL ANALYSIS

STATIC SEAL
APPLICATION -- PLASTIC DEFORMATION
SEAL TYPE --- GASKET

REFERENCE NO.	Weitzel, D.H., etc. August 1960 W031	
INFORMATION BASED ON TESTS RECOMMENDATION	X	X
TEMPERATURE MAX. RANGE	$20^{o}K$ - Amb.	
PRESSURE PSI MAX. RANGE	1000	
VELOCITY FPM	N.A.	
MATERIAL SEAL	Mylar	Nylon
MATE	Stainless Steel	
FLUID	Helium	
SURFACE FINISH (μ in. CLA)		
LEAK RATE	Less than 10^{-4}cc STP/hr/in of seal	
TEST DURATION LIFE	Several days of warm up and cool down	
REMARKS	10 mil thick gasket compressed to 3 mil Seal load provided by 6 - 1-1/4 - 28 bolts torqued to 5 ft # Mating surface of semicircular cross section of 0.084" diam. Vibrated for one hour at 30 - 500 cps, 1000 psi at 76°K	

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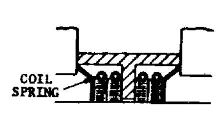
SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTATING
SEAL TYPE --- BUSHING

I.A.1.b.(2)
Chart 1

REFERENCE NO.	B.L. Johnson June 1958 AD201739	H.O. Pedersen et al Feb. 1959 AD225213			
INFORMATION BASED ON TESTS RECOMMENDATION	X	In Fixture			
TEMPERATURE °F MAX. RANGE		Gradient 34 to 75	Gradient 25 to 1310	Gradient 18 to 75	Gradient 28 to 1355
PRESSURE PSIA MAX. RANGE	600	ΔP = 125	ΔP = 250		ΔP = 300
VELOCITY FPM	1700 to 2100 RPM	16,400 to 17,300 RPM	10,000 to 16,000 RPM	14,000 to 17,800 RPM	12,000 to 16,000 RPM
MATERIAL SEAL	Carbon Sleeve Stellite Face with Synthetic Rubber O-Rings	"S" Monel (Segmented Design)		Teflon Glass (Segmented)	
MATE	Steel	NI-Resist. Sleeve		Electrolized #347SS	
FLUID	Military Gasoline	90% H_2O_2 One Side Air on Other	Decomposed Peroxide Gas on Other	Air on Other	Decomposed Peroxide Gas on Other
SURFACE FINISH (μ in. CLA)					
LEAK RATE (Atm. cc/sec)	No Leakage after Run-In	107		30	
TEST DURATION	1000 Hrs.	5 Min.	30 Min.		18 Min.
LIFE	More than 1000 Hrs.				
REMARKS	Pump Shaft Seal. Sleeve Type Mechanical Seal. Mfg. by J. Crane Co.	Rocket Engine, Turbo-pump System Fluid Cooled Bearing No Galling or Pick-up			Decomposed Peroxide Gas Deteriorated Glass-Teflon Seal
DESIGN					



I.A.2.b.
Chart 1

GENERAL REFERENCES

DYNAMIC SEALS -- ROTARY MOTION

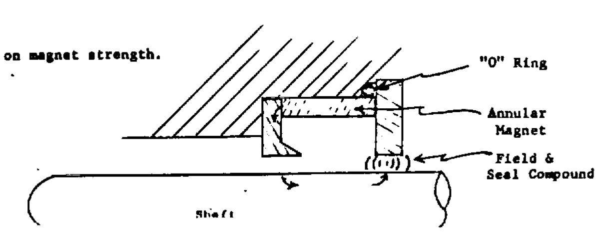
CATEGORY I.A.2.a.

1. Anonymous, Frozen Slugs Seal Shafts, Chem. Engg., V. 63, pp. 124 and 126 - March 1956, (A-155)

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ROTATING
SEAL TYPE --- MAGNETIC

REFERENCE NO.	J.W. Walley Mar/April 1962 W007	
INFORMATION BASED ON TESTS RECOMMENDATION	X	X
TEMPERATURE °F MAX. RANGE		
PRESSURE PSIG (Breakdown) MAX. RANGE	45-50	16
VELOCITY	Static	300-1500 RPM
MATERIAL SEAL Compound	Apielon G.W. Oil with Carbonyl Iron Powder →	
MATE	Steel →	
FLUID	Air →	
SURFACE FINISH (µ in. CLA)		
LEAK RATE (Atm. cc/sec)	See breakdown pressure above →	
TEST DURATION LIFE		
REMARKS	Breakdown pressure dependent on magnet strength. Shaft - 1.5" diam. Gap size, 3, 6, & 9 mils.	



DESIGN - See Schematic Sketch

I.A.3.a.
Chart 1

DYNAMIC SEAL
APPLICATION -- ROTATING
SEAL TYPE --- CENTRIFUGAL

SEAL ANALYSIS

REFERENCE	L.I. Gel'man Sept. 1956 AD259574
NO.	
INFORMATION BASED ON TESTS	X
INFORMATION BASED ON RECOMMENDATION	
TEMPERATURE °F MAX.	
TEMPERATURE °F RANGE	
PRESSURE PSIA MAX.	
PRESSURE PSIA RANGE	
VELOCITY FPM	1300 to 1550 RPM
MATERIAL SEAL	
MATERIAL MATE	
FLUID	Mercury (176°F)
	Inert Gas
SURFACE FINISH (μ in. CLA)	
LEAK RATE (Atm. cc/sec)	Does not leak when gas pressure is 2.4 psig or lower. Results were satisfactory.
TEST DURATION	
LIFE	
REMARKS	Tests on a mercury centrifugal packing used in a centrifugal pump for liquid metals. Mercury is air cooled to prevent overheating.
DESIGN	

GENERAL REFERENCES

DYNAMIC SEALS -- ROTARY MOTION

CATEGORY I.A.3.b

1. Frossel, W. Hydrodynamic Shaft Seals, Engrs. Digest, V. 18, pp. 18-20, Jan. 1957, (F-046)

2. McGrew, J.M., and McHugh, J.D., Analysis and Test of the Screw Seal in Laminar and Turbulent Operation, General Electric Co. - Adv. Tech. Lab. Report 63GL66 - May 1963, (Supp. 018)

I.B.

DYNAMIC SEAL

OSCILLATORY MOTION

GENERAL REFERENCES

DYNAMIC SEALS -- OSCILLATORY MOTION

CATEGORY -- I.B.1.a.(1)

1. McCray, C.R., Radial Positive-Contact Seals, Machine Design, The Seals Book, pp. 9-14, Jan. 19, 1961 (M-048).

CATEGORY I.B.1.a.(2)

1. Anonymous, The Design of O-Ring Seals for Rockets, U.S. Naval Ordnance Test Station Technical Memorandum #1046, March 1952, D.D.C. Report No. AD 82306.
2. Dawton, R.H.V.M., High Vacuum Shaft Seals, Flanged Joints, and the Gassing and the Permeability of Rubber-Like Materials, July 1956, D.D.C. Report AD 116938.

CATEGORY I.B.1.a.(3)

1. Symons, J. D., Engineering Facts About Lip Seals, SAE Automotive Eng. Congr., Detroit, Mich., Preprint 473A (14 pp.), Jan. 8-12, 1962, (S-140).

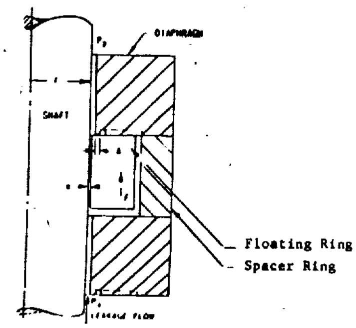
CATEGORY I.B.1.a.(4)

1. Jacobs, E., How They're Sealing Cylinders, Hydraulics and Pneumatics, V.15, pp. 67-76, 78, August 1962, (J-002).
2. Anonymous, The Design of O-Ring Seals for Rockets, U.S. Naval Ordnance Test Station Technical Memorandum #1046, March 1952, D.D.C. Report No. AD 82306.

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- OSCILLATING
SEAL TYPE --- FACE (Ring Type - Labyrinth)

REFERENCE NO.	Smaardyk, A. April 1954 S063	
INFORMATION BASED ON TESTS RECOMMENDATION	X	X
TEMPERATURE °F MAX. RANGE	85	130
PRESSURE PSI MAX. RANGE	2000 (12 rings)	2000 (12 rings)
VELOCITY CPM	175	175
MATERIAL SHAFT SEAL RING DIAPHRAGM	Colmonoy 6 on Inconel Cast Colmonoy 6 Cast Colmonoy 6	Vascoloy Ramet 171 Vascoloy Ramet 166 Vascoloy Ramet 166
FLUID	Distilled Water	Distilled Water
SURFACE FINISH (μ in. CLA)	Lapped	Lapped
LEAK RATE (Atm. lb/hr)	15/4 Initial/Final	70/100 Initial/Final
TEST DURATION LIFE (Hours)	927	2094
REMARKS	Transferred to reciprocating test.	Test discontinued.
	Seal Shaft Diameter - 1-1/4 inches. (All tests) Arc of Oscillation - 150 Degrees (All tests)	
DESIGN - See Sketch		



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SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- OSCILLATORY
SEAL TYPE --- LIP

REFERENCE NO.	D. F. Wood August 23, 1956 W076
INFORMATION BASED ON TESTS RECOMMENDATION	X
TEMPERATURE °F	212°F
PRESSURE PSIG	10 - 20 PSIG
VELOCITY	220° arc at 1/2 to 1 RPM
MATERIAL SEAL	Garlock No. 9100 split klozure
MATE	347 stainless steel
FLUID	Helium
SURFACE FINISH	
LEAK RATE (Atm.	0.001 SCF/H
TEST DURATION LIFE	17,128 cycles, 7.5 x 10^6 total degrees
REMARKS	Lubricant - plastilube No. 1 2 split seals in seal grooves. Helium between seal elements

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I.B.1.a.(4)
Chart 1

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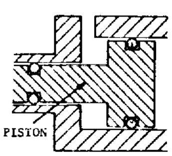
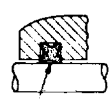
SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- OSCILLATING
SEAL TYPE --- RINGS

REFERENCE NO.	M.H. Everett Jan. 1961 E-034 →					R.L. Hayman March 1945 H-024
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	-20 to 350	-80 to 250	-65 to 225	-100 to 350	600 -40 to 450	
PRESSURE PSIA MAX. RANGE	See Remarks →					Pulsating 0 to 3000*
VELOCITY FPM						
MATERIAL SEAL	Polyacrylate - Acrylic Acid Ester Polymer	Butyl Rubber	Polysulfide Polymer	Fluorosilicone Polymers	Hexafluoropropylene and Vinylidene Fluoride Copolymers	Two "O" Rings in a Single Groove
MATE	Metal →					
FLUID	Oil, Ozone	Hyd. Fl. of Phosphate Ester Gases, Ozone, Veg. Oils	Ozone, Pet. Base Fuels & Oils, Aromatic Solv.	Pet. Based Fuel & Lub., Ester Type Lub., Hydrocarbons (Many)	Great Many Fluids and Chemicals	
SURFACE FINISH (μ in. CLA)	32 to 125 on Groove Sides; 16 on Diameter over which the O-Ring must slide →					
LEAK RATE (Atm. cc/sec)						Satisfactory
TEST DURATION						20,000 Cycles
LIFE						
REMARKS	5% Peripheral (Compression) Loading → "O"-Rings are good up to a pressure of 5000 PSI (Varies with the hardness of the ring material and the clearances)					Swing Joint (Rotary Action) *30 Times/ Min. at a Rate of Pressure Increase of 200,000 PSI/Sec.
DESIGN						





I.B.1.a.(4)
Chart 2

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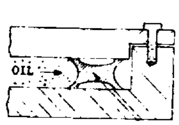
SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- OSCILLATING
SEAL TYPE --- RINGS

REFERENCE NO.	W. Sergeant March 1962 Supp. AD283459 →	
INFORMATION BASED ON TESTS RECOMMENDATION	X	X
TEMPERATURE °F MAX. RANGE		
PRESSURE PSIA MAX. RANGE	25	15 to 1015
VELOCITY FPM	8000 cy/day through a 57 degree arc.	120 cpm through a 93 degree arc.
MATERIAL SEAL	Blade Seal (MSO) Fluorinated by Quantum Incorporated →	O-rings (Nitrile Rubber)
MATE	Steel Plates*	Metal
FLUID	#9250 Oil →	
SURFACE FINISH (μ in. CLA)		
LEAK RATE (Atm. cc/sec)	Total = 21 cc	Not appreciable
TEST DURATION	400,000 oscillations	6 hrs. at 1000 psi and 16 hrs. at 400 psi.
LIFE		
REMARKS	Fluoridation was to reduce the coefficient of friction. *Top plate is stationary, bottom plate oscillates.	At 0 psi the torque is the same (10 in. lbs.) for treated or untreated rings. At 1000 psi the torque is 20 in. lbs. less for treated rings than for untreated rings (135 in. lbs.).
DESIGN		



BLADE SEAL

I.B.1.a.(4)
Chart 3

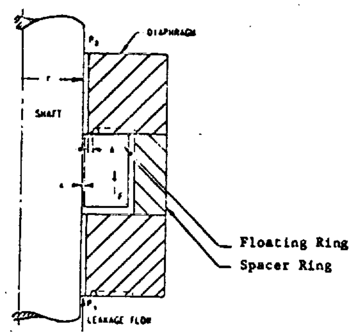
SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- OSCILLATING
SEAL TYPE --- RING (Face - Labyrinth)

REFERENCE NO.	Smaardyk, A. April 1954 S063	
INFORMATION BASED ON TESTS RECOMMENDATION	X	X
TEMPERATURE °F MAX. RANGE	85	130
PRESSURE PSI MAX. RANGE	2000 (12 rings)	2000 (12 rings)
VELOCITY CPM	175	175
MATERIAL		
SHAFT SEAL RING DIAPHRAGM	Colmonoy 6 on Inconel Cast Colmonoy 6 Cast Colmonoy 6	Vascoloy Ramet 171 Vascoloy Ramet 166 Vascoloy Ramet 166
FLUID	Distilled Water	Distilled Water
SURFACE FINISH (μ in. CLA)	Lapped	Lapped
LEAK RATE (Atm. lbs/hr)	15/4 Initial/Final	70/100 Initial/Final
TEST DURATION LIFE (Hours)	927	2094
REMARKS	Transferred to recipro-cating test.	Test discontinued.

Seal Shaft Diameter - 1-1/4 inches. (All tests)
Arc of Oscillation - 150 Degrees (All tests)

DESIGN - See Sketch



Typical Floating Ring Assembly

I.B.1.b.(1)
Chart 1

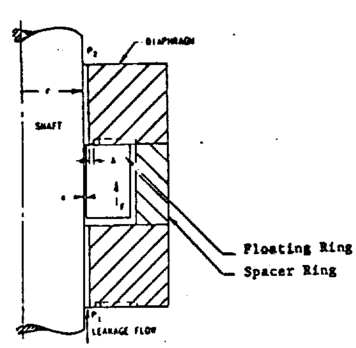
SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- OSCILLATING
SEAL TYPE --- LABYRINTH (Ring Type - Face)

REFERENCE NO.	Smaardyk, A. April 1954 S063	
INFORMATION BASED ON TESTS RECOMMENDATION	X	X
TEMPERATURE °F MAX. RANGE	85	130
PRESSURE PSI MAX. RANGE	2000 (12 rings)	2000 (12 rings)
VELOCITY CPM	175	175
MATERIAL		
SHAFT SEAL RING DIAPHRAGM	Colmonoy 6 on Inconel Cast Colmonoy 6 Cast Colmonoy 6	Vascoloy Ramet 171 Vascoloy Ramet 166 Vascoloy Ramet 166
FLUID	Distilled Water	Distilled Water
SURFACE FINISH (μ in. CLA)	Lapped	Lapped
LEAK RATE (Atm. lbs/hr)	15/4 Initial/Final	70/100 Initial/Final
TEST DURATION LIFE (Hours)	927	2094
REMARKS	Transferred to recipro-cating Test.	Test discontinued.

Seal Shaft Diameter - 1-1/4 Inches (All tests)
Arc of Oscillation - 150 Degrees(All tests)

DESIGN - See Sketch



Typical Floating Ring Assembly

I.C.

DYNAMIC SEAL

RECIPROCATING MOTION

GENERAL REFERENCES

DYNAMIC SEALS -- RECIPROCATING MOTION

CATEGORY I.C

1. Anonymous, Field Survey of Automotive Lubricants and Seals, Coordinating Research Council, Inc. - CRC Project No. CM-16-58, March 1960, (Supp. 001).

CATEGORY I.C.1

1. Cheney, L. E. and Mueller, W. J., and Duval, R. E., Frictional Characteristics of O-Rings with Typical Hydraulic Fluid, Am. Soc. Mech. Engrs.-Trans. V. 72, pp 291-7, April 1950, (C-025).

CATEGORY I.C.1.a

1. Isenbarger, R. O., Sealing Devices for Exclusion of Foreign Material from Hydraulic Mechanisms, Proc. National Conference Industrial Hydraulics, V. 14, pp. 207-19, Dics. pp. 219-22 - Oct. 1960, (I-009).

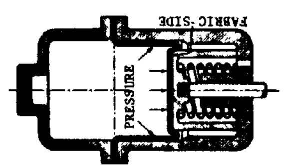
CATEGORY I.C.1.a.(1)

1. McCray, C. R., Radial Positive-Contact Seals, Machine Design, The Seals Book, pp. 9-14, Jan. 19, 1961, (M-048).

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- DIAPHRAGM

SEAL ANALYSIS

REFERENCE NO.	Anonymous May 1961 Supp. 003
INFORMATION BASED ON TESTS	
INFORMATION BASED ON RECOMMENDATION	X
TEMPERATURE °F MAX. RANGE	-94 to +392
PRESSURE PSIA MAX. RANGE	up to 115
VELOCITY FPM	
MATERIAL SEAL	Terylene & nylon. (1/2" to 4" dia.)
MATERIAL MATE	
FLUID	
SURFACE FINISH (μ in. CLA)	Not critical
LEAK RATE (Atm. cc/sec)	
TEST DURATION LIFE	
REMARKS	Requires no lubrication. Can accommodate long strokes.
DESIGN	



I.C.1.a.(1)
Chart 1

GENERAL REFERENCES

DYNAMIC SEALS -- RECIPROCATING MOTION

CATEGORY -- I.C.1.a.(2)

1. Elonka, S., Mechanical Packings, Power, V. 99, pp. 107-130 - March 1955, (E-019)

2. Garvin, J.M., Designing a Seal for Very High Pressure, Applied Hydraulics, V. 10, pp. 141-145, May 1957, (G-012)

3. Anonymous, The Design of O-Ring Seals for Rockets, U.S. Naval Ordnance Test Station Technical Memorandum #1046, March 1952, D.D.C. Report No. AD 82306

4. Tisch, R.E., Friction Study of Hydraulic Seals, June 1955, D.D.C. Report AD 102156

5. Dawton, R.H.V.M., High Vacuum Shaft Seals, Flanged Joints, and the Gassing and the Permeability of Rubber-like Materials, July 1956, D.D.C. Report AD 116938

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- PACKINGS

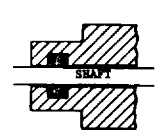
REFERENCE / NO.	R.B. Resek July 1956 AD97258					
INFORMATION BASED ON TESTS RECOMMENDATION	x	x	x	x	x	x
TEMPERATURE °F MAX. RANGE	80	400	30		400	
PRESSURE PSIA MAX. RANGE	Varying 200 to 3000		25	3000	10	3000
VELOCITY FPM	30 CPM (6" Travel)					
MATERIAL SEAL	Ring Packing (MX4439 Cumpac Seal) - Johns Manville - 95% Asbestos with Inconel Wire Mesh Core					
MATE						
FLUID	Hydraulic Fluid (OS-45)					
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm. cc/sec)	3.4 cc/1000 Cycles	2.6 cc/1000 Cycles	0	0	0	0 to 10 Drops per Hr.
TEST DURATION	5000 Cycles	25,000 Cycles	1 Hr.	1 Hr.	1 Hr.	1 Hr.
LIFE	3 Hrs.	14 Hrs.				
REMARKS	50 Lb-Ft of Loading		Hydraulic Actuator Rod (Tested Statically)			
DESIGN						

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- PACKINGS

I.C.1.a.(2)
Chart 2

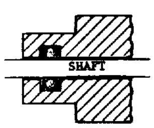
REFERENCE / NO.	F.R. Straus Jan. 1955 AD74089					
INFORMATION BASED ON TESTS RECOMMENDATION	x	x	x	x	x	x
TEMPERATURE °F MAX. RANGE	300	-40	-40	115	400	-65
PRESSURE PSIA MAX. RANGE	3000	25	3000			Both 20 and 3000
VELOCITY FPM	30 cpm (6 1/2" stroke)			30 cpm (6 1/2" stroke)		
MATERIAL SEAL	Neoprene WRT with Teflon Backups 1.5" ID	AN6227-28 O-Ring: Neoprene WRT 1.5" ID. With Spiral Teflon Back-up Rings.		AN6227-28 O-Ring: Buna N 1.5" ID. With Spiral Teflon Back-up Rings.	AN6227-28 O-Ring: Neoprene WRT. With Spiral Teflon Back-up Rings	
MATE						
FLUID	Hydraulic Oil MLO-8200			MIL-O-5606	MLO-8200	
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm. cc/sec)	2 cc Total	0	25 cc Total	One Seal = 5 cc Total; Second Seal = 38 cc Total	Satisfactory	None
TEST DURATION	40,000 Cycles	20 Hrs.	1 Hr.	50,000 Cycles		
LIFE						
REMARKS	Diametral Clearance = 0.0037"	Static Leak Test After Exposure to 40,000 Cycles at 300°F			Diametral Clearance = 0.0005" With Brass Ring Back-up added to hold clos clearance.	Static Leak Test After Exposure to 50,000 Cycles at 400°F.
DESIGN						

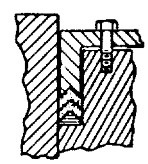


SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- PACKINGS

REFERENCE NO.	F.R. Straus Jan. 1955 AD74089 →		R.L. Hayman Jan. 1945 H-023	J.N. Smith Jan. 1961 S-070 →		
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	350 to 500	550		200 -65 to 180	500 -65 to 400	-20 to 600
PRESSURE PSIA MAX. RANGE	0-3000 →		3000	125,000	5000	8000
VELOCITY FPM	30 cpm (6 1/2" Stroke) →					
MATERIAL SEAL	AN6227-28 O-Ring: Acrylate	AN6227-28 O-Ring: Hycar-PA21	Synthetic Rubber "O"-Ring (AN-HH-P-114) with a Leather Back-up Ring on each side.	Leather "V"-Ring →	Synthetic Rubber (Homogeneous)	Fabric and Synthetic Rubber (Fabricated)
MATE				Metal →		
FLUID	Hydraulic Oil → MLO-8200 →			Oil, Water, Air	Oil, Water, Air, Steam, Solvents, Acid, Alkalies.	
SURFACE FINISH (μ in. CLA)				Maximum Roughness for Metal Surfaces → 63	16	32
LEAK RATE (Atm. cc/sec)	2 cc Total	None	Satisfactory			
TEST DURATION			Minimum - 20,000 Cycles			
LIFE	14,000 Cycles	2 Hrs.				
REMARKS	Diametral Clearance = 0.002". With Brass Ring Back-up Added to Hold Close Clearance.	Diametral Clearance = 0.0013".	Packing Groove is Rectangular in Shape with 10 degree Sloping Sides. Diametral Clearance = 0.015". Negligible "O"-Ring Wear. Table of Leather Ring Sizes included in the Report.	Number of Rings used varies with Pressure Applied. →		Alkalies-Just Fair
DESIGN						

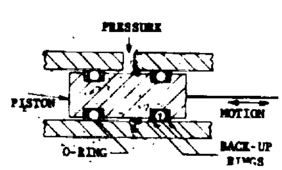




SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- PACKINGS

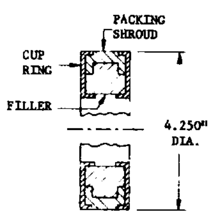
REFERENCE NO.	C.E. Hamlin March 1954 AD29114 →				L.E. Doughty June 1961 AD267145
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	350	400	450	500	400 ±25
PRESSURE PSIA MAX. RANGE	3000 →				4000
VELOCITY FPM	40 → 2 cps with 2" stroke →				3 inch stroke at 25 CPM
MATERIAL SEAL MATE	Neoprene "W" O-ring with split spiral aluminum-bronze back-up rings on both sides. →				Type 302 Stainless Steel spring molded in Viton B O-Ring (Precision Rubber Products Comp. #19009) AN6227-28. Shore A = 85. MS28782-28 spiral teflon back-up ring.
FLUID	Orthosilicate-Ester Hydraulic Fluid-OS-45-1 →				MLO 8515 Hydraulic Fluid; Polyphenyl Ether.
SURFACE FINISH (μ in. CLA)					
LEAK RATE (Atm. cc/sec)	One seal had very low rate; other was zero	One seal = 21.5 cc; other = zero	No leakage →		No leakage at 400°F. Leaked 5 cubic inches per minute at 80°F.
TEST DURATION	100,000 Cycles →				50,000 cycles
LIFE			40,000 Cycles	19,607 Cycles	
REMARKS	Recommended for extended duty at 400F and limited duty at 550F → No ring damage	No unusual ring damage	O-rings became brittle	O-rings extremely brittle	Room temp. tests showed a significant superiority of the composite O-Ring seal over the conventional non-cored O-Ring seal. The degradation of elastomer properties at elevated temp. precludes the use of this type seal without back-up rings in high temp. dynamic application. Tests at 400°F indicate sealing performance comparable to conventional O-Ring seals.
DESIGN					

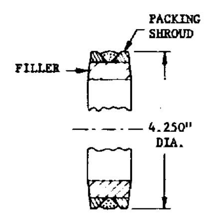


SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- PACKINGS

REFERENCE / NO.	F.K. Blessin et al, March 1961, AD259550L					
INFORMATION BASED ON TESTS / RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	-65; 75; 225					
PRESSURE PSIA MAX. RANGE	1100 At The Temperature of the Test					
VELOCITY FPM	3 to 4 CPM; Stroke 34 to 36 inches					
MATERIAL SEAL	Standard Shaped Packing with Cup Rings			Three Piece Shroud Without Cup Rings		
Filler Mat'l:	Nitrile Rubber	Silicone Rubber		Nitrile Rubber		Silicone Rubber
Shroud Mat'l:	Teflon		Filled Teflon	Teflon	Filled Teflon	Teflon
Packing Lubricators:	None	Polyester Felt	None	Polyester Felt	None	
MATE						
FLUID	Nitrogen and Hydraulic Oil (MIL-H-5606)					
Void Press. (N_2):	Atmos.	500 PSI	Atmosphere		500 PSI	
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm. cc/sec) -65°F:	797ML	750ML	1717ML	1675ML	700ML	12,415ML
75°F:	252ML	0	425ML	317ML	520ML	513ML
225°F:	95ML	0	95ML	77ML	196ML	810ML
TEST DURATION LIFE	8000 Cycles at -65°F and 75°F; 4000 Cycles at 225°F					
REMARKS	Investigation of Floating Piston Packings on 105MM Howitzer Recoil Mechanisms					
				Did Not Scratch the Recuperator Cylinder Wall		
DESIGN						

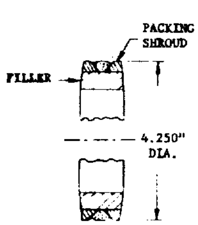


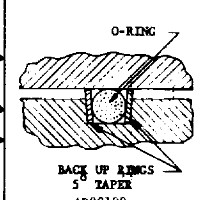


SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- PACKINGS

REFERENCE / NO.	F.K. Blessin et al, March 1961, AD259550L	E.R. Broadbent, April 1946, AD89199		
INFORMATION BASED ON TESTS / RECOMMENDATION	X	X	X	X
TEMPERATURE °F MAX. RANGE	-65; 75; 225	63 to 114		
PRESSURE PSIA MAX. RANGE	1100 at the temperature of the test.	15 to 3015		
VELOCITY FPM	3 to 4 CPM; Stroke 34 to 36 inches.	21 CPM		
MATERIAL SEAL	Three piece shroud without cup rings.	(New Packing)	(Aged Packing)*	
Filler Mat'l:	Silicone Rubber	"O"-Ring (Linear LT2-70 Compound)		
Shroud Mat'l:	Filled Teflon	Back-up ring-zero clearance-chrome retanned leather (0.062") Fed. Specification KK-L-231.		
Packing Lubricators:	Polyester Felt			
MATE		2 1/2" Piston		1 7/8" Rod
FLUID	Nitrogen & Hydraulic Oil (MIL-H-5606).	Oil		
Void Press. (N_2):	Atmos.			
SURFACE FINISH (μ in. CLA)				
LEAK RATE (Atm. cc/sec)	-65°F: 1887ML; 75°F: 383ML; 225°F: 64ML	613 cc (Total)	289 cc (Total)	1.9 cc (Total)
TEST DURATION LIFE	8000 Cycles at -65°F and 75°F; 4000 Cycles at 225°F	50,000 Cycles		
REMARKS	Investigation of floating piston packings on 105MM Howitzer Recoil Mechanisms. Did not scratch the recuperator cylinder wall.	5° Taper Groove. Back-up clearance equals 0.010"		
		*Packings soaked in AN-VV-0-3668 oil at 160°F for 7 days.		
		This packing possesses excellent life characteristics. At low temp., however, plastic and rubbers W8143 "O"-Rings show less leakage.		
		Oil leakage was low during operational tests and high during life cycling tests.		Leakage was low except during low temp. standbys and during warmup when it was extremely high. High leakage at the low temp. standbys was not satisfactory.
DESIGN		Friction values were not excessive.		





SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- PACKINGS

REFERENCE	E.R. Broadbent			R.E. Headrick		
NO.	Jan. 1946 AD89198			Oct. 1962 H-028		
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X*
TEMPERATURE °F MAX. RANGE	100			Room temp. to 1000 (5 cycles)	900	
PRESSURE PSIA MAX. RANGE	15 to 3015				1015	3015
VELOCITY FPM	21 CPM					
MATERIAL						
SEAL	(New Packing)	(Aged Packing)*		22% dense 430 stainless steel skeleton impregnated with a 16% indium-silver alloy.		
	"O"-Ring Plastic and Rubber Products Co.-Compound W-8143.					
	Leather back-up ring [Houghton Co. #2216 (0.093")unpressed].					
MATE	4" Piston	3 1/4" Rod				
FLUID	Oil					
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm. cc/sec)	579 cc (Total)	1404 cc (Total)	788 cc (Total)		Not excessive	
TEST DURATION	50,000 Cycles			114,000 cycles (30.5 hrs.)	250,000 cycles	25,000 cycles
LIFE						
REMARKS	To determine relative performance on large diameter struts compared with smaller diameter struts. *Packing soaked in AN-VV-0-3668 oil at 160°F for 7 days. Operation leakage at extreme temperatures (-65°F to +160°F) was nominal. Static breakout friction was exceptionally high at both extreme temperatures. Running friction was nominal at all temperatures. This leather "O"-Ring combination appears as satisfactory for the large diameter struts as for the smaller struts.				When evaluated as the first stage of a two-stage hydraulic piston-head seal.	*This test followed, and used the same equipment and seal as the test at 900°F and 1015 psia.
DESIGN	Rectangular groove with 5° tapered sides. Back-up clearance equals 0.010"					

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I.C.1.a.(2)
Chart 7

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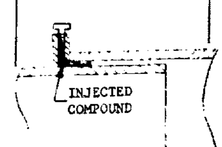
SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- PACKINGS

I.C.1.a.(2)
Chart 8

REFERENCE	W.J. Tocha et al				Anonymous	H.A. Bowman et al
NO.	April 1954 AD36456				Nov. 1932 A-053	July 1956 B-089
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE						
PRESSURE PSIA MAX. RANGE	2000				415	100,000 - 190,000*
VELOCITY FPM	12"-14" Stroke at 10 CPM				8 inch stroke at 10 strokes per hour*.	
MATERIAL						
SEAL	"V"-Ring Packings Green, Tweed & Co. (Palmetto 2520-100) Durometer "A" Hardness = 90 ± 5	Victor Manuf. Co. (Compound No.V352-2)	Henright Products Co. (Synthetic Rubber)	Garlock Packing Co. [9493(PHL53-15157 1/2)]	Semi-plastic, heat-resistant, composition whose principal ingredients are long fiber asbestos, graphite, and inorganic filler, impregnated with a high-test oil.	Teflon impregnated with 5% (by wt.) of Molybdenum disulfide. Copper-beryllium extrusion rings.
MATE						Hardened steel
FLUID	Oil (Navy symbol 2075)				Steam	High press. fluid
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm. cc/sec)	Did not exceed one (1) drop of oil per ten (10) cycles					
TEST DURATION LIFE	5000 Cycles				Equivalent of 20 yrs. Service	100 hrs. with load
REMARKS	Test of high pressure hydraulic packings for hydropneumatic counter-recoil systems of gun mounts. Volume of the packing did not change more than 7% while immersed in the oil for 70 hrs. at 158°F. Packings showed no evidence of adhesion, corrosion or pitting on brass, copper, monel metal or steel after being subjected to a mechanical force of 25 psi at 158°F for 168 hrs.				Steam expansion joint using a type of injection seal. *Run for 8 hrs. per day.	Pressure intensifier seal. *18 pressure applications. No signs of packing extrusion upon disassembly.
DESIGN						

GENERAL REFERENCES

DYNAMIC SEALS -- RECIPROCATING MOTION

CATEGORY I.C.1.a.(3)

1. Gillespie, L.H., How to Design with Teflon Packings, Applied Hydraulics and Pneumatics, V. 12, pp. 121-124 - July 1959, (G-022)

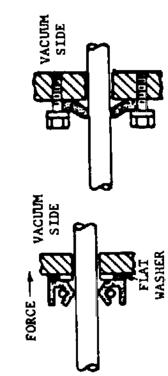
2. Symons, J.D., Engineering Facts About Lip Seals, SAE Automotive Eng. Congr., Detroit, Mich., Preprint 473A (14 pp.), Jan. 8-12, 1962, (S-140)

I.C.1.a.(3)
Chart 1

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- LIP OR RADIAL

REFERENCE / NO.	R.H.V.M. Dawton Oct. 1957 D-014	
INFORMATION BASED ON TESTS / RECOMMENDATION	X	X
TEMPERATURE °F MAX. RANGE	70	
PRESSURE PSIA MAX. RANGE	0 to 14.7	
VELOCITY FPM		
MATERIAL SEAL	Neoprene	
MATERIAL MATE	Buffed Steel	Stainless Steel
FLUID	Air	
SURFACE FINISH (μ in. CLA)		
LEAK RATE (Atm. cc/sec)	1.5×10^{-5} per stroke	4.5×10^{-5} per stroke
TEST DURATION / LIFE	Several thousand strokes	
REMARKS	Lubrication: Apiezon "B" Oil; Stroke = 2"	
DESIGN		

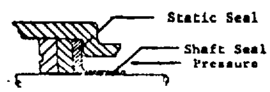


SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- LIP

REFERENCE	Hamlin, C. E. April 1959	
NO.	H010	
INFORMATION BASED ON TESTS RECOMMENDATION	X	X
TEMPERATURE °F MAX. RANGE	500°	
PRESSURE PSI MAX. RANGE	3000 PSI (Minimum)	
VELOCITY	2 Cycles/Sec. - 1 inch stroke - rectlinear motion	
MATERIAL		
SEAL	Ductle Iron (Meehanite)	
MATE	Nitrided Nitralloy - (3/4 inch diameter shaft)	
FLUID	Silicate Ester type	
SURFACE FINISH (μ in. CLA)	8 - 12 Microinches RMS	
LEAK RATE (Atm. cc/sec)	(Seal failure at 1/25 drop per cycle oil leakage)	
TEST DURATION LIFE	55000 cycles	90000 cycles
REMARKS	70% seal failure	30% seal failure
	Seals pressure energized	
	First leakage appeared at 15000 to 20000 cycles	

DESIGN - See sketch



SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- LIP

REFERENCE	Hamlin, C. E. April 1959	
NO.	H010	
INFORMATION BASED ON TESTS RECOMMENDATION	X	X
TEMPERATURE °F MAX. RANGE	Ambient	
PRESSURE PSI MAX. RANGE	3000 PSI (Minimum)	
VELOCITY	2 cycles/sec - 1 inch stroke - rectlinear motion	
MATERIAL		
SEAL	Ductle Iron (Meehanite)	
MATE	Nitrided Nitralloy - (3/4 inch diameter shaft)	
FLUID	Hydraulic oil	
SURFACE FINISH (μ in. CLA)	8 - 12 Microinches RMS	
LEAK RATE (Atm. cc/sec)	3 cc (Ave. Total)	5 cc(Total)
TEST DURATION LIFE	100000 cycles (5 sets)	700000 cycles (1 set)
REMARKS	Seal pressure energized	

GENERAL REFERENCES

DYNAMIC SEALS -- RECIPROCATING MOTION

CATEGORY -- I.C.1.a.(4)

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2. Cannizzaro, S., Lee, Jr., Schroder, R., Sealing in Severe Environments, National Conference on Industrial Hydraulics, V. XV, pp. 187-199, Oct. 1961, (C-008)

3. Elonka, S., Manual on Piston Rings, Power, V. 96, 7, pp. 103-26, July 1952, (E-017)

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8. Lee. J., Republic Has Metal Seals Working Well Up To 1000°F, SAE Journal, V. 70, No. 10, pp. 98-99, Oct. 1962, (L-010)

9. Niemeier, B.A., Seals to Minimize Leakage at Higher Pressure, ASME Trans., V. 75, pp. 369-379 - April 1953, (N-015)

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12. Hoblin, L.E., et al, Kinetic and Static Friction of the O-Ring Type Gasket, March 1952, D.D.C. Report AD 103928

13. Tipton, F.W., Design Data for O-Rings and Similar Elastic Seals, Nov. 1956, D.D.C. Report Supp. AD 110598

GENERAL REFERENCES

DYNAMIC SEALS -- RECIPROCATING MOTION

CATEGORY -- I.C.1.a.(4)

14. Trepus, G.E., Design Data for O-Ring and Similar Elastic Seals, Sept. 1957, D.D.C. Report AD 131094

15. Galloway, C.W. Evaluation of High Temperature Hydraulic Seals to Temp. of 550°F., Feb. 1958, D.D.C. Report No. AD142079

16. Tipton, F.W., Design Handbook for O-Rings and Similar Elastic Seals, Oct. 1959, D.D.C. Report Supp. AD 230639

17. Krivitsky, A.S. et al, Development of Self-lubricating and Non-adhering Elastomeric Materials, Oct. 1960, D.D.C. Report AD 254806

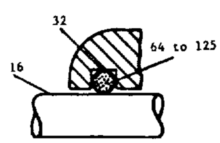
18. Walker, W.R., Design Handbook for O-Rings and Similar Elastic Seals, April 1961, D.D.C. Report AD 265443

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- RINGS

REFERENCE / NO.	S. Ivanami, et al Nov. 1959 I-012 →					A. MacCullen May 1960 AD241589
INFORMATION BASED ON TESTS / RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	122 →					275
PRESSURE PSIA MAX. RANGE	1437 →			72	157	3000
VELOCITY FPM	36 4" Stroke →	66	102 →			
MATERIAL SEAL	Acryle-Nitrile Rubber and Chloroprene JIS-Hardness #63 "O"-Rings →					Buna N "O"-Rings
MATE						
FLUID	Aircraft Hydraulic Oil →					Oronite #8515 Hydraulic Fluid
SURFACE FINISH (μ in. CLA)	Rod 0.4 to 0.8 →					
LEAK RATE (Atm. cc/sec)	2.7×10^{-4}	11×10^{-4}	20×10^{-4}	2.7×10^{-4}	8×10^{-4}	Not abnormal
TEST DURATION						260 Hrs.
LIFE						
REMARKS	Tests performed on rod and cylinder → 12 to 13% compression loading →					Performance satisfactory. Hydraulic system components subjected to a radiation dose of 10^9 ERG/Gm(C). See AD211733.
DESIGN						

I.C.1.a.(4)



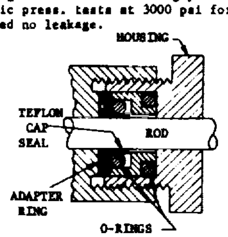
SURFACE FINISHES

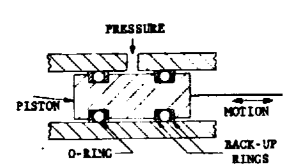
SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- RINGS

I.C.1.a.(4)
Chart 2

REFERENCE / NO.	J.O. Bruno Dec. 1957 AD118213	C.E. Hamlin March 1954 AD29114 →			
INFORMATION BASED ON TESTS / RECOMMENDATION	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	-68 to 300	350	400	450	500
PRESSURE PSIA MAX. RANGE	Press. Surge = 4500 Impulse Press. 0 to 3000 (30 cpm)	3000 →			
VELOCITY FPM	4" Stroke at 30 cpm	40 → 2 cps with 2" stroke →			
MATERIAL SEAL	1" AN6227-20 "O"-ring of variable composition as permitted by MIL-P-5516 and 0.031" teflon cap seal.	Neoprene "W" O-ring with split spiral aluminum-bronze back-up rings on both sides. →			
MATE	1" dia. rod of 440C stainless stl. (Rockwell C-55).				
FLUID	Hydraulic Oil (MIL-O-5606).	Orthosilicate-Ester Hydraulic Fluid-OS-45-1 →			
SURFACE FINISH (μ in. CLA)	18 RMS - Rod and inner surface of cap.				
LEAK RATE (Atm. cc/sec)	Less than 1 cc per 1000 cycles.	One seal = zero, other had very low rate	One seal = zero, other = 21.5 cc	No leakage →	
TEST DURATION	50,000 cycles	100,000 cycles →			
LIFE				40,000 cycles	19,607 cycles
REMARKS	Performed 43 tests. Cap ID = 0.007" less than rod diam. "O"-ring squeeze of 0.015" to 0.028" and side loading of up to 2° does not affect leakage. Radial extrusion gap = 0.003". Static press. tests at 3000 psi for 1 hr. showed no leakage.	Recommended for extended duty at 400F and limited duty at 550F →			
DESIGN					

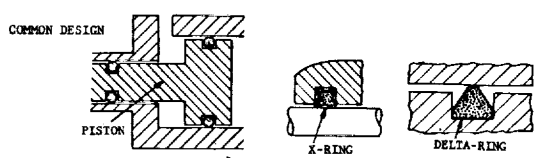




SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- RINGS

REFERENCE NO.	M.H. Everett Jan. 1961 E-034					R.L. Hayman Jan. 1945 H-023
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	-20 to 350	-80 to 250	-65 to 225	-100 to 350	600 -40 to 450	
PRESSURE PSIA MAX. RANGE	See Remarks					3000
VELOCITY FPM						
MATERIAL SEAL	Polyacrylate - Acrylic Acid Ester Polymer	Butyl Rubber	Polysulfide Polymer	Fluorosilicone Polymers	Hexafluoropropylene and Vinylidene Fluoride Copolymers	Synthetic Rubber "O"-Ring (AN-HH-P-114) with a leather back-up ring on each side.
MATE	Metal					
FLUID	Oil, Ozone	Hyd. Fl. of Phosphate-Ester Gases, Ozone, Veg. Oils	Ozone, Pet. Base Fuels & Oils, Aromatic Solv.	Pet. Based Fuel & Lub., Ester Type Lub., Hydrocarbons (Many)	Great Many Fluids and Chemicals	
SURFACE FINISH (μ in. CLA)	32 to 125 on Groove Sides; 16 on Diameter over which the O-Ring must slide					
LEAK RATE (Atm. cc/sec)						Satisfactory
TEST DURATION						Minimum - 20,000 Cycles
LIFE REMARKS	5% Peripheral (Compression) Loading "O"-Rings are good up to a pressure of 5000 PSI (Varies with the hardness of the ring material and the clearances).					Packing Groove is rectangular in shape with 10 degree sloping sides. Diametral clearance = 0.015". Negligible "O"-Ring Wear. Table of leather ring sizes included in the report.
DESIGN						



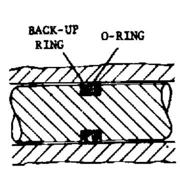
I.C.1.a.(4)
Chart 3

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- RINGS

I.C.1.a.(4)
Chart 4

REFERENCE NO.	F.R. Straus Jan. 1955 AD74089					
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	300	-40		115	400	-65
PRESSURE PSIA MAX. RANGE	3000	25	3000			Both 20 and 3000
VELOCITY FPM	30 cpm (6 1/2" Stroke)			30 cpm (6 1/2" Stroke)		
MATERIAL SEAL	Neoprene WRT Teflon Back-up 1.5" ID	AN6227-28 "O"-Ring Neoprene WRT; 1.5" ID With Spiral Teflon Back-up Rings		Buna N; 1.5" ID	Neoprene WRT	
MATE						
FLUID	Hydraulic Oil MLO-8200			MIL-O-5606	MLO-8200	
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm. cc/sec)	2 cc Total	0	25 cc Total	One Seal = 5 cc Total; Second Seal = 38 cc Total	Satisfactory	None
TEST DURATION	40,000 Cycles	20 Hrs.	1 Hr.	50,000 Cycles		
LIFE REMARKS	Diametral Clearance = 0.0037"	Static Leak Test After exposure to 40,000 cycles at 300°F.			Diametral Clearance = 0.0005" With brass ring back-up added to hold close clearance.	Static Leak Test After Exposure to 50,000 Cycles at 400°F.
DESIGN						

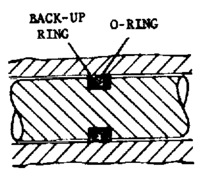


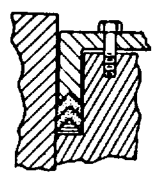
SEAL ANALYSIS

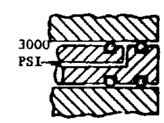
DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- RINGS

REFERENCE / NO.	F.R. Straus, Jan. 1955, AD74089 →		J.N. Smith, Jan. 1961, S-070 →			R.D. Ford, Feb. 1962, AD272431
INFORMATION BASED ON TESTS / RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. / RANGE	350 to 500	550	200 / -65 to 180	500 / -65 to 400	-20 to 600	160
PRESSURE PSIA MAX. / RANGE	0-3000 →		125,000	5000	8000	Cycle 0 to 3000
VELOCITY FPM	30 cpm (6 1/2" Stroke) →					100 cpm
MATERIAL SEAL	AN6227-28 "O"-Ring: → Acrylate	Hycar-PA21	Leather / "V" Ring →	Synthetic Rubber (Homogeneous)	Fabric and Synthetic Rubber (Fabricated)	Viton B
MATE			Metal →			
FLUID	Hydraulic Oil → / MLO-8200 →		Oil, Water, Air	Oil, Water, Air. Steam, Solvents, Acid, Alkalies.		Cellulube 220 (Phosphate Ester)
SURFACE FINISH (μ in. CLA)			Maximum Roughness for Metal Surfaces → 63	16	32	
LEAK RATE (Atm. cc/sec)	2 cc Total	None				Performance Satisfactory
TEST DURATION						>5,000,000 cyc.
LIFE	14,000 Cycles	2 Hrs.				
REMARKS	Diametral Clearance → = 0.002" / With brass ring back-up added to hold close clearance.	= 0.0013"	Number of rings used varies with pressure applied. →		(Alkalies-Just Fair)	Only slight "O"-ring wear

DESIGN







I.C.1.a.(4)
Chart 5

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- RINGS

I.C.1.a.(4)
Chart 6

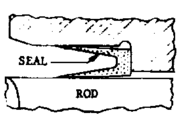
REFERENCE / NO.	A.E. Barrett, March 1961, AD252996 →					A.E. Barrett, July 1961, AD260465
INFORMATION BASED ON TESTS / RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. / RANGE	180	220	300 →			180
PRESSURE PSIA MAX. / RANGE	Cycle 0 to 3000 →					Cycle 0 to 3000
VELOCITY FPM	100 cpm (1 1/2" stroke) →					
MATERIAL SEAL	Viton B "O"-Ring → / 70 Shore A → / Teflon Back-up Ring →			90 Shore A →		Viton B "O"-Ring / 70 Shore A / Teflon Back-up
MATE						
FLUID	Cellulube 220 →				ASTM #3 Oil	Pet. Base and Phosphate Ester Fluids
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm. cc/sec)	0.08 cc/1000 cycles	0.10 cc/1000 cycles	Ave. 0.06 cc/1000 cycles	Ave. 0.25 cc/1000 cycles	0.08 cc/1000 cycles	Satisfactory
TEST DURATION / LIFE	133,000 cycles	104,000 cycles	Ave. 188,000 cycles	Ave. 75,000 cycles	Ave. 162,000 cycles	
REMARKS	No visible change in the "O"-rings. Back-up rings were worn and extruded.		Both "O"-rings and back-up rings were badly worn.	"O"-rings grooved but not flattened. Back-up rings worn and extruded.	"O"-rings grooved and flattened. Back-up rings worn and extruded.	

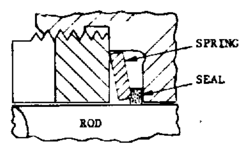
DESIGN

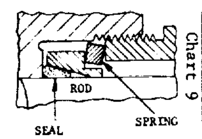
SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- RINGS

REFERENCE / NO.	E.T. Raymond April 1961 R-019 →					
INFORMATION BASED ON TESTS RECOMMENDATION	x	x	x	x	x	x
TEMPERATURE °F MAX. RANGE	400 →		500	Room	500	Room
PRESSURE PSIA MAX. RANGE	3000 →		4000			4000
VELOCITY FPM				30 to 80 cy/sec. →		
MATERIAL SEAL	Cup cross-section AISI-4340 steel.	Rectangular cross-section → AISI-52100; AISI-4340; Nitrided nitralloy steels →				Bronze; silver
MATE	Nitrided nitralloy rod →					
FLUID	OS-45 Hydraulic Fluid →		Versilube F-50	MIL-H-5606	Versilube F-50	MIL-H-5606
SURFACE FINISH (μ in. CLA)	4 to 8 RMS on rod	4 RMS on rod →				
LEAK RATE (Atm. cc/sec)	Outstanding sealing integrity.			No seal failures		
TEST DURATION LIFE	Excess of 200,000 cy.	Generally in excess of 100,000 cycles.	Approx. 20,000 cy.	3,600,000 cycles.	Approx. 600,000 cy.	Excess of 200,000 cy.
REMARKS	4 inch stroke. → Friction was very high during most of the life of the seal.	Seal was held in contact with gland face by means of a Belleville spring. →		0.04 inch stroke → No evidence of severe rod wear due to the seals.		Spring required to keep seal in contact with gland & rod.
DESIGN						







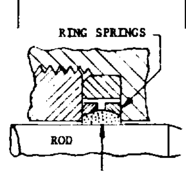
I.C.1.a.(4)
Chart 9

SEAL ANALYSIS

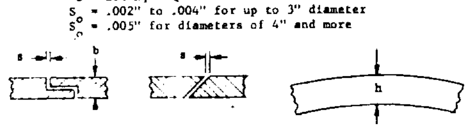
DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE -- RINGS

I.C.1.a.(4)
Chart 10

REFERENCE / NO.	E.T. Raymond April 1961 R-019	I.M. Rashta 1957 AD162882 →				
INFORMATION BASED ON TESTS RECOMMENDATION	x	x	x	x	x	x
TEMPERATURE °F MAX. RANGE	500					
PRESSURE PSIA MAX. RANGE	4000	2135	3560	3400 →		
VELOCITY FPM			1070 to 1600			
MATERIAL SEAL	AM355 Stainless steel ring springs; Rex AAA tool steel seal.	Piston ring 4" dia. → Cast iron Brinell hardness #170-220 →		Piston ring 2 3/4" dia. →		
MATE	*				With rubber O-Ring expander → For 1 ring	For 2 rings
FLUID	MLO-7277 Hyd. Fluid	Liquid	Hydraulic oil →			
SURFACE FINISH (μ in. CLA)	4 RMS on rod	High quality →				
LEAK RATE (Atm. cc/sec)	3 cc/hr. or less	.08 to .17 cc/sec		4.3 cc/sec	0.5 drops per sec.	0.05 drops per sec.
TEST DURATION LIFE	Greater than 100,000 cycles.		Long time			
REMARKS	4 inch stroke. *Rod is AM355 St.Stl flame plated with LW-1 tungsten carbide.	Principal sealing is performed by first ring and the balance by the second ring. More rings reduce wear and protect against faulty rings.	2 or 3 rings on piston	6 rings on piston. →		
DESIGN						



S = Free state gap
S_g = Gap when in cylinder
S = 3.4 h; b $\leqslant$ 0.75 h diam. = 16 to 24 h
S_o = .002" to .004" for up to 3" diameter
S_o = .005" for diameters of 4" and more



SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- RINGS

REFERENCE / NO.	S. Cannizzaro et al, Oct. 1961, C-008 →					
INFORMATION BASED ON TESTS / RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	+80 to +600 →					
PRESSURE PSIA MAX. RANGE	Pulsing Pressure of 65 to 3015 →			3015 →		
VELOCITY FPM	40 CPM (2 inch Stroke) →			240 CPM (1/8 inch Stroke) →		
MATERIAL SEAL	O-Ring Compound #1700-90 →; 2 back-up Rings →		O-Ring Comp. #XWAV-5	O-Ring Comp. #1700-90 →		O-Ring Comp. #XWAV-5
MATE	Piston Rod of Hardened Chrome Plated Steel →					
FLUID	Mineral Oil (MLO-7459 and MLO-60-294) →					
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm. cc/sec)	Failed →					Failed
TEST DURATION - Cycles				1,000,000 (79.5 hrs.)	776,064 (48.6 hrs.)	
LIFE - Cycles	173,682 (90.8 hrs.)	140,000 (60.7 hrs.)	105,760 (47.8 hrs.)			1,000,000 (90.8 hrs.)
CYCLES AT 600°F	152,813 (70.3 hrs.)	119,845 (52.0 hrs.)	73,187 (31.0 hrs.)	781,645 (61.0 hrs.)	638,584 (41.0 hrs.)	785,132 (61.3 hrs.)
REMARKS	Thermal Cycles → 9.5	7.5	4.0	9.0	6.5	10
	Thermal Cycling was Performed Concurrently with Mechanical Actuation →					
DESIGN	Gland Configuration: MIL-P-5514C-Modification B →					

I.C.1.a.(4)
Chart 7

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- RINGS

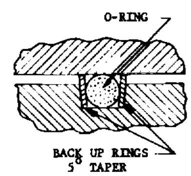
I.C.1.a.(4)
Chart 8

REFERENCE / NO.	W.R. Walker, April 1961, AD265443 →					
INFORMATION BASED ON TESTS / RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	400 →		-65 to 400 →			
PRESSURE PSIA MAX. RANGE	5000 →		4000 →			
VELOCITY FPM	392 cpm (1/4 inch stroke) →					
MATERIAL SEAL	AN6227-19 size O-ring (Viton A compound Number 9-0129). →; Spiral teflon back-up ring.	Single turn teflon back-up ring.	Viton A O-ring compound Number 9-0129. →; Back-up ring →; 20% silica filled teflon.	20% asbestos filled teflon.	Duroid 5613	20% copper filled teflon.
MATE	Rod →					
FLUID	MLO-8200 →		Versilube F-50 →			
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm. cc/sec)	* →		Trace of leakage below 0°F →			Trace of leakage at low temperature.
TEST DURATION	222,660	146,610	1,922,775	2,098,048	1,069,552	1,530,760
LIFE	*Cycles to failure →		Rod cycles →			
REMARKS	Vibrating the test fixture at 500 cps (±7G's) and at 1000 cps (±30G's) seemed to extend the life of the seal.		18 Thermal cycles →	19	10	13
DESIGN						

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- RINGS

REFERENCE NO.	E.R. Broadbent April 1946 AD89199	E.R. Broadbent Jan. 1946 AD89198
INFORMATION BASED ON TESTS RECOMMENDATION	X X X	X X X
TEMPERATURE °F MAX. RANGE	63 to 114 -65 to +160	100
PRESSURE PSIA MAX. RANGE	15 to 3015	15-3015
VELOCITY FPM	21 CPM	21 CPM
MATERIAL SEAL	(New Packing) (Aged Packing)* O-Ring (Linear LT2-70 compound) Back-up ring - zero clearance - chrome retanned leather (0.062) Fed. Specification KK-L-231.	(New Packing) (Aged Packing)* O-Ring (Plastic & Rubber Products Co. — Compound W-8143). Leather back-up ring [Houghton Co. #2216 (0.093) unpressed].
MATE	2 1/2" piston — 1 7/8" rod	4" piston — 3 1/4" rod
FLUID	Oil	Oil
SURFACE FINISH (μ in. CLA)		
LEAK RATE (Atm. cc/sec)	613cc (total) 289cc (total) 1.9cc (total)	579cc (total) 1404cc (total) 788cc (total)
TEST DURATION LIFE	50,000 cycles	50,000 cycles
REMARKS	*Packings soaked in AN-VV-O-3668 oil at 160°F for 7 days. Oil leakage was low during operational tests and high during life cycling tests. Leakage was low except during low temp. standbys & during warmup when it was extremely high.	To determine relative performance on large dia. struts compared with smaller dia. struts. *Packings soaked in AN-VV-O-3668 oil at 160°F for 7 days. Operation leakage at extreme temp. (-65°F to +160°F) was nominal. Static breakout friction was exceptionally high at both extreme temp. Running friction was nominal at all temperatures. This leather O-ring combination appears as satisfactory for the large dia. struts as for the smaller struts.
DESIGN	5° taper groove. Back-up clearance equals 0.010". Friction values were not excessive. High leakage at the low temp. standbys was not satisfactory. This packing possesses excellent life characteristics. At low temp., however, Plastic and Rubbers W8143 O-Rings show less leakage.	Rectangular groove with 5° tapered sides. Back-up clearance equals 0.010".



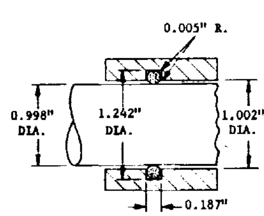
I.C.1.a.(4)
Chart 11

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- RINGS

I.C.1.a.(4)
Chart 12

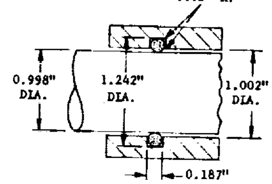
REFERENCE NO.	G.E. Trepus Sept. 1957 AD131094					
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	75	75	150	200	300	75
PRESSURE PSIA MAX. RANGE	5000					
VELOCITY FPM	46 CPM 4" stroke					
MATERIAL SEAL	O-Rings (AN6227-19)(0.984"ID x 0.139" wide) Philprene VP15	10 different nitrile rubbers (O-Rings)				3 different nitrile rubbers lubricated with petrolatum.
MATE						
FLUID	Velocite-K					
SURFACE FINISH (μ in. CLA)	10 (shaft)					
LEAK RATE (Atm. cc/sec)	Failure indicated by wetting of talc dust at shaft end.					
TEST DURATION LIFE	Cycles to failure: Max. - 5700 Ave. - 924	Max. - 1580 Ave. - 487	Max. - 890 Ave. - 304	Max. - 930 Ave. - 279	Max. - 320 Ave. - 66	Max. - 690 Ave. - 603
REMARKS	Ring lubricated with petrolatum. Best of 39 different materials. Physical properties listed in test data tables are not necessarily taken at the test conditions.	No lubrication.				
DESIGN						



SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- RINGS

REFERENCE / NO.	G.E. Trepus Sept. 1957 AD131094					L.R. Raisbeck April 1957 AD129444
INFORMATION BASED ON TESTS / RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	75			600	550	-70 to +160
PRESSURE PSIA MAX. RANGE	5000			3000	4000	880 to 1320
VELOCITY FPM	46 CPM 4" stroke			15 CPM	20 CPM	
MATERIAL SEAL	O-Rings (0.984"ID x 139" wide) 3 different nitrile rubbers lubricated with petrolatum AN6227-19.			AN6227-19 Viton-A with duroid 5600 back-up rings.		O-Ring (Goshen #1052) Approx. 5" dia.
MATE				Nitrided nitralloy rod		Metal
FLUID	Velocite-K			Silicone fluid G.E. Co. #81644.		Nitrogen
SURFACE FINISH (μ in. CLA)	10 (shaft)					63-80 RMS
LEAK RATE (Atm. cc/sec)	Failure indicated by wetting of talc dust at shaft end.			1 to 2 drops per cycle.	No significant leakage.	Satisfactory
TEST DURATION LIFE	Cycles to failure: Max. - 580 Ave. - 350	Max. - 760 Ave. - 370	Max. - 150 Ave. - 130	9,500	27,187	40,000 cycles
REMARKS	Sideloading of shaft causing .001" ring deflection	Misalignment of shaft.	Shaft vibrated at 60 cps causing .002" ring deflections.	Test jig clearance between rod and gland reduced to 0.0015".		
DESIGN						Packing lubricant: MIL-H-13866



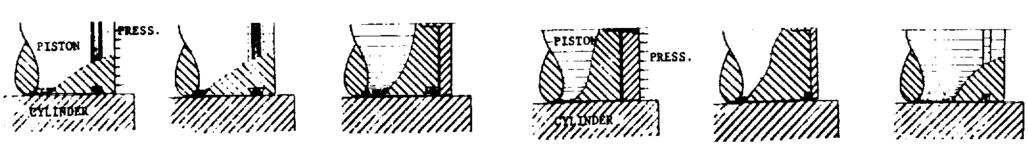
I.C.1.a(4)
Chart 13

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- RINGS

I.C.1.a.(4)
Chart 14

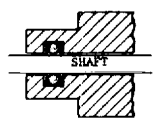
REFERENCE / NO.	B.A. Niemeier April 1953 N-015					
INFORMATION BASED ON TESTS / RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	95	91	90	84	86	
PRESSURE PSIA MAX. RANGE	45,765	37,515	39,015	21,000	13,745	
VELOCITY FPM						
MATERIAL SEAL						
MATE						
FLUID	Water					
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm. cc/sec)	No leaks		2 cc	5.5 cc	75 cc	
TEST DURATION (Seconds)	330	300	330	250	77	
REMARKS	Tested statically					
DESIGN	Pistons had a nominal 2 inch diameter. Normal O-Ring seal.	O-Ring in non-standard cavity.	Metal shrouded O-Ring.	A continuous ring similar to a K-ring but with a mechanical advantage member.	Continuous rectangular ring in a two-part piston.	A grooved piston with the groove filled with a visco-elastic material.



SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- RINGS

REFERENCE / NO.	F.H. Pollard Nov. 1956 Supp. 021 →					L.R. Raisbeck June 1960 AD240838
INFORMATION BASED ON TESTS / RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX.					450	
RANGE	75-450	75-500	75-550*	0-500*		-70 to +160
PRESSURE PSIA MAX.					3000	
RANGE	3015 in one direction; 65 in the other direction. →					945 to 1322
VELOCITY FPM					600 CPM (1/8" stroke)	
MATERIAL SEAL	Viton A O-Rings. Teflon loaded with ceramic fiber back-up rings. →					Quad rings-Minnesota Rubber & Gasket Co. Comp. # 388D. (Nitrile). Shore A. Hardness 70.
MATE	Hardened steel. →					
FLUID	Hydraulic oil OS45-1. →					Oil (MIL-H-13866)
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm. cc/sec)	Did not leak at room temp.	Failed. →				Did not fail.
TEST DURATION	76,000 mech. cycles at 450°F. 7 thermal cycles.	86,000 mech. cycles at 500°F.			246,000 cycles	40,000 cycles
LIFE			44,000 mech. cycles at 550°F. 38 thermal cycles.	91,000 mech. cycles		
REMARKS	4 inch stroke. →					Investigate the replacing of O-Rings with quad rings. Quad rings when used with 155MM gun carriage equilibrator: seal as well or better than O-Rings, have slightly higher frictions than O-Rings and eliminate spiral twisting.
DESIGN			*One thermal cycle for every 1000 mech. cycles at 550°F.	*1000 mech. cycles at 500°F; 1000 mech. cy. at 0°F & repeat. Cycling was continuous including heating & cooling phases.		



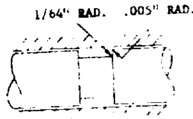
I.C.1.a.(4)
Chart 15

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- RINGS

I.C.1.a.(4)
Chart 16

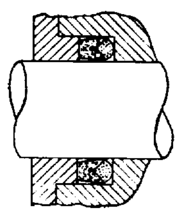
REFERENCE / NO.	D.E. Gerberich et al Aug. 1953 AD27728	W.J. Tocha et al April 1954 AD36456 →			
INFORMATION BASED ON TESTS / RECOMMENDATION	X	X	X	X	X
TEMPERATURE °F MAX. / RANGE	-65				
PRESSURE PSIA MAX. / RANGE	815	2000 →			
VELOCITY FPM	1/2" stroke at 10 CPM	12" - 14" stroke at 10 CPM →			
MATERIAL SEAL	#8 O-Rings (0.362"ID; 0.103" cross-section dia.)-Synthetic rubber: Parker PS-10-13N; Acushnet PX-1278-1; F.C. Wolfe; Stillman Rubber Co.; Vernay VL-123X5.	"V"-King packings → Greene, Tweed & Co. (Palmetto 2520-100) Durometer "A" Hardness = 90 ±5 →	Victor Manuf. Co. (Comp. # V352-2)	Henright Products Co. (Synthetic Rubber)	Garlock Packing Co. (9493-PHL53-15157 1/2)
MATE	Chrome plated cylinder and rod.				
FLUID	Iso-octane	Oil (Navy symbol 2075) →			
SURFACE FINISH (μ in. CLA)					
LEAK RATE (Atm. cc/sec)	No leakage.	Did not exceed one (1) drop of oil per ten (10) cycles. →			
TEST DURATION		5000 cycles →			
LIFE	20 to 120 minutes.				
REMARKS	This test was performed to ascertain a suitable O-Ring inclosure for a dynamic seal to be used with aviation fuel. Before testing all rings were soaked in 30% aromatic fuel for 24 hrs., then oven dried for 168 hrs. at a temp. of 158°F.	Test of high pressure hydraulic packings for hydropneumatic counter-recoil systems of gun mounts. Volume of the packing did not change more than 7% while immersed in the oil for 70 hrs. at 158°F. Packings showed no evidence of adhesion, corrosion or pitting on brass, copper, monel metal or steel after being subjected to a mechanical force of 25 psi at 158°F for 168 hrs.			
DESIGN	Diametral clearance between rod and cylinder = 0.001". A rectangular inclosure with 18% squeeze was used.				



I.C.1.a.(4)
Chart 17

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- RINGS

REFERENCE	Anonymous June 1955	F.W. Tipton Nov. 1956
NO.	A-147	Supp. AD110598
INFORMATION BASED ON TESTS RECOMMENDATION	X	X
TEMPERATURE °F MAX. RANGE		
PRESSURE PSIA MAX.		315*
RANGE	0 to 50,000 each stroke	
VELOCITY FPM	1 inch stroke	2.75" stroke
MATERIAL SEAL	U-Ring of synthetic rubber impregnated fabric with synthetic rubber molded into the recess.	AN6227-28 size O-ring. Butadiene-acrylonitrile polymer, 3300 series comp. #24.
MATE		
FLUID	Hydraulic Fluid	Velocite K
SURFACE FINISH (μ in. CLA)		Cylinder Surf. 5-10 RMS
LEAK RATE (Atm. cc/sec)		
TEST DURATION LIFE	71,000 strokes before failure.	20,000 cycles to failure.
REMARKS		Piston Seal. *Press. cycled 35 times/min. Best of 11 tested.
DESIGN		

I.C.1.a.(4)
Chart 18

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- RING

REFERENCE	Galloway, C. W. February 1961
NO.	G003
INFORMATION BASED ON TESTS RECOMMENDATION	X
TEMPERATURE °F MAX. RANGE	-65°F to 150°F
PRESSURE PSI MAX. RANGE	0 - 3000 PSI
VELOCITY	15 CPS at 0.35 inches peak to peak
MATERIAL SEAL	Teflon slipper seal with square rubber load ring
MATE	Steel
FLUID	Phosphate ester
SURFACE FINISH (μ in. CLA)	
LEAK RATE (Atm. cc/sec)	Excellent leakage and life characteristics
TEST DURATION LIFE	2,400,000 cycles
REMARKS	90 times longer life than AN 6227 O-Ring. Used on DC8 hydraulic damper cylinders.

I.C.1.a.(4)
Chart 19

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SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- RING (WEDGE)

REFERENCE / **NO.**	John Lee April 1962 L010	
INFORMATION BASED ON **TESTS** **RECOMMENDATION**	X	X
TEMPERATURE °F **MAX.** **RANGE**	800°F	1000°F
PRESSURE PSI **MAX.** **RANGE**	4000 PSI	4000 PSI
VELOCITY FPM	1/8" stroke at 350 CPM	
	2" stroke at 35 CPM	
MATERIAL **SEAL**	Graphite Material	Metallic Composite
MATE	Chrome Plate Rod	
FLUID	Polyphenyl Ether	
SURFACE FINISH		
LEAK RATE (Atm. cc/sec)	Negligible	Negligible
TEST DURATION **LIFE**	19 Hours	17 Hours
REMARKS	Seal Material developed by Armour Research Foundation. The composites consist of metal fibers which are sintered, brazed, or mechanically interlocked to form a porous skeleton, and then are filled with a softer metal.	

DESIGN

I.C.1.a.(4)
Chart 20

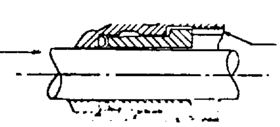
101

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- RING ("C")

REFERENCE / **NO.**	John Lee April 1962 L010	
INFORMATION BASED ON **TESTS** **RECOMMENDATION**	X	X
TEMPERATURE °F **MAX.** **RANGE**	600 F	1000 F
PRESSURE PSI **MAX.** **RANGE**	3000 PSI	
VELOCITY	1/8" stroke at 350 CPM	
	2" stroke at 35 CPM	
MATERIAL **SEAL**	Metallic Composite	
MATE	1/2" dia. chrome plate	1-1/2" dia. flame-plated Inconel X
FLUID	Dewaxed mineral oil	Polyphenyl ether
SURFACE FINISH		
LEAK RATE (Atm. cc/sec)	Good sealing at high pressure, leakage at low pressure.	
TEST DURATION **LIFE**	33000 cycles	324 hours
REMARKS	Metallic composite developed by Armour Research Foundation.	

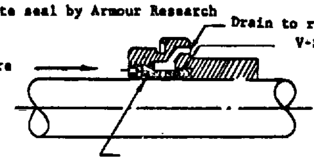
DESIGN



SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- RINGS (With V)

REFERENCE NO.	Lee, John April 1962 L010		
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X
TEMPERATURE °F MAX. RANGE	Ambient to 1000°F	670°F (Fluid)	-65°F to +650°F
PRESSURE PSI MAX.	4000 PSI 1st stage - rings → 80 PSI 2nd stage - V →		
VELOCITY	1/8" stroke at 350 CPM → 2" stroke at 35 CPM →		
MATERIAL			
SEAL	Metallic contracting ring and "V" shape composite ring - metallic		Conventional Mat'l
MATE	Inconel X		
FLUID	Polyphenyl ether		MLO 60-294 Dewaxed Mineral Oil (Amb. to 650F) MIL-H-5606 Hydraulic Fluid (Low Temp.)
SURFACE FINISH			
LEAK RATE (Atm. cc/sec)	No leakage	No leakage	No leakage at -65F tests Negligible - usually at heat-up of each thermal cycle
TEST DURATION	34 Hours	11 Hours Cycling	537, 586 cycles total 84 hours. 12 thermal cycles
REMARKS	Seals subjected to three thermal excursions from room temperature to 1000°F Metallic composite seal by Armour Research		
DESIGN - See sketch			



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SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- O-RING

REFERENCE NO.	Green, J. June 1963 Supp 069				
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X
TEMPERATURE °F MAX.	160 →				
PRESSURE PSI MAX. RANGE					
VELOCITY	3600 CPM - 1/2 inch stroke →				
MATERIAL Formulation Nos.	① 613-1, 4-Polybutadiene Compound	② Butyl Compound	③ Butyl Compound	⑤ Ethylene-Propylene Rubber Compound	⑥ Hydropol Compound
	Thiokol - RMD Dynamic O-Ring Tester →				
FLUID	50/50 Hydrazine - UDMH →				
SURFACE FINISH					
LEAK RATE (Atm. cc/sec)					
TEST DURATION LIFE	10 000 cycles →				
REMARKS	Little surface wear except No. 2 which exhibited excessive wear. → No Leakage data taken				

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I.C.1.a.(4)
Chart 23

SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- RECIPROCATING
SEAL TYPE --- O-RING

REFERENCE NO.	Green, J. June 1963 Supp 069		
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X
TEMPERATURE °F MAX. RANGE	Amb. and 160°F →	Amb. and 160°F →	Amb. →
PRESSURE PSI MAX.	250	250	250
VELOCITY	3600 CPH - 1/2 inch stroke →		
MATERIAL SEAL	Peroxidecured Polyethylene	EPR Composition 5	Butyl Composition 10
	(Tests conducted on Thiokol - RMD Dynamic "O" Ring Tester) →		
FLUID	Liquid N_2O_4 →		
SURFACE FINISH (μ in. CLA)			
LEAK RATE	None	Satisfactory	Liquid Leak
TEST DURATION LIFE	10 000 cycles	10 000 cycles	7 000 cycles

II
STATIC SEALS

GENERAL REFERENCES

STATIC SEALS

CATEGORY II

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2. Bridgman, P.W., Methods of Handling Very High Pressures, Comp. Air Mag., V. 27, pp. 17-19, Jan. 1922, (B-099)

II.A.1.a.

STATIC SEAL

ELASTIC DEFORMATION

GENERAL REFERENCES

STATIC SEALS -- SELECTION AND CONTROL OF GEOMETRY

CATEGORY II.A.1.a

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CATEGORY II.A.1.a.(1)

1. Anonymous, Seals Against Fluid Pressure, Part II; Materials Used as Principal Constituents of Packing and Gasket, Power Plant Engineering, V. 40, No. 6, pp. 354-6, June 1936, (A-064).

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3. Anonymous, Evaluating Pressure Sealing Properties of Rubber and Rubber-Like Materials, ASTM D-1081-60, (Supp. 030).

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12. Whalen, J. J., Select and Apply Gaskets Effectively, Chemical Engineering - V. 69, No. 20, pp. 83-88, Oct. 1, 1962, (W-041).

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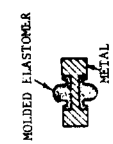
14. Bloomquist, C. R., Gasket Seals, Sept. 1956, D.D.C. Report AD 121339.

15. Walker, W. R., Design Handbook for O-Rings and Similar Elastic Seals, April 1961 - D.D.C. Report AD 265443.



STATIC SEAL
APPLICATION -- ELASTIC DEFORMATION
SEAL TYPE --- GASKETS

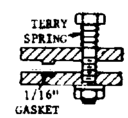
SEAL ANALYSIS

REFERENCE NO.	INFORMATION BASED ON TESTS / RECOMMENDATION	TEMPERATURE °F MAX. RANGE	PRESSURE PSIA MAX. RANGE	MATERIAL SEAL	MATE	FLUID	SURFACE FINISH (μ in. CLA)	LEAK RATE (Atm. cc/sec)	TEST DURATION LIFE	REMARKS	DESIGN
J. Jordon et al April 1960 J-027	x	-80 to 300		Buna N		Oil and Liquid Fuel			450F (60 to 70 Hrs.)		
	x	-65 to 350		Neoprene		Water, Steam and Ozone			450F (80 Hrs.)		
	x	-80 to 275		Butyl		Hyd. Oils, Phos. Ester					
	x			Rubber Nat.		Water					
	x	-180 to 500		Silicone		Hot Air, Alcohols, Phos. Esters			450F (100 Hrs.)		
	x	-65 to 450		Viton A		Hydrocarbons, Fuels, Lub. Oil, Disebacate Esters, Siloxane			450F (95 Hrs.)		

SEAL ANALYSIS

STATIC SEAL
APPLICATION -- ELASTIC DEFORMATION
SEAL TYPE --- GASKETS

REFERENCE / NO.	Anonymous Sept. 1949 Supp. 002	→	→	→	R.H. Dawton Oct. 1957 D-014	→
INFORMATION BASED ON TESTS / RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX.	194	→	→		572	392
RANGE				-58 to +122	70 to 572	70 to 392
PRESSURE PSIA MAX.				50		
RANGE	29.7	14.7	24.7		0 to 14.7	→
MATERIAL SEAL	Langite Cork-Grade H.P.	Neo-K-Tex (Rubber Bonded Cork)	Duram (Rubber Bonded Cork)	Langite Cork-Grade H.P.-Varnished with Glyptal 2458	Silastic	→
MATE	Metal Tank Flange	→	→	→	Metal	→
FLUID	Transformer Oil	→	→	Air	Air	→
SURFACE FINISH (μ in CLA)					Smooth "Γ" Emery	→
LEAK RATE (Atm. cc/sec)	No Leakage	→	→	None		
TEST DURATION	10 days	7 days	20 Hrs.			100 Hrs.
LIFE					70 Hrs.	
REMARKS	Gaskets intended for use in oil filled tanks for electrical gear. Loaded sufficiently to produce a 50% thickness reduction.		18 to 30% thickness reduction			
DESIGN						



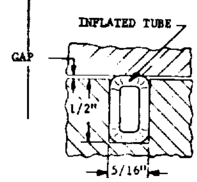
SEAL ANALYSIS

STATIC SEAL
APPLICATION -- ELASTIC DEFORMATION
SEAL TYPE --- GASKETS

REFERENCE / NO.	A.E. Barrett Sept. 1956 AD145361	→	→	Anonymous Sept. 1954 AD41652	M. Sabanas April 1960 AD238020	J.S. Islinger June 1956 AD106754
INFORMATION BASED ON TESTS / RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX.				400 (applied between spray and submersion tests)	1000	
RANGE					*	-85 to 200
PRESSURE PSIA MAX.	10,000	5,000	3,000	Atmospheric		12 psi differential
RANGE					Pulsating 0 to 5000	
MATERIAL SEAL	Hycar 1041 plus Hycar 1312 (Acrilonitriles) in pipe and tube union.			Silastic #675	Molybdenum Fibers impregnated with Ag	Silicone Rubber
MATE				Terminal Box		
FLUID	Petroleum Base (MIL-O-5606) or Phosphate Ester (Cellulube).		Air	Water spray and water submersion	Air	
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm. cc/sec)	Good Seal	→	→	No Leak	Good sealing ability	Seals out moisture, air and rain.
TEST DURATION	1/2 Hr.	60 days				1000 open and close cycles at 75F and 25 at -65F.
LIFE						
REMARKS	Preconditioned at 220F in test fluid for 36 days.		Preconditioned 36 days, 220F in air.	Used in rocket launcher terminal box.		Seal is inflatable aircraft canopy seal. Inflatable seal appeared more effective than other types tested.
DESIGN						

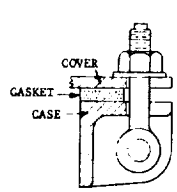
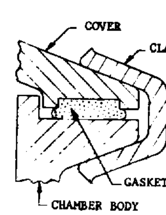
SEAL ANALYSIS

STATIC SEAL
APPLICATION -- ELASTIC DEFORMATION
SEAL TYPE --- GASKETS

REFERENCE NO.	M.B. Donald et al 1957 D-056	T.H. Hake April 1953 AD23155 →	→	→	O.C. Holderer et al July 1956 AD107960	J.D. Moffatt et al July 1961 AD272493
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	Approx. 70					-322 to +20
PRESSURE PSIA MAX. RANGE	1600	3015 for 8 hrs. and 14.7 for 16 hrs. in each 24 hr. period →	→	→	0 to 600	100 microns to 415 psia
MATERIAL SEAL	Rubber Bonded Compressed Asbestos Fiber (Klingerit)	Teflon (Circular crossection)	Teflon Rectangular crossection → Shore A = 100+ →	Nylon	Surgical Rubber (3/8" OD - 3/32" wall)	John L. Dore Co.; Fluorogreen (composition of Teflon, glass & non-organic coloring).
MATE	3" Dia. Metal Flanges	Metal (Union Joint) →	→	→	Metal	Stainless Steel
FLUID	Water	Air →	→	→		Liquid Nitrogen; Liquid oxygen.
SURFACE FINISH (μ in. CLA)		Regular Machine Finish →	→	→		
LEAK RATE (Atm. cc/sec)		No Leakage →	→	→	Chambers sealed	Zero
TEST DURATION LIFE		1475 Hrs. (Ave.)	1000 Hrs.	1550 Hrs.		
REMARKS	Shows effects of ring flange and gasket thickness on leakage pressure. Bolt stress may decrease with increase of system pressure. Gasket factors do not agree with ASME boiler and pressure vessel code section VIII 1962.	Loading: 40% Diametrical squeeze. The gaskets are intended for use in oxygen and air systems. The resistance of the gaskets to oxygen aging was determined by suspending three specimens of each type in an oxygen atmosphere for 200 hours at 158°F. The Teflon and Nylon gaskets were considered satisfactory. The test joints were continuously vibrated at 760 cpm at an amplitude of 3/32".	Loading: 20% Compression →	→	The seal is inflated tubing. Used for sealing access doors and wind tunnel parts.	Is suitable for cryogenic pipe complexes. Can be reused if initial torque is not satisfactory. Not fragile. Remains comparatively flexible in thin sections at cryo temp. Does not flake or chip. Will not chemically react with metal flanges. There is no fiber migration
DESIGN						

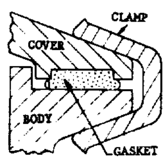
SEAL ANALYSIS

STATIC SEAL
APPLICATION -- ELASTIC DEFORMATION
SEAL TYPE --- GASKETS

REFERENCE NO.	C.F. Scharfenstein Jr. Nov. 1958 AD206773-L →	→	→	→	→	→
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE						
PRESSURE PSIA MAX. RANGE	29.4* →	→	→	→	→	→
MATERIAL SEAL	Non-curing Polysulphide (MMM Sealer EC-1126) 1/4" Dia. x 12" Long Strips	Neoprene cork composition (MIL-G-1223-3 Type II) → (Armstrong Cork Company Material No. DC-100) → 5/8" Thick, 1 3/8" Wide 24" ID →	1/2" Thick, 1 3/8" Wide	3/8" Thick, 1 3/8" Wide	3/16" Thick, 1/2" Wide 25 1/2" ID →	3/8" Thick, 1/2" Wide
MATE	Flat Pocket Flange (metal) and Mating Cover (steel) →	→	→	→	40°V-Band coupling with grooved top (steel)	→
FLUID	Air-Water →	→	→	→	→	→
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm.cc/sec)	No leak →	→	→	→	→	→
TEST DURATION LIFE	5 Minutes →	→	→	5 Min. and 10 Min.	5 Minutes	10 Minutes
REMARKS	Test for Sealing Buoy Pockets → *Buoy pocket was submerged in the test tank and 14.7 PSIG of air was applied internally. → Did not lose its flexible or compressible qualities while exposed to temperatures -10°F to 20°F.	→	→	→	→	→
DESIGN						

SEAL ANALYSIS

STATIC SEAL
APPLICATION -- ELASTIC DEFORMATION
SEAL TYPE --- GASKETS

REFERENCE NO.	C.F. Scharfenstein Jr. Nov. 1958 AD206773-L	C.F. Scharfenstein Jr. June 1961 Supp. 023	B.H. Swift Oct. 1961 AD265555 →			Anonymous July 1952 A-133
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE		40 to 75				250 to 1800*
PRESSURE PSIA MAX. RANGE	29.4*	29.4*	Held at 2715 for 2 hours then cycled from 15 to 2715 thirty times in rapid succession.			20 to 27 (internal)
MATERIAL SEAL	Neoprene cork composition (MIL-G-1223-3 Type II) (Armstrong Cork Company Material No. DC-100) 3/16" Thick, 1/2" wide, 24 3/8" ID	Alum. Flat Ring-24" ID x 30 1/2" OD x 0.119" Th.; Neoprene Ring-25 3/8" ID x 26" OD x 0.178" Th. x 0.625" wide.	MI 295-55 Double Lip Gasket →	10-M-1 (Hypalon 40 compound)	10-D-18 (Hycar 1072 Compound)	Spiral-wound 347 stainless steel (0.175" thick) with asbestos filler and stainless retaining ring.
MATE	40°V-Band coupling with ungrooved top (steel)	Flat Flange and cover	Metal →			
FLUID	Air-Water	Air-Water	Water →			Hot gases
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm.cc/sec)	No Leak	None	No Leakage →			Satisfactory
TEST DURATION LIFE	10 Minutes	5 Minutes	See above (Pressure) →			60 hrs.
REMARKS	Test for Sealing Buoy Pockets. *Buoy pocket was submerged in the test tank and 14.7 PSIG of air was applied internally	For sealing Buoy Pockets. *Buoy pocket was submerged in the test tank and 14.7 PSIG of air was applied internally	Also tested a single lip gasket which weeped slightly at low pressure (200 to 700 PSIG) but was leak tight while held at 2700 PSIG.		Also tested a single lip gasket which did not leak.	Approx. 10 inch diameter gasket. *Alternating cycle-30 minutes at 1800F & 30 minutes at 250F.
DESIGN						

SEAL ANALYSIS

STATIC SEAL
APPLICATION -- ELASTIC DEFORMATION
SEAL TYPE --- GASKETS

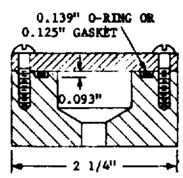
REFERENCE NO.	J.A. Parker et al Feb. 1961 AD267111 →			
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X
TEMPERATURE °F MAX. RANGE				
PRESSURE PSIA MAX. RANGE	14 →			
MATERIAL SEAL	Buna-N Gaskets (ID=2.5"; OD=3.74") → T-760*	T-761*	T-762*	T-764*
MATE	Serrated surface on one side-Flat surface on other side →			
FLUID	Nitrogen →			
SURFACE FINISH (μ in. CLA)				
LEAK RATE (Atm.cc/sec)	1 PSI/10 Min. →			
TEST DURATION LIFE				
REMARKS	*See report for actual composition →			
	← Flange Pressure (PSI) Required to Maintain the above Leak Rate →			
Dosage of Gamma Flux (ergs/g carbon) 0	490	250	530	620
8×10^9	800	880	2430	1950
2×10^{10}	---	620	3000	2275
DESIGN				

II.A.1.a.(1)
Chart 8

SEAL ANALYSIS

STATIC SEAL
APPLICATION -- ELASTIC DEFORMATION
SEAL TYPE --- GASKETS

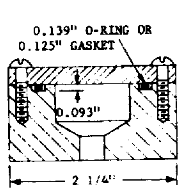
REFERENCE / NO.	E.A. Meyer et al Nov. 1953 AD41362 →					
INFORMATION BASED ON TESTS / RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. / RANGE	77 →					
PRESSURE PSIA MAX.	Vacuum system →					
RANGE	0-14.7 →					
MATERIAL SEAL	Vulcanized black rubber →			Cork Neoprene Armstrong DC100	DC167	DC100
MATE	Al to Al	Brass to Brass	Steel to Steel	Al to Al →		Brass to Brass
FLUID	Air & H_2 →					
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm. cc/sec) (per seal inch) (air)	$<3\times10^{-8}$ * 100	$<3\times10^{-4}$ 2	$<3\times10^{-8}$ 65	$<3\times10^{-4}$ 56	$<3\times10^{-4}$ 5	$<3\times10^{-8}$ 5
TEST DURATION LIFE						
REMARKS	*% of total seals tested which perform within the stated leak rate. A quantitative type leak detector was developed for this program. Seals preconditioned prior to leak testing by exposure to:- Pressure (0 to 30 psi) - 100% Rel. Hum.--Mechanical vibration & temperature cycling. →					
DESIGN						

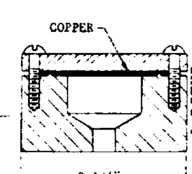


SEAL ANALYSIS

STATIC SEAL
APPLICATION -- ELASTIC DEFORMATION
SEAL TYPE --- GASKETS

REFERENCE / NO.	E.A. Meyer et al Nov. 1953 AD41362 →					
INFORMATION BASED ON TESTS / RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. / RANGE	77 →					
PRESSURE PSIA MAX.	Vacuum system →					
RANGE	0-14.7 →					
MATERIAL SEAL	Cork Neoprene(Armstrong) DC167 →	DC100	Johns-Manville Asbestos Fiber (Durabla)		1/16" thick copper	1/16" thick lead
MATE	Brass to Brass	Steel to Steel	Brass to Brass	Steel to Steel →		
FLUID	Air & H_2 →					
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm. cc/sec) (per seal inch) (air)	$<3\times10^{-4}$ * 2	$<3\times10^{-4}$ 60	$<3\times10^{-4}$ 12	$<3\times10^{-4}$ 16	$<3\times10^{-8}$ 95	$<3\times10^{-8}$ 50
TEST DURATION LIFE						
REMARKS	*% of total seals tested which perform within the stated leak rate. A quantitative type leak detector was developed for this program. Seals preconditioned prior to leak testing by exposure to:- Pressure (0 to 30 psi)-100% Rel. Hum.-Mechanical vibration & temperature cycling. →					
DESIGN						



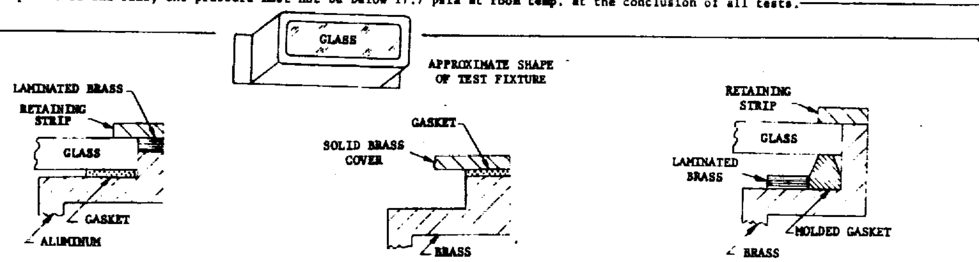


II.A.1.a.(1)
Chart 9

SEAL ANALYSIS

STATIC SEAL
APPLICATION -- ELASTIC DEFORMATION
SEAL TYPE --- GASKETS

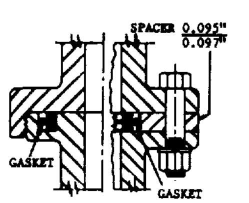
REFERENCE / NO.	E.L. Dickerson et al Oct. 1954 AD48807 →	→	→	→	→	→
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	See Remarks →	→	→	→	→	→
PRESSURE PSIA MAX. RANGE	Test fixture pressurized with 18.7 psia at room temperature at start of test* →	→	→	→	→	→
MATERIAL SEAL	Flat Gasket → Buna-N Rubber 1/32" thick; 80 Duro.	G.E. Silicone Rubber SE-360 1/16" thick; 70 Duro.	Neoprene Rubber 1/32" thick; 80 Duro.	Garlock Silicone Rubber #8773 1/16" thick; 80 Duro.	Molded Gasket → Hycar #OR-25	Neoprene Rubber 55 Durometer
MATE	Aluminum (Large Fixture) Glass; Laminated Brass →	Aluminum (Medium Fixture)	Brass (Large fixture)	Brass (Small Fixture)	Aluminum (Large fixture) Glass; Laminated Brass →	Aluminum (Small Fixture)
FLUID	Air →	→	→	→	→	→
SURFACE FINISH (μ in. CLA)	63 RMS →	→	→	→	→	→
LEAK RATE (Atm. cc/sec)	Passed all tests* →	→	→	→	→	→
TEST DURATION LIFE	14 days →	→	→	→	→	→
REMARKS	Evaluation of optical seals and optical sealing methods - conventional type static seals. Following are tests to which seals were subjected: (1) Fire hose test; (2) 24 hr. cold test at -20°F; (3) 24 hr. hot test at +165°F; (4) 72 hr. weathering test; (5) Vibration test (vert. and horiz.); (6) Shock test (vert. and horiz.). *For acceptance of the seal, the pressure must not be below 17.7 psia at room temp. at the conclusion of all tests. →	→	→	→	→	→
DESIGN						



SEAL ANALYSIS

STATIC SEAL
APPLICATION -- ELASTIC DEFORMATION
SEAL TYPE --- GASKETS

REFERENCE / NO.	Anonymous May 1957 A-178 →	→	Anonymous April 1961 A-207 →	→	→
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	To 1200	See Remarks	1202	70-1000	1000
PRESSURE PSIA MAX. RANGE	To 6000		15 to 5015 (pulsating) →	→	
MATERIAL SEAL	Inconel X (silver plated) → Wall thickness from 0.003" to 0.01". Dia. from 11/32" to 33 1/4".	→	Molybdenum fiber skeleton impregnated with silver. →	→	→
MATE					
FLUID		Ethylene oxide decomposition products.	Hot air →	→	Hydraulic fluid
SURFACE FINISH (μ in. CLA)					
LEAK RATE (Atm. cc/sec)		No leakage.	No leakage. →	→	→
TEST DURATION LIFE					93 hrs. (336,984 impulse cycles).
REMARKS	Grooves in seal are machined. The seal is generally reusable. →	→ 900°F for 3 min. 1200°F for 5 1/2 min. 1700°F for 5 min.	Stainless steel may also be used as a fiber skeleton. →	Temp. cycled 3 times. No apparent loss of bolt torque.	Included repeated assembly, disassembly & 12 thermal cycles.
DESIGN					



GENERAL REFERENCES

STATIC SEALS -- SELECTION AND CONTROL OF GEOMETRY

CATEGORY -- II.A.1.a.(2)

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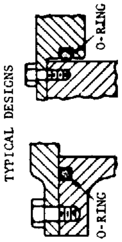
SEAL ANALYSIS

STATIC SEAL
APPLICATION -- ELASTIC DEFORMATION RINGS
SEAL TYPE ---

REFERENCE NO.	E.J. Parker Oct. 1957 P-010 →					
INFORMATION BASED ON TESTS / RECOMMENDATION	x	x	x	x	x	x
TEMPERATURE °F MAX. RANGE	-65 to 300	-65 to 160	-65 to 275	-65 to 170	-65 to 160 →	
PRESSURE PSIA MAX. RANGE						
MATERIAL SEAL	Natural Rubber	Neoprene	Buna N	Thiokol	Butyl	Buna S
MATE						
FLUID	Water, Steam, Alcohols, Silicons, Castor Oil	Refrigerants, CO_2	Pet. Lub. and Hyd. Oils, Pet. Fuels	Alcohols, High Aromatic Fuels	Phosphate Ester	Water, Alcohols, Ketones
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm. cc/sec)	Satisfactory →					
TEST DURATION						
LIFE						
REMARKS	$W/L_o = 4.8 \times 10^{-7} \times \partial^{1.2} \times \epsilon^{1.3} \times H^{4.5}$ (where $\epsilon < 0.15$) (Source I-012)			W = Compressive Load (Kg) ∂ = Material Diam. (CM) $\epsilon = \delta/\partial$		δ = Deformation Magnitude (CM) H = JIS Hardness L_o = Outer Ring Circumference (CM)
DESIGN						

TYPICAL DESIGNS

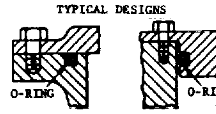


SEAL ANALYSIS

STATIC SEAL
APPLICATION -- ELASTIC DEFORMATION
SEAL TYPE --- RINGS

REFERENCE / NO.	E.J. Parker Oct. 1957 P-010 →					
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	500 -130 to 400	-20 to 325	-300 to 500	-320 to 250	-20 to 350	-20 to 500+
PRESSURE PSIA MAX. RANGE						
MATERIAL SEAL	Silicone	Acrylate	Teflon	KEL-F and Fluoroethane	KEL-F Elastomer	Viton A
MATE						
FLUID	Air, Inert Gases-High Analine Point Oils	Pet. Oils and Hydraulic Fluids	Various Chemicals →		Fuming Nitric Acid	Broad range of Fluids
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm. cc/sec)	Satisfactory →					
TEST DURATION						
LIFE						
REMARKS	See REMARKS on preceding page (II.A.1.a.(2) - Chart 1) →					
DESIGN						

TYPICAL DESIGNS

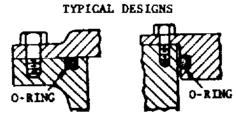


SEAL ANALYSIS

STATIC SEAL
APPLICATION -- ELASTIC DEFORMATION
SEAL TYPE --- RINGS

REFERENCE / NO.	E.J. Parker Oct. 1957 P-010 →		M.W. Varrell May 1957 AD140116	F.O. Rick Feb. 1953 AD11484 →		S. Iwanami, et al Nov. 1959 I-012
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	-20 to 400	-20 to 250		-65 to 75F →		
PRESSURE PSIA MAX. RANGE			1000			4000
MATERIAL SEAL	Poly-FBA	"V"yram	#73902-5 Ring Packing Plug	AF934 O-Ring →		"O"-Ring: Acryle-Nitrile Rubber and Chloroprene
MATE			Sonar Stuffing Tube			
FLUID	Diester Lub., Silicate Ester Hyd. Fluid, Organic Phosphates, Pet.	Phosphate Esters, Organic Phosphates, Pet.	Water	100% Isooctane	70% Isooctane-30% PhMe (Toluene)	Oil
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm. cc/sec)	Satisfactory →		Satisfactory	No Leakage →		No Leakage
TEST DURATION				37 Days →		
LIFE						
REMARKS	See REMARKS on preceding page (II.A.1.a.(2) - Chart 1).			O-Ring interconnects-Fuel Resistance Tests. No appreciable wear or damage to "O"-Rings and parts.		5 to 9% Squeeze
DESIGN						

TYPICAL DESIGNS

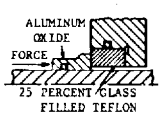


SEAL ANALYSIS

STATIC SEAL
APPLICATION -- ELASTIC DEFORMATION
SEAL TYPE --- RINGS

REFERENCE NO.	H.O. Pedersen et al Feb. 1959 AD225213 →		J.W. Hull Sept. 1960 H-079	M.H. Everett Jan. 1961 E-034	R.D. Ford Sept. 1957 AD145846 →	
INFORMATION BASED ON TESTS RECOMMENDATION	In Fixture →		X	X	X	X
TEMPERATURE °F MAX. RANGE	47 to 91	40 to 140	-65 to 700	-130 to 500	300	212
PRESSURE PSIA MAX. RANGE	25 to 160	15 to 25	4000 Surge to 6000			
MATERIAL SEAL	KEL "F" →		Tool and Stainless Steel	Silicone Polymers	Silicone W96 "O"-Ring →	
MATE	Metal →		Nitraloy, Tool Steel	Metal		
FLUID	90% H_2O_2 One Side N_2 On Other	90% H_2O_2 One Side Air On Other	Hydraulic Fluids	High Analine Oils, High Temperature Air, Ozone	ASTM Oil #2	ASTM Oil #3
SURFACE FINISH (μ in. CLA)			8 to 32	32 to 125 on groove sides. 16 on diameter over which the O-ring must slide.		
LEAK RATE (Atm. cc/sec)	Inlet Seal 0.08 to 1.33	Outlet Seal 0.075 to 1.26	0		Satisfactory Seal →	
TEST DURATION LIFE						
REMARKS	Used in conjunction with a rotary face seal at 16,000 rpm. Rocket Engine, Turbo-pump, Fluid Cooled Bearing Seal remained excellent but recommend KEL "F" O-rings be replaced with metal.		Design formulae and tables are given for metal face seals.	5% peripheral loading. "O"-rings are good up to a pressure of 5000 psi (varies with the hardness of the ring material and the clearances).	See Report AD145307 →	

DESIGN

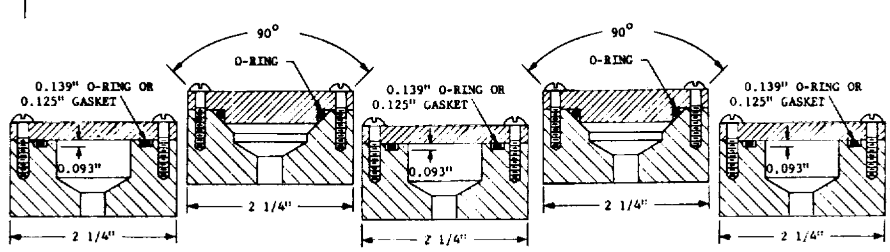


SEAL ANALYSIS

STATIC SEAL
APPLICATION -- ELASTIC DEFORMATION
SEAL TYPE --- RINGS

REFERENCE NO.	A. MacCullen May 1960 AD241589	E.A. Meyer et al Nov. 1953 AD41362 →				
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	275	77 →				
PRESSURE PSIA MAX. RANGE	3000	Vacuum System → 0 to 14.7 →				
MATERIAL SEAL	Buna N "O"-Rings	Buna N O-Ring (1.03" O.D.) →				
MATE		Al to Al →		Brass to Brass →		Steel to Steel
FLUID	Oronite #8515 Hydraulic Fluid	Air & H_2 →				
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm. cc/sec)	Not abnormal	$\leq 3x10^{-8}$ (per inch of seal) (air) → 95*	45	90	65	65
TEST DURATION LIFE	260 Hrs.					
REMARKS	Performance satisfactory. Hydraulic system components subjected to a radiation dose of 10^9 ERG/Gm(C). See AD211733.	*% of total seals tested which perform within the stated leak rate. → A quantitative type leak detector was developed for this program. → Seals preconditioned prior to leak testing by exposure to:- pressure (0 to 30 psi) - 100% Rel. Hum.-mechanical vibration & temperature cycling.				

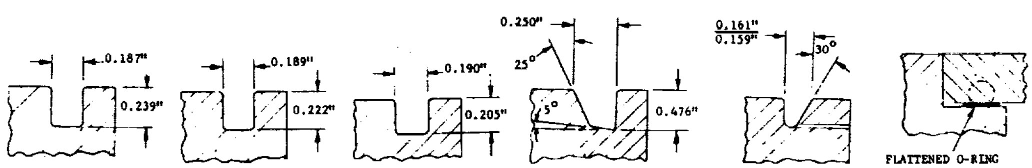
DESIGN



SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ELASTIC DEFORMATION
SEAL TYPE --- RINGS

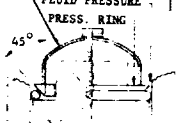
REFERENCE / NO.	D.E. Gerberich et al, Aug. 1953, AD27728 →					
INFORMATION BASED ON TESTS / RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	$-65^{+}0.5$ →					
PRESSURE PSIA MAX. / RANGE	All seals tested over all ranges shown: → Cycled from 115 to 515 twice; from 115 to 765 twice; from 115 to 1015 three times; from 115 to 1265 twice; from 25 to 2015 five times. All cycling was done at a rate of 1 cycle per hour.					
MATERIAL SEAL	#27 O-Rings (1.484" ID; 0.139" cross-section dia.) – Synthetic rubber: Acushnet PX-1278-1; Acushnet PX-1775; Goshen 1120; Parker PS-10-13N; Stillman Rubber Co.; Vernay VL-12X5.[The Parker, Stillman and Vernay compounds conform to spec. MIL-P-5315 (USAF)].					
MATE	Metal →					
FLUID	Iso-octane →					
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm. cc/sec)	Percentage of O-Rings tested which showed no leakage in this enclosure: → Presoaked = 50%	Presoaked = 60% Dry = 5%	Presoaked = 85% Dry = 15%	Presoaked = 75% Dry = 30%	Presoaked = 78% Dry = 15%	Presoaked = 60% Dry = 35%
TEST DURATION LIFE	Total = approximately 14 hrs. [See Pressure (above)] →					
REMARKS	This series of tests were performed to ascertain a suitable O-Ring enclosure for a static seal to be used with aviation fuel. → Before testing, all O-Rings were soaked in 30% aromatic fuel for 24 hours, and then oven dried for 168 hours at a temperature of 158°F. A thin layer of Petrolatum was used to assure proper assembly. Dry rings: aged rings installed in a dry condition. – Presoaked rings: aged rings soaked for 16 hrs. in iso-octane at room temp., just prior to installation in the test fixture. No. of tests = 191	No. of tests = 259	No. of tests = 357	No. of tests = 272	No. of tests = 151	No. of tests = 50
DESIGN	Standard rectangular enclosure with a nominal squeeze value of: → 12%	18%	24%	Enclosure with inclined bottom.	Enclosure with inclined side.	Plug type enclosure

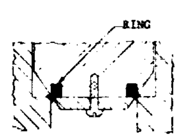


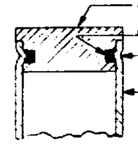
SEAL ANALYSIS

DYNAMIC SEAL
APPLICATION -- ELASTIC DEFORMATION
SEAL TYPE --- RINGS

REFERENCE / NO.	D.E. Gerberich et al, Aug. 1953, AD27728	G.F. Ervin, June 10, 1952, AD2400 →		S.R. Crockett, Oct. 1957, AD150474 →		
INFORMATION BASED ON TESTS / RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	$-65^{+}0.5$	$-65^{\pm}2$	$-65^{\pm}2$	-65	194	
PRESSURE PSIA MAX.	All seals tested over all ranges shown. See II.A.1.a.(2) Chart 6			Hydrostatic → 1500 (rise time = 0.015 sec) →		
RANGE		68 to 87	79 to 90			2200 to 3927
MATERIAL SEAL	See II.A.1.a.(2) Chart 6	Dwg. 165-58476 - Made from MIL-P-5516A Material.		Aluminum →		
MATE	Metal	Metal (Poppet valve seat) →				
FLUID	Iso-octane	Hydraulic fluid (MIL-O-5606) →		Liquid →		
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm. cc/sec)	Percentage of O-Rings tested which showed no leakage in this enclosure: Presoaked = 90% Dry = 55%	0 to 1 drop/2Min. After reseating →	0 to 1 drop/4Min.	19 units had no leaks; 1 unit leaked 3 to 5 cc/sec.	18 units had no leaks; 2 units leaked 3 to 5 cc/sec.	Leakage pressure for this group ranged as shown above.
TEST DURATION LIFE	Total = approximately 14 hrs. [See Pressure (above)]					
REMARKS	See II.A.1.a(2) Chart 6 No. of tests = 44			20 units preconditioned at -65F for 2 weeks.	Preconditioned at 194F for 30 days; same units previously tested at -65F.	10 units previously tested at -65F and 194F.
DESIGN	Enclosure with spring loaded O-Ring			O-Ring is a temporary expedient – a better sealing medium can be devised.		



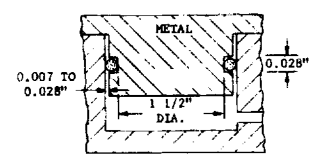




SEAL ANALYSIS

STATIC SEAL
APPLICATION -- ELASTIC DEFORMATION
SEAL TYPE --- RINGS

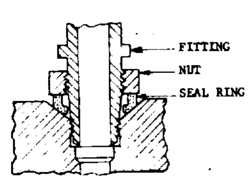
REFERENCE NO.	T.H. Hake April 1953 AD23155 →			G.E. Trepus Sept. 1957 AD131094 →		
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE						
PRESSURE PSIA MAX. RANGE	3000 for 8 hrs. and 14.7 for 16 hrs. in each 24 hr. period.			0 to 5000 (pulsed 37 CPM) →		
MATERIAL SEAL	Teflon (circular cross-section)	Teflon Rectangular cross-section Shore A = 100+ →	Nylon	3300-28 Nitrile rubber O-Ring (AN6227-23) →		
MATE	Metal (Union Joint) →			Metal →		
FLUID	Air →			Velocite-K →		
SURFACE FINISH (μ in. CLA)	Regular machine finish →			15 to 20	15 to 20	28 to 35
LEAK RATE (Atm. cc/sec)	No leakage →			Test failure indicated by wetting of talc dust on surface.		
TEST DURATION	1475 hrs. (Ave.)	1000 hrs.	1550 hrs.			
LIFE				69,660 pulses	4,030 pulses	3,510 pulses
REMARKS	Loading: 40% Diametrical squeeze.	Loading: 20% Compression →		Clearance = 0.007"	Clearance = 0.028" →	
	The gaskets are intended for use in oxygen and air systems. The resistance of the gaskets to oxygen aging was determined by suspending three specimens of each type in an oxygen atmosphere for 200 hrs. at 158°F. The teflon and nylon gaskets were considered satisfactory. The test joints were continuously vibrated at 760 CPM at an amplitude of 3/32".			O-Ring size AN6227-23 — Approx. 1.234"IDx0.139" wide →		
DESIGN						



SEAL ANALYSIS

STATIC SEAL
APPLICATION -- ELASTIC DEFORMATION
SEAL TYPE --- RINGS

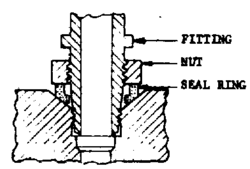
REFERENCE NO.	H.P. Kupiec April 1955 AD74322 →					
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE				400	550	75
PRESSURE PSIA MAX. RANGE	10,000 →			5000 →		20,000
MATERIAL SEAL	#1020 Steel Bar Stock (Ring) →					
MATE						
FLUID	Hydraulic Fluid (MIL-O-5606) →					
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm. cc/sec)	No leakage →					No rupture
TEST DURATION LIFE				5 minutes →		1 minute
REMARKS	Installation Torque → = 600 in. lbs. Seal Size #8	= 900 in. lbs. Seal Size #12 →	= 1200 in. lbs.	= 20 to 100 ft. lbs. → Metal boss seal for numbers 5,8,12 and 10056, 10057 fittings. 48 rings tested.	24 rings tested. →	
	Each fitting removed and reinstalled 15 times. → Seal ring is preset on fitting with tool. Tightening specifications: Data sheet and 10064 and MIL-F-5506A.					
DESIGN						



SEAL ANALYSIS

STATIC SEAL
APPLICATION -- ELASTIC DEFORMATION
SEAL TYPE --- RINGS

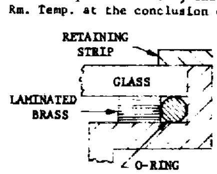
REFERENCE NO.	H.P. Kupiec April 1955 AD74322 →					→
INFORMATION BASED ON TESTS RECOMMENDATION	x	x	x	x	x	x
TEMPERATURE °F MAX. RANGE		75	-65			
PRESSURE PSIA MAX. RANGE	10,000	Cycled 0 to 7500 at 60 CPM	10,000 →			→
MATERIAL SEAL MATE	#1020 Steel Bar Stock - Ring →					→
FLUID	Hydraulic Fluid (MIL-O-5606) →					→
SURFACE FINISH (u in. CLA)						
LEAK RATE (Atm. cc/sec)	No leakage →					→
TEST DURATION LIFE	5 minutes	200,000 cycles	10 minutes			
REMARKS	Metal boss seal for numbers 5,8,12 and 10056, 10057 fittings. → Fittings vibrated at 1750 CPS at 1/8" to 1/4" amplitude.		Preconditioned at -65F for 32 hrs.	Installation torque → = 180 in. lbs. Seal Size #5 → Each fitting removed and reinstalled 15 times. →	-240 in. lbs.	= 450 in. lbs. Seal Size #8
DESIGN	Seal ring is preset on fitting with a tool. → Tightening Specifications: Data sheet and 10064 and MIL-F-5506A. →					→

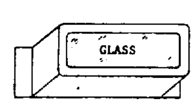


SEAL ANALYSIS

STATIC SEAL
APPLICATION -- ELASTIC DEFORMATION
SEAL TYPE --- RINGS

REFERENCE NO.	H.P. Kupiec April 1955 AD74322	E.L. Dickerson et al Oct. 1954 AD48807 →		→	W.R. Walker April 1961 AD265443 →	→
INFORMATION BASED ON TESTS RECOMMENDATION	x	x	x	x	x	x
TEMPERATURE °F MAX. RANGE	-65	See Remarks →		→	1000 →	→
PRESSURE PSIA MAX. RANGE	Cycled 0 to 5000	Test fixture pressurized with 18.7 psia at Rm. Temp. at start of test*			315 (applied only while checking seal)	
MATERIAL SEAL	Cold rolled steel	O-Ring → Tel. Mk. 72	Tel. Mk. 60	Tel. Mk. 50	Viton A O-Ring Compound #9-0129 69% Squeeze	58.5% Squeeze
MATE		Aluminum (large fixture). Glass; laminated brass →	Aluminum (medium fixture)	Brass (small fixture)		
FLUID	Air	Air →		→	Nitrogen →	→
SURFACE FINISH (μ in. CLA)		63 RMS →		→		
LEAK RATE (Atm. cc/sec)	No air bubble leaks at 10,000 psia	Passed all tests* →	→	See Remarks		
TEST DURATION LIFE		14 days →	→	8 days	10 minutes	2 minutes
REMARKS	Metal boss seal for numbers 5,8,12 and 10056, 10057 fittings.	Evaluation of optical seals and optical sealing methods. Following are tests to which seals were subjected: (1) fire hose test; (2) 24 hr. cold test at -20°F; (3) 24 hr. hot test at +165°F; (4) 72 hr. weathering test; (5) vibration test (vert. & horiz.); (6) shock test (vert. & horiz.). *For acceptance of seal, the press. must not be below 17.7 psia at Rm. Temp. at the conclusion of all the tests.			Flange seal test →	→
DESIGN	Seal ring is preset on fitting with a tool. Tightening Specifications: Data sheet and 10064 and MIL-F-5506A.	(see sketch) →		Failed after the hot test because of the porosity of the brass test fixture. (14.7 psia).		



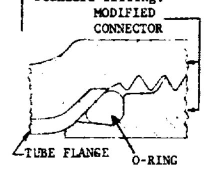


APPROXIMATE SHAPE OF TEST FIXTURE

SEAL ANALYSIS

STATIC SEAL
APPLICATION -- ELASTIC DEFORMATION
SEAL TYPE --- RINGS

REFERENCE	Anonymous April 1961 →			B.H. Hodder May 1958
NO.	A-207			H-058
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X
TEMPERATURE °F MAX. RANGE	1202	70-1000	1000	up to 500
PRESSURE PSIA MAX. RANGE	15 to 5015 (Pulsating) →			Vacuum or low pressure (under 1000 psig)
MATERIAL				
SEAL	Molybdenum fiber skeleton impregnated with silver. →			Neoprene or teflon O-Rings
MATE				
FLUID	Hot Air →		Hydraulic Fluid.	
SURFACE FINISH (μ in. CLA)				
LEAK RATE (Atm. cc/sec)	No Leakage. →			
TEST DURATION			93 hrs. (336,984 impulse cycles).	
LIFE				
REMARKS	Stainless steel may also be used as a fiber skeleton. →	Temp. cycled 3 times. No apparent loss of bolt torque.	Included repeated assembly, disassembly and 12 thermal cycles.	Modified standard refrigeration-type, flare tube fittings. Seals by hand tightening alone. Dimensions for modifying 1/4, 3/8, 1/2 and 5/8 in. fittings are included in cluded in the report. This modification removes a number of the disadvantages of the standard fitting.
DESIGN				



SEAL ANALYSIS

STATIC SEAL
APPLICATION -- ELASTIC DEFORMATION
SEAL TYPE --- O-RING

REFERENCE	Ludtke, P.R., and others 1962		
NO.	Supp 048		
INFORMATION BASED ON TESTS RECOMMENDATION	X (Continuous Test) →		
TEMPERATURE	Amb. to 76°K →		
PRESSURE PSIG MAX.	100 psig	500 psig	1000 psig
STATIC FORCE - Start - 76°K	11500 lbs. 4100 lbs.	8000 lbs. 1700 lbs.	7500 lbs. @ 123°K - 400 lbs.
MATERIAL			
SEAL	ASD Compound IV-29B of is -4 Polybutadiene →		
MATE	Steel →		
FLUID	Helium →		
SURFACE FINISH (μ in. CLA)	32 →		
LEAK RATE (Atm. cc/sec)	None	None	Leak occurred at 123°K Greater than .0003 Atm. cc/sec after 1 sec.
TEST DURATION LIFE	2 Hr. period	2 Hr. period	2 Hr. period
REMARKS	O-Ring 0.140 in. thick, 1 in. ID, compressed to 0.032 in. (77%) 23% compression set. 14000 lb. initial force, 29% loss before cool-down (Relaxation of elastomer) Three groups of elastomers tested (16 total) This material best of all elastomer samples tested At 14000 lb. initial force.		

SEAL ANALYSIS

STATIC SEAL
APPLICATION -- ELASTIC DEFORMATION
SEAL TYPE --- O-Ring

REFERENCE	Luotke, P.R., and others 1962		
NO.	Supp 048		
INFORMATION BASED ON TESTS RECOMMENDATION	X(Continuous Test) →		
TEMPERATURE MAX. RANGE	Amb. to 76°K →		
PRESSURE PSIG MAX. RANGE	100	500	1000
STATIC FORCE - Start - 76°K	13 000 Lbs. 8 500 lbs.	12 000 lbs. 7 900 lbs.	12 000 lbs. 7 400 lbs.
MATERIAL			
SEAL	ASD Compound IV - 8A of natural rubber		
MATE	Steel →		
FLUID	Helium →		
SURFACE FINISH	32 μ in. →		
LEAK RATE	None	None	None (Leak occurred when removed from liquid Nitrogen bath at 1250 psig Helium pressure.
TEST DURATION LIFE	2 Hr. period	2 Hr. period	2 Hr. period
REMARKS	O-Ring 0.140 in. thick, 1 in. ID Compressed to 0.024 in. -(83%) Compression set 58% 18 000 lb. initial force, 27% loss before cool-down (Relation of elastomer)		

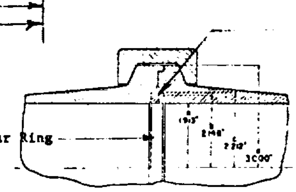
SEAL ANALYSIS

STATIC SEAL
APPLICATION -- ELASTIC DEFORMATION
SEAL TYPE --- RING

REFERENCE	Logan, S.E. 1962	
NO.	Supp 035	
INFORMATION BASED ON TESTS RECOMMENDATION	X	X
TEMPERATURE FLANGE	160°R Nitrogen 75°R Hydrogen	222°R Nitrogen 123°R Hydrogen
PRESSURE PSI	1200 - 5 cycles 1800 - 1 cycle	0 - 1 cycle 1200 - 8 cycles 1800 - 1 cycle
VELOCITY FPM	N.A. →	
MATERIAL		
SEAL	Elastomer molded onto ring of invar →	
MATE	Aluminum Flange	Stainless Steel Flange
FLUID	Liquid Nitrogen → Liquid Hydrogen →	
LEAK RATE (Atm. cc/sec)	Zero	1 cycle - 4 ml total at low pressure 8 cycles - zero total 1 cycle - 36 ml total - stopped in few min.
TEST DURATION LIFE	6 test cycles with five tests Cycles in sequence without disassembly	10 test cycles with six tests Cycles in sequence without disassembly
REMARKS	Elastomer - Adiprene - C (100 parts by weight Adiprene - C; 30 parts P33 Carbon Black; 5 parts Di Cup 40) Hardness - 70 Durometer - A Test cycle - cool with liquid Nitrogen - 30 minutes; pressurize to 1200 psi in 200 psi increments over one minute period holding constant pressure 5 to 10 minutes. Drop pressure in same increments and refill with liquid Nitrogen to restore original temperature. Equilibrium, (10 minutes) Drain Nitrogen, purge with Helium, liquid Hydrogen transfer - 5 liters/min - 15 mins., pressurize. Design information contained in "Static Seal for Low Temperature Fluids," J. American Rocket Soc., V. 25, No. 7, 7-55.	
DESIGN - See Sketch		

Elastomer

Invar Ring



Sketch of Test Flange

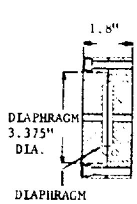
SEAL ANALYSIS

STATIC SEAL
APPLICATION -- ELASTIC DEFORMATION
SEAL TYPE --- RING

REFERENCE NO.	Weitzel, D.H., etc. August 1960 W031		
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X
TEMPERATURE MAX. RANGE	20^{o}K - Amb.		
PRESSURE PSI MAX. RANGE	1000		
VELOCITY FPM	N.A.		
MATERIAL SEAL	Rubber	Nitrile Rubber	Vitcn A
MATE	Stainless Steel		
FLUID			
SURFACE FINISH (μ in. CLA)			
LEAK RATE	Less than 10^{-4}cc STP.hr/in of seal		
TEST DURATION LIFE	Several days of warm up and cool down		
REMARKS	1 inch "O" Rings 1/8 inch diameter compressed to 0.025 inches Seal pressure, - 20 - 30 ft-lb., six 3/8 - 24 bolts		

SEAL ANALYSIS

STATIC SEAL
APPLICATION -- ELASTIC DEFORMATION
SEAL TYPE --- DIAPHRAGMS

REFERENCE NO.	A.J. Davies Oct. 1958 D-006
INFORMATION BASED ON TESTS RECOMMENDATION	X
TEMPERATURE °F MAX. RANGE	up to 570
PRESSURE PSIA MAX. RANGE	Vacuum to 315
MATERIAL SEAL	4.5 inch diameter spring steel diaphragm coated with .00025" thick teflon.
MATE	Stainless steel flanges.
FLUID	
SURFACE FINISH (μ in. CLA)	
LEAK RATE (Atm. cc/sec)	
TEST DURATION LIFE	
REMARKS	
DESIGN	

II.A.1.b.

STATIC SEAL

PLASTIC DEFORMATION

GENERAL REFERENCES

STATIC SEALS -- SELECTION AND CONTROL OF GEOMETRY

CATEGORY II.A.1.b

1. Anonymous, Sealing Of Hydraulic Compartment Access Door, C54 Series, Douglas U.S. Air Force, P.B. 91771 - Off. Tech. Ser., Mar., 1948, (A-111).

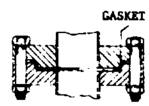
CATEGORY II.A.1.b.(1)

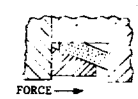
1. Anonymous, Seals Against Fluid Pressure, Part II; Materials Used As Principal Constituents Of Packing And Gasket, Power Plant Engineering, V. 40, No. 6, pp. 354-6, June 1936, (A-064).
2. Anonymous, Metallic Gasket Materials, Paper Ind., V27, p. 926, Sept. 1945, (A-100).
3. Anonymous, Design Manual on Nonmetallic Gaskets, Machine Design, V. 26, pp. 157-188, Nov. 1954, (A-141).
4. Anonymous, Non-Metallic Gaskets for Pipe Flanges, American Standards Association: ASA B16.21 - 1962, (Supp. 031).
5. Anonymous, Ring-Joint Gaskets and Grooves for Steel Pipe Flanges, American Standards Association: ASA B16.20 - 1956, (Supp. 032).
6. Boltz, R. W., Gaskets in Design, Mach. Design, V. 17, pp. 151-156 - March 1945, (B-070).
7. Elonka, S., Gaskets, Power, V. 98, pp. 105-124, March 1954, (E-018)
8. Whalen, J. J., How To Select The Right Gasket Material, Product Engineering, V. 31, pp. 52-56 - Oct. 3, 1960, (W-040).
9. Whalen, J. J., Select and Apply Gaskets Effectively, Chemical Engineering - V. 69, No. 20, pp. 83-88, Oct. 1, 1962, (W-041).
10. Galloway, C. W., Evaluation of High Temperature Hydraulic Seals to Temperature of 550°F, Feb. 1958, D.D.C. Report AD 142029.

SEAL ANALYSIS

STATIC SEAL
APPLICATION -- PLASTIC DEFORMATION
SEAL TYPE --- GASKETS

REFERENCE NO.	R. Brymmer June 1959 B-124			J. Holden et al June 1959 H-060	J.W. Hull Sept. 1960 H-079	L.N. Navarre et al Feb. 1951 AD204937
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	-306 to +1472		-306 to +932	68 to 1022	-65 to 700	530 to 1025
PRESSURE PSIA MAX. RANGE	0 to 14.7			0 to 14.7	4000 Surge to 6000	900
MATERIAL SEAL	Soft Iron	Stainless Steel	Cu, Fe	Al	Tool and Stainless Steel	Flexitallic Gasket (347 SS)
MATE	Mild Steel Flange	Stainless Steel Flange	Mild Steel Union	Steel	Nitralloy, Tool Steel	Flat Flange
FLUID	Air			Air	Hydraulic Fluids	Air and Water (simulating a steam line)
SURFACE FINISH (μ in. CLA)				Fine Emery on Ring	8 to 32	
LEAK RATE (Atm. cc/sec)	10^{-9}	10^{-8}		0	0	No difference in before and after leak rates.
TEST DURATION LIFE				10 cycles		Can tolerate 50 to 100 cycles of thermal shock.
REMARKS	Tests on flange seal. When breaking the joint it was found that the soft iron gasket had welded to the mild steel conical surfaces. It is felt therefore that the upper limit of temp. for this seal must be fixed at approx. 932°F.	Seal easily taken apart after test.	Tests on a union seal.	Loading - to 4100 lb/in. compression	Design formulae and tables are given for metal face seals.	Loading: 40,000 in/lbs. bending moment
DESIGN						





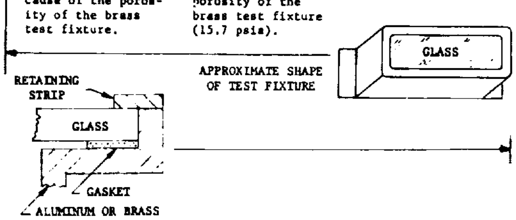
II.A.1.b.(1)
Chart 1

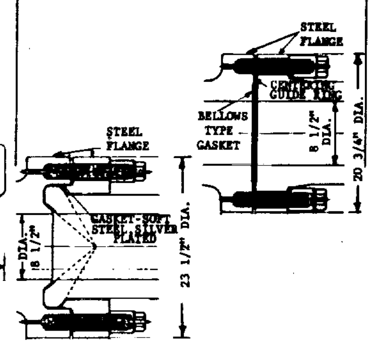
SEAL ANALYSIS

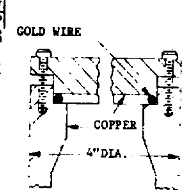
STATIC SEAL
APPLICATION -- PLASTIC DEFORMATION
SEAL TYPE --- GASKETS

II.A.1.b.(1)
Chart 2

REFERENCE NO.	E.L. Dickerson et al Oct. 1954 AD48807			E.C. Bailey et al Jan. 1953 B-002		A. Wexler et al March 1950 W-038
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	See Remarks			1050		70 to -454
PRESSURE PSIA MAX. RANGE	Test fixture pressurized with 18.7 psia at room temp. at start of test.*			1815		Vacuum
MATERIAL SEAL	Lead wires and press tite wax (INGF shop L31)			Soft steel-silver plated.	Cobalt base alloy L-605.	Gold wire (.020" dia.)
MATE	Brass (large fixture).	Brass (med. fixture).	Aluminum (small fixture).	10" dia. flanges (Nom.)-One of ferritic low-alloy forged steel; one of austenitic stainless forged steel.		Copper surfaces
	Glass					
FLUID	Air			Steam		Helium gas and liquid Helium.
SURFACE FINISH (μ in. CLA)	63 RMS					
LEAK RATE (Atm. cc/sec)	See Remarks		Passed all tests*	No leakage		Less than $5x10^{-9}$ (micro liters per second).
TEST DURATION LIFE	12 days	8 days	14 days	17 months	8700 hrs.	
REMARKS	Evaluation of optical seals and optical sealing methods. Following are tests to which seals were subjected: (1) fire hose test; (2) 24 hr. cold test at -20F; (3) 24 hr. hot test at +165F; (4) 72 hr. weathering test; (5) vibration test (vert. & horiz.); (6) shock test (vert. & horiz). *For acceptance of the seal, the pressure must not be below 17.7 psia at room temp. at the conclusion of all tests. Failed after the weathering test because of the porosity of the brass test fixture.	Failed after the hot test because of the porosity of the brass test fixture (15.7 psia).		Pressure-seal joint	Bellows type joint	Assembly survived 12 cooling-heating cycles between room temp. and -319°F & 6 openings & closures, with no evidence of leakage.
DESIGN						







GENERAL REFERENCES

STATIC SEALS -- SELECTION AND CONTROL OF GEOMETRY

CATEGORY -- II.A.1.b.(2)

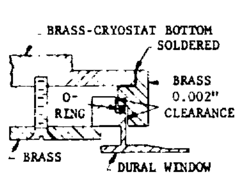
1. Andrews, J.N., Hollow Metal O-Rings. Seals for Extreme Environments, Product Engineering, V. 33, pp. 47-57, Nov. 1962, (A-015)

SEAL ANALYSIS

STATIC SEAL
APPLICATION -- PLASTIC DEFORMATION
SEAL TYPE --- RINGS

II.A.1.b.(2)
Chart 1

REFERENCE NO.	P.P. Craig Aug. 1962 Supp. 011
INFORMATION BASED ON TESTS / RECOMMENDATION	X
TEMPERATURE °F MAX. RANGE	80 to cryogenic
PRESSURE PSIA MAX. RANGE	10^{-5}mm Hg
MATERIAL	
SEAL	Soft solder or cerroseal 35
MATE	Brass & 6061 dural
FLUID	Air
SURFACE FINISH (μ in. CLA)	
LEAK RATE (Atm. cc/sec)	Mass spectrometer leak tight
TEST DURATION LIFE	Remain reliable after many thermal cycles.
REMARKS	The sealed Dural window permitted passage of low energy gamma radiation into a helium cryostat.
DESIGN	



GENERAL REFERENCES

STATIC SEALS -- SELECTION AND CONTROL OF GEOMETRY

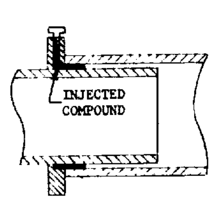
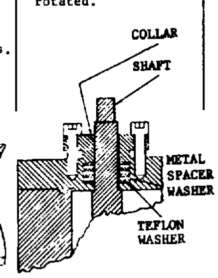
CATEGORY -- II.A.1.b.(3)

1. Anonymous, Seals Against Fluid Pressure, Part II; Materials Used As Principal Constituents Of Packing And Gasket, Power Plant Engineering, V. 40, No. 6, pp. 354-6, June, 1936, (A-064)

II.A.1.b.(3)
Chart 1

SEAL ANALYSIS

STATIC SEAL
APPLICATION -- PLASTIC DEFORMATION
SEAL TYPE --- PACKING

REFERENCE / NO.	M.W. Varrell May 1957 AD140116	P.A. House Feb. 1962 AD273284	E.T. Leidigm et al June 1957 AD140572	Anonymous Nov. 1932 A-053	E.A. Billett et al Feb. 1958 B-054	Anonymous July 1955 Supp. 007
INFORMATION BASED ON TESTS / RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. / RANGE			Cycling -65 to 160F		70-392	
PRESSURE PSIA MAX. / RANGE	1000			415 (steam)	Vacuum	170,000
MATERIAL SEAL	#73902-5 Ring Packing Plug	Polypropylene Oxide-Toluene Diisocyanate Adduct thickened with Colloidal Silica and Triethylene Tetramine.	Polyurathane Foam	Semi-plastic, heat-resistant, composition whose principal ingredients are long fiber asbestos, graphite, & inorganic filler, impregnated with a high-test oil.	.062" thick teflon washers & .062" thick polished steel spacer washers.	Sapphire insulating bushing; soft steel electrodes; rubber O-Ring; soft steel pad.
MATE	Sonar Stuffing Tube					
FLUID	Water			Steam	Helium	
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm. cc/sec)	Satisfactory	Seal punctures caused by 22 caliber bullets	No exudation (Material in perfect condition)		$11x10^{-5}$ μ/sec.	
TEST DURATION / LIFE			4 Weeks	Equivalent of 20 yrs. service.		
REMARKS		Aerospace Vehicle Application. Punctured wall sealer. Hollow wall containing two fluids separated by a membrane. Puncture causes fluids to mix and solidify.	Bomb Sealer	Steam expansion joint using a type of injection seal. 8 in. stroke at 10 strokes per hr.-run for 8 hrs. per day.	The shaft can be rotated.	An insulated electrical (4 wire) feedthrough designed by the National Bureau of Standards.
DESIGN						

II.A.1.c.

STATIC SEAL

HERMETIC

GENERAL REFERENCES

STATIC SEALS -- SELECTION AND CONTROL OF GEOMETRY

CATEGORY -- II.A.1.c

1. Anonymous, Ceramic-to-Metal Seals for High Temperature Operation, Los Alamos Scientific Laboratory for U.S. Atomic Energy Commission - 1963, 35 pages (LAMS-2917), (Supp. 008)

2. Reimann, A.L., Vacuum Technique, Textbook published by Chapman & Hall - 1952, (Supp. 022)

3. Meyer, E.A., et al, Determination of Leakage Values of Seals, Nov. 1953, D.D.C. Report AD 41362

4. Forster, G.A. Hermetic Sealing, April 1956, D.D.C. Report AD 107274

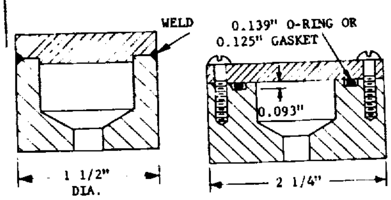
CATEGORY II.A.1.c.(1)

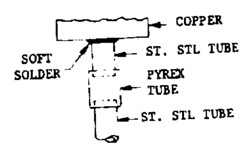
1. McCormick, J. B., Development of Pressure Weld Pad Cartridge Seal, Feb. 1963, D.D.C.Report Supp. AD 299586.

SEAL ANALYSIS

STATIC SEAL
APPLICATION -- HERMETIC
SEAL TYPE --- WELDING

REFERENCE NO.	E.A. Meyer et al Nov. 1953 AD41362		A.R. Timmins May 1956 AD107713	N.C. Einspruch et al Aug. 1963 Supp. 015
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X
TEMPERATURE °F MAX. RANGE	77		Room Temperature.*	75 to 3140 cycled
PRESSURE PSIA MAX. RANGE	Vacuum System 0-14.7		29.4	Vacuum
MATERIAL SEAL	Oxy-acetylene with weld rod.	Arc weld with weld rod.	Body material - Steel, SAE 1335 Envelope material - Stainless Steel.	Pyrex tubing (Corning 7740), 304 St. Stl., lead coated copper.
MATE	Steel to Steel			Metal
FLUID	Air & H_2		Test gas was Helium	Helium
SURFACE FINISH (µ in. CLA)				
LEAK RATE (Atm. cc/sec)	(per seal inch) (air) $<3\times10^{-8}$ * 40	3×10^{-4} 23	Satisfactory	Less than 4×10^{-9}
TEST DURATION LIFE				
REMARKS	*% of total seals tested which perform within the stated leak rate. A quantitative type leak detector was developed for this program. Seals preconditioned prior to leak testing by exposure to:- Pressure (0 to 30 psi) - 100% Rel. Hum.--Mechanical vibration & temperature cycling.		*The seals were tested: (1) as received from vendor. (2) after temp. & humidity cycling (-65 to 160°F; 95% RH) (3) after exposure to 140°F, 90% RH, .5mm pressure ammonia. (4) after jolt test; (5) after jumble test; (6) after vibration test. - Duration of tests (2) & (3) was 28 days each.	No leaks were detected even when the feed-through was isolated from the pump, after evacuation, for periods in excess of an hour.
DESIGN			Series of overlapping spot welds.	





II.A.1.c.(1)
Chart 1

SEAL ANALYSIS

STATIC SEAL
APPLICATION -- HERMETIC SEALING
SEAL TYPE --- WELDING (By Explosion)

REFERENCE NO.	Warren E. Fogg 1962 Supp 070
INFORMATION BASED ON TESTS RECOMMENDATION	X
TEMPERATURE °F MAX. RANGE	Amb.
PRESSURE	Pressurized to 10 psig
SEALING TIME	18, 19, 21, 22, 25 microseconds (5 tests)
MATERIAL	Aluminum tubing 3003 H 14 0.5 in. O.D. x 0.065 in. wall
FLUID	Air
SURFACE FINISH (µ in. CLA)	N.A.
LEAK RATE	46 tested - 2 leaked at less then 10 psi
TEST DURATION LIFE	
REMARKS	The explosive load consists of 17.5 inches of 30 Gr/Ft Petn prima cord wound in five tight turns about neoprene sheet covering tubing.

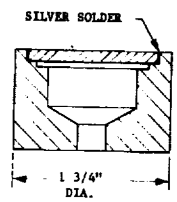
II.A.1.c.(1)
Chart 2

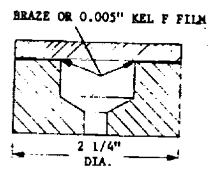
II.A.1.c.(2)
Chart 1

SEAL ANALYSIS

STATIC SEAL
APPLICATION -- HERMETIC
SEAL TYPE --- BRAZING

REFERENCE NO.	E.A. Meyer, et al Nov. 1953 AD41362 →		
INFORMATION BASED ON TESTS RECOMMENDATION	x	x	x
TEMPERATURE °F MAX. RANGE	77 →		
PRESSURE PSIA MAX. RANGE	Vacuum system → 0-14.7 →		
MATERIAL SEAL	Silver solder →		
MATE	Brass to brass	Steel to steel →	
FLUID	H_2 and Air		
SURFACE FINISH (μ in. CLA)			
LEAK RATE (Atm. cc/sec)	$\leq 3 \times 10^{-8}$ air per inch of seal → * 100	90	100
TEST DURATION LIFE			
REMARKS	*% of total seals tested which performed within the stated leak rate. A quantitative type leak detector was developed for this program. Seals preconditioned prior to leak testing by exposure to:- pressure (0 to 30 psi) - 100% Rel. Hum. - Mechanical vibration & temperature cycling.		
DESIGN			

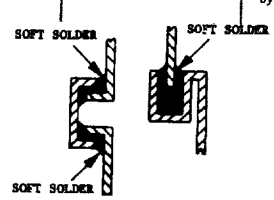


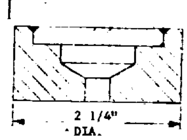


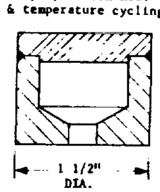
SEAL ANALYSIS

STATIC SEAL
APPLICATION -- HERMETIC
SEAL TYPE --- SOLDERING

REFERENCE NO.	W.H. Bashfield Dec. 1955 AD94590	W. Brown et al May 1957 AD130772	R.R. Bigler et al March 1961 AD256214	E.A. Meyer et al Nov. 1953 AD41362 →	
INFORMATION BASED ON TESTS RECOMMENDATION	x	x	x	x	x
TEMPERATURE °F MAX. RANGE	140	-85 to 185		77 →	
PRESSURE PSIA MAX. RANGE	0.7	32	14.7	Vacuum System → 0 - 14. 7 →	
MATERIAL SEAL	Glass	Tin plated steel	Laminated ceramic wafers metalized with molybdenum-manganese paste & sintered.	Aluminum Solder	Soft Solder
MATE	Solder-97 1/2% Pb 2 1/2% Ag	Soft Solder	Brass Cap.	Al to Al	Brass to Brass
FLUID	Air	Air	Helium	H_2 & Air →	
SURFACE FINISH (μ in. CLA)					
LEAK RATE (Atm. cc/sec)		0.3×10^{-6} (per in. of seal -Max.)	Satisfactory units obtained.	$\leq 3 \times 10^{-8}$ (air) (per seal inch) * 100	95
TEST DURATION LIFE	Satisfactory 500 V RMS	Opened & resealed 15 times.			
REMARKS		Resealable cases for electronic components. Tolerated shock & vibration tests.	Crystal micro-element package sealed by ultrasonic fluxless soldering technique. Edges sealed by electroplate.	*% of total seals tested which perform within the stated leak rate. → A quantitative type leak detector was developed for this program. → Seals preconditioned prior to leak testing by exposure to:-Pressure (0 to 30 psi) - 100% Rel. Hum. - Mechanical vibration & temperature cycling.	
DESIGN					







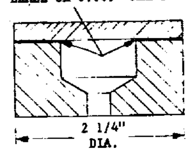
II.A.1.c.(3)
Chart 1

SEAL ANALYSIS

STATIC SEAL
APPLICATION -- HERMETIC
SEAL TYPE --- CEMENTING

REFERENCE NO.	R.W. Bryant et al July 1958 AD200898	E.A. Meyer et al Nov. 1953 AD41362 →				
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	Cycling	77 →				
PRESSURE PSIA MAX. RANGE	4 1/2 ft. drop to concrete	Vacuum System → 0 to 14.7 →				
MATERIAL SEAL	Threaded Joint RD1286 Cement RD1284(Luting MX8)	Aluminum .005" KEL-F Film	Nitro-seal	Selectron 5208 (polyester)	Silastic 120 (Silicone)	Brass .005" KEL-F Film
MATE		Aluminum →				Brass
FLUID		H_2 & Air →				
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm. cc/sec)		<$3x10^{-8}$ (air) (per seal inch) → * 80	25	65	95	73
TEST DURATION						
LIFE						
REMARKS	Avoid dissimilar metals.	*% of total seals tested which perform within the stated leak rate. A quantitative type leak detector was developed for this program. Seals preconditioned prior to leak testing by exposure to:- Pressure (0 to 30 psi) - 100% Rel. Hum. - Mechanical vibration & temperature cycling.				
DESIGN	Use shoulder on threaded part. See also Report AD154972.					

BRAZE OR 0.005" KEL F FILM

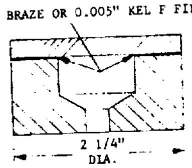


SEAL ANALYSIS

STATIC SEAL
APPLICATION -- HERMETIC
SEAL TYPE --- CEMENTING

REFERENCE NO.	E.A. Meyer et al Nov. 1953 AD41362 →		
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X
TEMPERATURE °F MAX. RANGE	77 →		
PRESSURE PSIA MAX. RANGE	Vacuum System → 0 to 14.7 →		
MATERIAL SEAL	Brass → Nitroseal	Adraldite Type XI (epoxy)	Silastic 120
MATE	Brass →		
FLUID	H_2 and Air →		
SURFACE FINISH (μ in. CLA)			
LEAK RATE (Atm. cc/sec)	< $3x10^{-8}$ (air) (per seal inch) → * 37	98	35
TEST DURATION			
LIFE			
REMARKS	*% of total seals tested which perform within the stated leak rate. A quantitative type leak detector was developed for this program. Seals preconditioned prior to leak testing by exposure to: - Pressure (0 to 30 psi) - 100% Rel. Hum. - Mechanical vibration and temperature cycling.		
DESIGN			

BRAZE OR 0.005" KEL F FILM



GENERAL REFERENCES

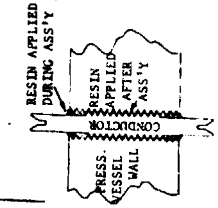
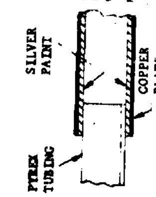
STATIC SEALS -- SELECTION AND CONTROL OF GEOMETRY

CATEGORY -- II.A.1.c.(5)

1. Battelle Memorial Inst., Columbus, Ohio, Investigation of Encapsulents for Ground Electronic Equipment, June 1956, D.D.C. Report #AD97767

SEAL ANALYSIS

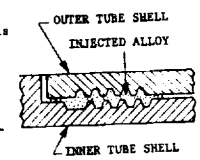
STATIC SEAL
APPLICATION -- HERMETIC
SEAL TYPE --- MOLDED IN PLACE

REFERENCE / NO.	INFORMATION BASED ON TESTS / RECOMMENDATION	TEMPERATURE °F MAX. RANGE	PRESSURE PSIA MAX. RANGE	MATERIAL SEAL	MATE	FLUID	SURFACE FINISH (μ in. CLA)	LEAK RATE (Atm. cc/sec)	TEST DURATION	LIFE	REMARKS	DESIGN
C.F. Scharfenstein May 1958 AD218038	X			PRC 1201T Caulk Material	Al to Al Joint or Fe to Al Joint	10% Aqueous Na-Cl			30 days + 200 hrs. accel. weathering		Insulating mat'l: Mim. Mining chrome-lock tape type N-MIL-G20241, symbol 2291.	Method of insulated bimetal joining in boat construction.
S.S. Pigman April 1958 AD200230	X	500		Silicone Base Type Sealant	Electrical Cable TCTX-3	Air				96 hrs.-little change in insul. resistance	Tests were performed on cable specimens filled with sealant material.	
R.W. Bryant et al July 1958 AD200898	X	Cycling	4 1/2 ft. drop to concrete	Threaded Joint RD1286 Cement RD1284 Luting	MK8						Avoid dissimilar metals.	Use shoulder on threaded part. See also Report AD154972.
Anonymous March 1961 Supp. 005	X	1200	Vacuum	Supramica 620BB (Mycalex Corp. of America).		Helium		Less than $2x10^{-10}$*			*After severe environmental cycling.	
D.F. Gibbs Dec. 1958 Supp. 016	X		66,150	Insulation: Marco resin type SB.28C with 16% by weight mica flour filler. Approx. .005" thick.						No electrical or mechanical failures.	High-pressure electrical lead-in. 100 pf capacity. Resistance greater than 1000M~.	
N.A. Eckstein et al April 1950 Supp. 014	X	300	Vacuum tight	Pyrex; ceramic type silver paint.	Copper tubing.*						*This is an electroforming process in which the copper tubing is formed by electroplating. This seal is reasonably insensitive to thermal shock.	

SEAL ANALYSIS

STATIC SEAL
APPLICATION -- HERMETIC
SEAL TYPE --- MOLDED IN PLACE

REFERENCE	G.J. Dashefsky April 1957 AD134497			A.R. Timmins May 1956 AD107713	
NO.					
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X
TEMPERATURE °F MAX. RANGE				Room Temperature*	-50 & 180
PRESSURE PSIA MAX. RANGE	40			29.4	
MATERIAL SEAL	Stuffing Tube Steel-type 30-2A size G. Conventional packing.	Plastic type 5D, grommet size 120. Vendor: Danielson Manufacturing Co. Phenoweld 7 adhesive (H.V. Hardman Co., Inc.)	EC1159 adhesive (Minn. Mining & Manufact.)	Low melting point (217°F) ternary alloy composed of Bismuth (54%), Tin (26%), Cadmium (20%). Injected through hole in outer shell.	
MATE	Cable DSGA-30	TSGA-50 (Armor removed)	TSGA-50 (Armored)	Steel (SAE 1335) and stainless steel	
FLUID	Water			Test gas was helium.	
SURFACE FINISH (μ in. CLA)					
LEAK RATE (Atm. cc/sec)				Satisfactory.	**
TEST DURATION LIFE					
REMARKS	Evaluation of plastic stuffing tubes for bulkhead penetration. (1) After vibration test: Leak tight. (2) After shock test: Leak tight. (3) After cable pull out test: Not tested (4) After fire test - condition (A)-(Least severe) Leak tight (5) After fire test - condition (B)-(Medium severity) Leaked (6) After fire test - condition (C)-(Most severe) Leaked	(3) Leak tight (5) Leak tight (6) Destroyed	(3) Leaked	*The seals were tested: (1) As received from vendor. (2) After temp. & Humidity cycling ($-65^{o}F$ to $160^{o}F$; 95% RH). (3) After exposure to $140^{o}F$, 90% RH, .5MM pressure ammonia. (4) After jolt test. (5) After jumble test. (6) After vibration test.-Duration of tests (2) and (3) was 28 days each.	**With the mass spectrometer leak detector sensitivity at $1x10^{-9}$ cc/sec, 1 out of 8 leaked at -50F & none out of 5 leaked at +180F.
DESIGN				Alloy expands on solidification. The seal is entirely dependent on this characteristic since there is no alloying of the injected metal with either the steel or st. steel.	



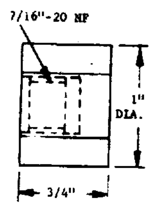
SEAL ANALYSIS

STATIC SEAL
APPLICATION -- HERMETIC
SEAL TYPE --- MOLDED IN PLACE

REFERENCE	G.A. Forster April 1956 AD107274					
NO.						
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X	X	X	X
TEMPERATURE °F MAX. RANGE	-85 to 302		-49 to 212		-85 to 302	
PRESSURE PSIA MAX. RANGE						
MATERIAL SEAL	Hermetic Sealing Terminals. Soft glass to steel.	Compression type glass to steel-One insulator for all conductors.	Compression type glass to steel - individual insulators for each conductor.	Gasket type.	Compression type glass to steel.	Hard glass; also metalized.
MATE						
FLUID						
SURFACE FINISH (μ in. CLA)						
LEAK RATE (Atm. cc/sec)	10^{-9}					
TEST DURATION LIFE						
REMARKS	The above samples were subjected to and passed the following tests: (1) Mounting; (2) Voltage strength; (3) Corona voltage; (4) Current capacity; (5) Capacitance and loss factor; (6) High temperature aging; (7) Temperature cycling; (8) Temperature shock; (9) Vibration; (10) Shock; (11) Humidity; (12) Terminal strength; (13) Salt spray. Acceptance is largely based upon electrical performance and leakage rate rather than upon absence of cracks, surface appearance, or materials.					
DESIGN						

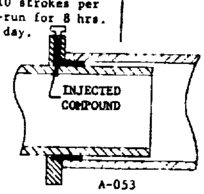
SEAL ANALYSIS

STATIC SEAL
APPLICATION -- HERMETIC
SEAL TYPE --- MOLDED IN PLACE

REFERENCE NO.	L.F. Hatakeyama et al June 1960 AD242347
INFORMATION BASED ON TESTS RECOMMENDATION	X X X X X X
TEMPERATURE °F MAX. RANGE	
PRESSURE PSIA MAX. RANGE	35 psia applied for 3 minutes.
MATERIAL SEAL	Impregnating material: American Metaseal 19V5 and Dow-Corning R-7501.
MATE	Sintered, Powder Metal Rods (MIL-B-5687).
FLUID	Air
SURFACE FINISH (μ in. CLA)	
LEAK RATE (Atm. cc/sec)	No leakage.
TEST DURATION LIFE	15 days
REMARKS	Tests involve the impregnation of porous metal castings. Samples of cured specimens were wire brushed and then tested. If they showed no leakage they were conditioned for 15 days as follows: In air at 300°F. In air at 500°F. In Ethylene Glycol At 385 °F. In Hydraulic Fluid (MIL-F-17111) at 210°F. In Lubricating Oil (MIL-L-15016) at 210°F. In Boiling Water At 212°F. The impregnants were tested for curing compatibility with Aluminum, Copper, Iron, Magnesium and Zinc and were determined to be compatible.
DESIGN	

SEAL ANALYSIS

STATIC SEAL
APPLICATION -- HERMETIC
SEAL TYPE --- MOLDED IN PLACE

REFERENCE NO.	L.F. Hatakeyama et al June 1960 AD242347	Anonymous Nov. 1932 A-053	G.A. Forster April 1956 AD107274
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X
TEMPERATURE °F MAX. RANGE			-85 to 302
PRESSURE PSIA MAX. RANGE	35 psia applied for 3 minutes	415 (steam)	
MATERIAL SEAL	Impregnating material: Amer. Metaseal 19V5 & Dow-Corning R-7501.	Semi-plastic, heat resistant, composition whose principal ingredients are long fiber asbestos, graphite, & inorganic filler, impregnated with a high-test oil.	Hard glass - one insulator for all conductors. (Hermetic Sealing Terminals.)
MATE	Sintered, powder metal rods (MIL-B-5687).		
FLUID	Air	Steam	
SURFACE FINISH (μ in. CLA)			
LEAK RATE (Atm. cc/sec)	No leakage		10^{-9}
TEST DURATION LIFE	15 days	Equivalent of 20 yrs. service.	
REMARKS	Tests involve the impregnation of porous metal castings. Samples of cured specimens were wire brushed & then tested. If they showed no leakage they were conditioned for 15 days as follows: In hydrocarbon fluid (MIL-H-3136) at room temp.	Steam expansion joint using a type of injection seal. 8 in. stroke at 10 strokes per hr.-run for 8 hrs. per day.	See II.A.1.c.(5) chart 3.
DESIGN	The impregnants were tested for curing compatibility with Al., Copper, Iron, Magnesium, Zinc & were determined to be compatible.		

161

II.A.1.d.

STATIC SEAL

MISCELLANEOUS

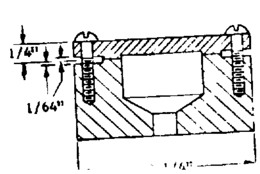
162

SEAL ANALYSIS

STATIC SEAL
APPLICATION --
SEAL TYPE --- LAPPED JOINT

II.A.1.d.
Chart 1

REFERENCE	E.A. Meyer et al Nov. 1953 AD41362		
NO.			
INFORMATION BASED ON TESTS RECOMMENDATION	X	X	X
TEMPERATURE °F MAX. RANGE	77		
PRESSURE PSIA MAX. RANGE	Vacuum System		
MATERIAL	Lapped with jewler rouge and water		
SEAL MATE	Al to Al	Brass to Brass	Steel to Steel
FLUID	Air & H_2		
SURFACE FINISH (μ in. CLA)	Max. scratch width & depth 1 micron		10 microns
LEAK RATE (Atm. cc/sec)	(per seal inch) (air) $< 3\times10^{-8}$ * 5	15	$< 3\times10^{-4}$ 95
TEST DURATION LIFE			
REMARKS	*% of total seals tested which perform within the stated leak rate. A quantitative type leak detector was developed for this program. Seals preconditioned prior to leak testing by exposure to:- Pressure (0 to 30 psi) - 100% Rel. Hum. - Mechanical vibration & temperature cycling.		
DESIGN			



CHAPTER IX

Bibliography of Literature on Seals

Bibliography of ASTIA Literature on Seals

ACKNOWLEDGEMENT

This volume is to large extent the result of assistance and suggestions given to the authors by many people.

Mr. John P. Laniewski performed a significant service in categorizing the content of these many references, for the ultimate construction of the Subject Index.

Miss Esther Kalis and Mr. W. J. Leahy of the Main Library assisted in the acquisition of documents and preparation of the form of the Subject Index.

ASTIA personnel (Mrs. Henery and Mr. Brown) were very cooperative in searching their material and furnishing abstracts.

To all of the above, and the others who helped in many other ways, we would like to express our appreciation for the courtesy and helpfulness offered to us in this project.

R. L. George
R. C. Elwell

INTRODUCTION

This is a bibliography prepared by the General Engineering Laboratory of the General Electric Company under NASA contract number NAS 7-102. It contains unclassified abstracts furnished by ASTIA (Armed Services Technical Information Agency, Arlington Hall Station, Arlington 12, Va.) on static and dynamic seals.

A seal has been defined as an assembly, including sealing medium, mounted between mechanical components to prevent the interchange of fluids across an interface. When the relative motion between the mechanical components is limited to that induced by internal, thermal, elastic, or pressure effects, the seal is defined as a static seal. If the relative motion is not limited, the seal is called a dynamic seal.

The objective of this bibliography is to present abstracts of prominent seal references, and a subject index which may be used as a tool to quickly guide the reader to ASTIA literature pertinent to his particular interest.

ASTIA collects, abstracts, and files literature written under Armed Service contracts by assigning them an ASTIA number for identification and description headings for the classification of information they contain. At our request, this agency searched the literature in their possession as of June 1962 and supplied 1100 abstracts pertinent to seal technology, as classified under the descriptor headings listed below:

Carbon Seals	Hydraulic Seals	Pneumatic Packings
Cork Seals	Metal Seals	Rotary Seals
Fuel Seals	Nylon Seals	Rubber Seals
Gas Seals	Oil Seals	Rubber Gaskets
Gas Leaks	Plastic Seals	Seals
Gaskets	Plastic Gaskets	Vacuum Seals

The ASTIA bibliography number assigned to this group of references is ARB 2021.

The 1100 abstracts were read and those that did not contain information pertinent to the advancement of seal technology or to the designing and operation of seals were discarded. The remaining ASTIA document references are listed according to ascending ASTIA numbers in this volume with titles, date of publication, author, and abstract if supplied by ASTIA. At the end of this listing, a subject index is presented to increase the value of the bibliography. In the subject index are listed ASTIA numbers of documents which contain information outlined by the category heading. The letters AD which precede most document numbers have been omitted for brevity.

The major divisions of the subject index are:

1) the kind or type of seal discussed in the document and if it is a dynamic or static seal,
2) a specific area of interest in seal design such as the effect of friction, surface finish, material properties, etc.,
3) the materials used in the seal, and
4) the applications for which the seal was designed and operated.

Each of these divisions is further broken down into specific categories, which contain no more than 75 references for easy use.

The subject index is not limited to the specific headings listed. By cross referencing between categories, a new category of particular interest may be constructed. Two examples are given for illustration.

Example 1: What is in the possession of ASTIA about synthetic rubber gaskets in launch vehicles?

A cross reference is made between gaskets (p. 189) and launch vehicle seals (p. 221). It is found that document number AD 240-573 is the only publication on this subject.

Example 2: What references are available on leakage of dynamic packings in hydraulic equipment?

A cross reference is made between dynamic packings (p. 191), hydraulic equipment (p. 220), and dynamic leakage (p. 199). It is found that three ASTIA documents contain information on all three subjects: AD 036-456, AD 234-860, and AD 259-550.

This is the first edition of this bibliography, which is presented in the interest of the advancement of seal technology. It would be appreciated if the authors' attention would be drawn to any errors, omissions, or suggestions which will improve the usefulness of this bibliography.

AD-1046

Pitman-Dunn Labs., Frankford Arsenal, Philadelphia, Pa.
INJECTED METAL SEALS FOR THE T2031 FUZE, by Sara S. Lipton. Dec 52. 7p. illus. table (Memo. rept. no. MR-533)

The variables governing the injection of fusible alloy seals for joining components of the T2031 fuze were investigated. A conventional wax injector, designed for wax pattern production for investment casting, was modified for injecting the seals. Of two methods of injection, that which involved the use of air pressure to force the metal into the sealing groove was selected. A 52.5% In-32% Pb-15% Sn eutectic alloy was employed. The sealing grooves were successfully filled at moderate injection pressures and temperatures. No difficulty was encountered with flashing. 22/1

AD-2737

Frankford Arsenal, Philadelphia, Pa.
BINOCULAR M5. EFFECT OF TROPICAL EXPOSURE ON SEAL EFFICIENCY AND LOW TEMPERATURE PERFORMANCE, by John P. Yannello. 20 Feb 53, 4p. incl. tables (Test rept. no. T53-10-1)

Three binoculars were submitted to the dew point test after exposure to high humidity conditions to test the quantitative amount of leakage. Results indicate that the binoculars are not hermetically sealed and should not be used in tropic and temperate zones if long usable life is desired. High internal humidity conditions may cause corrosion of the metallic elements and fogging of the optical elements. Lubricated with MIL-G-3278, these instruments are adequate for low temperature service without overhaul, even after exposure to tropical conditions.

AD-2011

Atomic Energy Research Establishment (Gt. Brit.).
THE PRINCIPLES OF LEAK DETECTION, by A. H. Turnbull. Mar 50 [21]p. incl. illus. tables. (AERE rept. no. G/R 478)

An attempt was made to define the optimum conditions for various methods of leak detection and to establish the ultimate sensitivities of these methods. Dynamic methods of leak detection are discussed for the condition where the surface of a continuously evacuated system is probed with a jet of a gas or vapor which is not initially present to any appreciable extent. Leak detection with the detector on the high vacuum side and on the backing side of the main diffusion pump is treated, and equations are derived for the partial pressures of air and probe gas in the system at any time. Optimum leak detection conditions are determined, and data are given enabling the minimum detectable leak to be calculated when certain detectors and probe gases are used in any system.

AD-2991

Bjorksten Research Labs., Inc., Madison, Wis.
SEALS: DETERMINATION OF LEAKAGE VALUES. Interim engineering rept. no. 2, 26 Aug-26 Nov 52. 17p. incl. illus.
(Contract AF 33(038)15981)

The leakage rate of gases through fabricated seals is to be measured quantitatively before and after various environmental tests. A portable leak meter was constructed for measuring leakage rates from 5 x 10^{-4} to 0.1 cc/sec; the Gross leak meter served as the prototype in the development. Quantitative measurements of leakage rates through sealed components on the production line appears possible within these limits. All air leaking through the test specimen is made to pass through a capillary tube; the leakage rate measurement is made by timing the rate of movement of a water pellet through the glass capillary. A procedure for lapping brass and dead soft Al on a motor driven lapping plate is described. The results are superior to the hand (over)

AD-2400

North American Aviation, Inc., Los Angeles, Calif.
DEVELOPMENT TESTS OF A HYDRAULIC POWER CONTROL SYSTEM RESERVOIR RELIEF VALVE (165-58061) APPLICABLE TO F-86D AIRPLANES (N. A. A. MODEL NA-165) CLASS-FIGHTER, by G. F. Ervin. 10 June 52 [15]p. incl. illus. tables (Rept. no. NA-52-657)
(Contract AF 33(038)9211)

A special seal was developed for the poppet of the hydraulic control reservoir relief valve which had displayed satisfactory characteristics at room and elevated temperatures and excessive leakage at -65°F. The satisfactory operation of the special molded seal was attributed to smoothness and closer tolerances.

AD-2991

lapping methods tried. Scratch depths are reduced by this method to less than 1 μ in 10 to 30 min.

AD-3133

Kellogg, M. W., Co., Jersey City, N.J.
PROTECTIVE COATINGS FOR EXPOSED METAL SURFACES IN THE ACID TANK AND ATO COMPARTMENTS OF THE B-47 AIRCRAFT, by F. J. Honn. Progress rept. 20 Jan-20 Apr 52. May 52, 12p. incl. tables (Rept. no. RL 52-198)
(Contract AF 33(038)21253)

Efforts were directed toward the development of a polychlorotrifluoroethylene (Kel-F) or modified Kel-F coating and sealant for the Al surfaces. The coating is to be resistant to WFNA penetration, serviceable from -65° to +160°F, and applicable to metal surfaces without affecting adversely their mechanical properties. Unplasticized Kel-F films and sheets had excellent resistance to WFNA. No molecular degradation and little weight or dimensional change occurred after 168 hr in WFNA at 194°F. No satisfactory method was found for applying Kel-F to Al. Kel-F copolymers, particularly the CF_2=CFCl/CF_2=CH_2 system, were
(over)

AD-3133

investigated which could be applied to Al as lacquers or low-fusion temperature dispersions. Compositions approaching ketone solubility at room temperature (where they could be applied as lacquers) were almost as resistant to WFNA as Kel-F. WFNA penetrated plasticized Kel-F more rapidly than the unplasticized material.

AD-5181

Detroit Arsenal, Center Line, Mich.
HYDRAULIC FITTINGS WITH RUBBER SEAL, by Cameron A. Burges. Technical rept. 2 Mar 53, 6p. illus. tables (Rept. no. 2259)

(Not abstracted)

Classification cancelled by authority of Executive Order 10501 [illegible] by Ch. [illegible] 12/5/53
AD-6087
5 Nov 53

Bjorksten Research Labs., Inc., Madison, Wis.
SEALS: DETERMINATION OF LEAKAGE VALUES
Interim engineering rept. no. 3, 20 Nov 52-26 Feb 53, 23p. incl. illus. tables.
(Contract AF 33(038)15981)

The essential sensitivity and reproducibility requirements of an ideal seal tester for electronic equipment are discussed. An analysis of nondestructive testing methods indicated the superiority of providing access to the interior of the component rather than sealing a test gas into the component. The use of this technique for measuring the leakage rates of sealed components is described, and photographs are included of 10 different component types ready for testing. Leakage-rate measurements which were performed during a 3-mo period are summarized in a chart. (See also AD-2991)

AD-7347

Detroit Arsenal, Center Line, Mich.
LEAKAGE AROUND SHAFT OF MANUAL ELEVATION PUMP, ORD PART NO. 7363146, SERIAL NO. 2404 MD, AFTER 100 MILES VEHICULAR OPERATION AT PROVING GROUND, by Homer E. Helm. 2 Feb 53, 5p. incl. illus. (Rept. no. 2195)

The pump was pressurized to 1500 psi which produced a leakage of 1 drop in 12 min around the shaft at the switch end. The pump seal failure was caused by 1 or more of the following conditions: shaft roughness, shaft eccentricity, improper assembly of leather backup washer, or tool marks in the end cap. 22/3

AD-8640

Rock Island Arsenal Lab., Ill.
FRICTION STUDIES OF O-RING PACKINGS IN PNEUMATIC EQUILIBRATORS AT SUB-ZERO AND NORMAL TEMPERATURES, by Warren E. Peterson. 8 Apr 53, declassified 15 June 54, 14p. illus. tables. (Proj. no. TR1-1031R. rept. no. 4; RIA Lab. no. 52-6287)
Unclassified report

Friction tests were made of O-ring packings in the pneumatic equilibrators of 155 m/m gun and 8-in. M1 howitzer carriages after storage at subzero temperature (ST) and normal temperature (NT) with various oils as packing lubricants. Dynamic and break-away frictions of the equilibrator increased with longtime static storage at ST and at NT. The equilibrators which were not in longtime static storage showed insignificant break-away friction at NT regardless of the oil used. At -50° to -60°F, the break-away frictions of about 1200 to 1600 lb were more than 100% greater than the dynamic frictions. The MIL-O-5606 Hydraulic oil was the most suitable oil
(over)

AD-8640

for reducing friction in both ST and NT operation. Colloidal graphite in the lubricant increased the frictions. (See also AD-6252)

AD-8669

Detroit Arsenal, Center Line, Mich.
HYDRAULIC FITTINGS WITH RUBBER SEALING ELEMENT, by Cameron Burges and Russell G. Burbridge. Technical rept. 27 Mar 53, declassified 15 June 54 [15]p. incl. illus. (Rept. no. 2329)
Unclassified report

(Not abstracted)

AD-9079

Naval Proving Ground, Dahlgren, Va.
GUNFIRE QUALIFICATION TEST OF MODEL F2H-3 AIRPLANE FUSELAGE SELF SEALING FUEL CELL INSTALLATION, by H. P. Branscome. Final rept.
Final rept. on Self Sealing Fuel Cells Manufactured by U. S. Rubber Co. 11 Dec 52, declassified 15 June 54, 10p. illus. tables (NPG rept. no. 1068)
Unclassified report

Tests were conducted to determine the resistance of the F2H-3 fuselage fuel-cell installation to leakage, slosh resistance, and gun fire. The installation consisted of three US Rubber Co. self-sealing cells in the monocoque-type fuselage between stations 124.4 and 336.9. No leakage was observed during the application of 3.5 psi of air to the cells. The slosh test of the 2/3-full cells consisted of rocking the installation along the axis of the aircraft. After 4 hrs (a failure of the test equipment prevented the completion of the required 25 hrs), no damage or leakage was observed. Twelve 0.50-cal AP projectiles
(over)

AD-9079

fired from 75 ft produced 23 qualifying wounds of which 14 sealed satisfactorily. Only a small amount of fuel was lost from the remainder, although cords which were frayed into the wounds prevented the sealant from acting quickly. A 40-mm HEP projectile detonated 2.5 ft from the port side of the installation produced 14 wounds of which 3 leaked slightly. These sealed within the specified 2 min. Only minor damage to the supporting and surrounding structure was produced by the projectiles and/or hydraulic fuel surge. The installation was considered to conform to requirements.

AD-0494

Aeronautical Electronic and Electrical Lab., Naval Air Development Center, Johnsville, Pa.
EVALUATION OF SEAMS AND GASKETS USED IN SHIELDING ENCLOSURES, AND DEVELOPMENT OF CORRECTIVES FOR IMPROVEMENT, by W. E. Stinger. Final rept. 3 Feb 53, 43p. incl. illus. (Proj. TED no. ADC EL-536; Rept. no. NADC-EL-52139)
Unclassified report

Research was directed toward the investigation and development of a method of suppressing the radio-interference leakage emanating from the seams of shielding enclosures which are used in engine ignition systems. Conclusions were obtained from test results relative to the following items: (1) RF impedance and effective pressure, (2) the point of contact, (3) the shape of the contact area, (4) resistivity of the gasket material, (5) sealing for air pressure, (6) RF current path, (7) corrosion and aging, (8) protective coatings, (9) protective sealing, (10) rapidity of field decay, and (11) compari- (over)

AD-0494

son of various gaskets and seams. An AEEL RF gasket and a saw-tooth seam were developed. The gasket satisfactorily attenuates the leakage from seams, including anodized or other coated surfaces. A description is given of a test method developed for type-testing gaskets and seams which gives a satisfactory evaluation of the leakages present with regard to radio-interference specification requirements.

AD-10 270

Sylvania Electric Products, Inc., Kew Gardens, N. Y.
INVESTIGATION OF TUBE STRUCTURES ADAPTED TO AUTOMATIC PRODUCTION, by W. R. Wheeler. Interim technical rept. no. 6, 1 Oct 52-1 Jan 53. 30 Mar 53, 1v. incl. illus. tables (Rept. no. YF52(B1-1464-6))
(Contract NObsr-52627)

Development of the titanium-core sealing technique is being conducted for use with various ceramic components. Among these are ceramic-to-ceramic seals for joining the stem wafer to the bulb, and metal-to-ceramic seals for sealing the pins through the ceramic stem. Glazed-zircon stem seals to 7052 tubing were made; only a few of the glass pieces cracked. The cathode design of the SN-1724B and C was modified to eliminate distortion caused by expansion and to permit easier assembly. The titanium-core BT sealing method was incorporated in SN-1593B, a redesigned SN-1593A (over)

AD-10 270

tube. The B design also includes an eyelet-type support. The spread of dimensional tolerances of Sylvania ceramic spacers (0.0015 in.) was too great for the close grid-to-cathode spacing.

AD-10 979

Detroit Arsenal, Center Line, Mich.
[DETERMINE COMPRESSION SET AND BRITTLE POINT CHARACTERISTICS OF VARIOUS "O" AND "T" RINGS] by James K. Matter. Memo. rubber process rept. 8 May 53, 3p. tables (Rept. no. 2437)

Results are tabulated for two T-type and three O-type rings.

AD-11 060

Aberdeen Proving Ground, Md.
TEST OF JOHN CRANE FACE TYPE SEALS, by A. Bornstein. Rept. no. 2, 19 Nov 51-12 Dec 52. 28 Apr 53, 8p. illus. (Proj. no. TT1-19H, rept. no. 5)

(Not abstracted)

AD-11 404

Lockheed Aircraft Corp., Burbank, Calif.
FUEL RESISTANCE TESTS: 3/4, 1-1/2 AND 3-1/2 INCH O-RING INTERCONNECTS [MODEL F-94C] by F. O. Rick. 12 Feb 53, 10p. incl. illus. tables. (Rept. no. 9068)

Tests were conducted on O-ring interconnects providing near-maximum and -minimum cavity volumes to determine whether they were satisfactory seals when assembled with AF934 O-rings. The test fluids were (1) 100% isooctane, and (2) a 70% isooctane-30% PhMe mixture. The interconnects were subjected to a total soaking of 37 days, two dry-air bakes totaling 8 days, and a temperature reduction in 10° increments from room temperature to -65°F. No leakage resulted; the O-rings and parts showed no appreciable wea r damage.

AD-11 [illegible]

[Detroit Arsenal, Center Line, Mich.]
[DETERMINE CAUSE OF LEAKAGE AND/OR LOCKING UP OF EIGHT MANUAL ELEVATION PUMPS] CODE "A," by Homer E. Holm. 20 May 53, 2p. (Rept. no. 2470)

Tests were made to determine the cause of leakage and/or locking up of 8 manual elevation pumps, Ord part no. 7363146, code "A." A table shows the performance of the pumps at 1500 psi supercharge pressure, 100 psi differential. The pump defects are listed.

AD-11 915

Aberdeen Proving Ground, Md.
QUALIFICATION TEST OF ROAD WHEEL SPINDLE SEAL, ALTERNATE TO NO. B8364672, by A. Bernstein. 14 May 53, 21p. illus. (Proj. no. TT1-19H, rept. no. 6)

(Not abstracted)

AD-12 531

Naval Supply Research and Development Facility, Bayonne, N. J.
HEAT SEALING MACHINES FOR MILITARY BARRIER MATERIALS. 15 Feb 53 [91]p. incl. illus. tables, 11 refs. (Proj. NT 003-021(A); Engineering rept. no. 2, 5050)

Machines and materials used in the moisture-proof packaging of military items were evaluated according to Spec MIL-P-116. Tests were made on 4 representative moisture-vapor barrier materials utilizing a vinyl heat-sealable film laminated to aluminum foil with an adhesive. The 3 commercial heat-sealing machines studied were of the following types: (1) continuous sealer with counterrotating, roller-type, heat-sealing elements preceded by an endless chain conveyer fitted with preheaters; (2) the band or endless-belt type with 2 cartridge-type heaters and 2 forced-water coolers; and (3) a jaw-type machine equipped with a single bar heater. The sealing properties of the materials were satisfac- (over)

AD-12 531

tory if certain limits of temperature, pressure, and dwell time were observed. None of the machines were completely adequate for unlimited application. The present method of specifying heat-seal requirements is considered inadequate and dangerous. The development of a laboratory-standard heat sealer is recommended.

AD-13 311

Naval Proving Ground, Dahlgren, Va.
GUNFIRE QUALIFICATION TEST OF MODEL AD-5 AIRPLANE INTERNAL AUXILIARY SELF SEALING FUEL CELL INSTALLATION, by H. P. Branscome. Final rept. 2 July 53, 9p. illus. (NPG rept. no. 1145)

The fuel cells, enclosed in a framework consisting of extruded hat sections and angle corner supports, were 2/3 filled with isooctane fuel and bombarded from a range of 75 ft by eight 0.50-cal armor-piercing projectiles and one 20-mm practice projectile. Fifteen wounds were produced, 3 of which were above the fuel level and were discounted. Four of the 12 qualifying wounds sealed satisfactorily. The 8 leaking wounds sealed during the test, and only about 1 pint of fuel was lost. One 40-mm HEP projectile, exploded 24 in. from the outside of the installation, produced 26 wounds in the fuel cell. Of the 14 wounds below the fuel level, 12 sealed satisfactorily. The fuel cell installation conformed to requirements.

AD-13 714

Detroit Arsenal, Center Line, Mich.
INVESTIGATION OF GREASE COMPATIBILITY AND AND PHYSICAL PROPERTIES OF THREE RUBBER TURRET BEARING SEALS, by E. R. King. Technical rept. 18 June 53, declassified 15 June 54, 6p. tables. (Rept. no. 2405) Unclassified report

The materials tested included 1 rubber bearing seal from each of 3 suppliers and 10 lb of grease from each of 2 suppliers. The seals were tested for conformance with ASTM Spec D735-52aT, grade SC-615-ABCEFF. Seals 1 and 2 met requirements except for original elongation, compression set, and (for seal 1 only) tensile strength after aging. Grease B produced less increase in length than grease A. The dimensional stability of seal no. 3 was the best after exposure to both greases. None of the seals complied fully with requirements.

AD-14 468 Classification cancelled by authority of 1st Ind. - Frankford by M. Hayes Arsenal date 2 Mar 54

Frankford Arsenal, Philadelphia, Pa.
COMPUTER COMPONENTS STANDARDIZATION PROGRAM. EVALUATION OF SPECIAL DRIVE WASHERS FOR 23 TYPE SYNCHROS, by Henry Moore. 25 May 53, 5p. illus. (Technical memo. no. M53-23-1)

Special drive washers were designed and evaluated by performance testing to permit gears with existing keyways to be used on splined shaft synchros without modification of the gears, as on the Director M33. Evaluation tests were performed on the washers to determine their suitability for use with the size 23 synchro with a splined shaft. Performance under the following considerations was especially considered: backlash between members, the re-use of mating members, and the shear values and deformation of the tab. The modified washer with the tapered tab, which eliminated the play between the tab and the keyway, produced satisfactory results. Half-hard sheet brass was superior (over)

AD-14 468

to stainless steel as the material for the washers. The tab of the washer will transmit more than 15 times the minimum applied torque before there is an indication of elastic deformation, and 100 times the amount before there is any permanent deformation. It was recommended that the original design of the special drive washer be modified to incorporate a tapered tab.

AD-15 012 Accession No.

Naval Proving Ground, Dahlgren, Va.
GUNFIRE TEST OF MODEL P2V-4 AIRPLANE CENTER SECTION SELF SEALING FUEL CELL INSTALLATION INCORPORATING A SELF SEALING OIL CELL MANUFACTURED BY U. S. RUBBER COMPANY, by H. P. Branscome. Final rept. Final rept. on Gunfire Test of Model P2V-3W Airplane and Model P2V-4 Airplane Self Sealing Fuel Cell Installations. 27 May 53, 8p. illus. table (NPG rept. no. 1133)

Six wounds, all of which sealed satisfactorily, were produced by five 0.50-cal AP projectiles fired into the oil cell. Round number 3 blew the drain plug free, and the oil was lost. Reinforcement or support was recommended for the area around the sump assembly access plate which was damaged during the test.

AD-16 330 Accession No.

Naval Proving Ground, Dahlgren, Va.
GUNFIRE QUALIFICATION TEST OF SELF-SEALING FUEL CELLS, by H. P. Branscome. Final rept. 25 Mar 53, declassified 15 June 54, 8p. illus. tables. (NPG rept. no. 1160) Unclassified report

(Not abstracted)

AD-16 848 Accession No.

Royal Aircraft Establishment (Gt. Brit.).
LEAKAGE TESTS ON SOLID AND INFLATABLE RUBBER SEALS DESIGNED FOR USE IN WIND TUNNELS, by D. L. T. P. Llewelyn-Davies and M. T. Carver. Jan 53, 9p. illus. (Technical note no. Aero 2219) (In cooperation with National Aeronautical Establishment (Gt. Brit.))

Leakage tests were conducted on an hour glass cross-section solid rubber seal designed for machined flange joints and a rubber tubing inflatable seal for gaps up to 0.10 in. between the edges of removable supersonic nozzles and the tunnel walls. The former satisfactorily sealed gaps up to 0.08 in. at ±1 atm from atmospheric pressure with a 5-16 in. sealing pressure; the leakage was less than 0.1×10^{-6} lb/sec/ft. Optimum results were obtained with the number of seal joints minimized and the seal fitting loosely in its groove. A minimum inflation of 30 psig was required for the inflatable seal to ensure adequate characteristics against ±1 atm dif- (over)

AD-16 848

ferences from atmospheric pressure. Leakage rates of 0.44×10^{-6} and 3.1×10^{-6} lb/sec/ft, respectively, were obtained for a 0.128-in. gap against a vacuum and pressure.

AD-18 810 Accession No.

Post Office Engineering Dept. (Gt. Brit.).
THE USE OF RADIOACTIVE MATERIALS TO FACILITATE THE LOCATION OF LEAKS IN THE SHEATHS OF GAS-PRESSURISED CABLES, by E. A. Speight and L. W. Hill. Feb 53, 18p. incl. table. (Research rept. no. 13112; [CRB ref. no. 53/2127]) Unclassified report

An examination was made of the possibility of using radioactive materials in the detection of leaks in the sheaths of ducted telephone cables. The study indicated that a method could be developed with the use of radioactive methyl bromide which would give an unequivocal indication of the cable length in which a leak existed. The sensitivity of the method would be several times greater than that of the best manometric methods, and would enable the whole of a 2000-m cable section to be tested in a maximum time of 24 hr.

AD-19 627 Accession No.

Lerner, Joseph, and Son, Inc., Brooklyn, N. Y.
SEALING TECHNIQUES FOR WAVEGUIDE AND COAXIAL ROTARY JOINTS, by C. R. Kuiper. Quarterly engineering progress rept. no. 3, 15 Feb-15 May 53. 15 June 53, 9p. illus. (Rept. no. D-102-3) (Contract DA 36-039-sc-42558)

(Not abstracted. See AD-12 108)

AD-21 274 Accession No.

Pitman-Dunn Labs., Frankford Arsenal, Philadelphia, Pa.
DISASSEMBLY TECHNIQUES FOR T2031 FUZES ASSEMBLED BY MEANS OF INJECTED METAL SEALS, by A. Sala and S. Lipson. Oct 53, 4p. illus. (Proj. no. TA1-6102; Memo. rept. no. MR-572) Unclassified report

Zone heating by high-frequency induction and by a conductive method was investigated in an effort to develop disassembly techniques for fuze components joined by injected metal seals. A comparison of the heating rates obtained by the 2 methods indicated greater efficiency with the conductive method with the inductive technique. The 500-w iron of the conductive apparatus raised the temperature of the fuze body to 210°F at a distance of 1 in. from the sealing groove in approximately 21 sec. The inductive method required 27 sec to produce the same temperature rise. The separation time for the fuze components and the hazards associated with the techniques are discussed. (See also AD-1046)

AD-21 275 Accession No.

Pitman-Dunn Labs., Frankford Arsenal, Philadelphia, Pa.
MECHANICAL PROPERTIES OF INJECTED METAL SEALS FOR THE T2031 FUZE, by A. Sala and S. Lipson. Sep 53, 4p. illus. (Proj. no. TA1-6102; Memo. rept. no. MR-571) Unclassified report

Static and dynamic shear properties were determined for injected metal seals made with two Pb-Sn-Bi alloys. The results showed that the shear and impact strengths are reduced with increasing temperatures. The shear strength data for both alloys were identical. The impact values, obtained with Newton's alloys, were slightly higher than those for the ternary eutectic. An ample factor of safety was observed with regard to the shear strength at the maximum operating temperature of the fuse (165°F). At this temperature, the maximum service stresses, applied to the joint, were calculated to be 128 psi; the shear strength was approximately 1100 psi. Although a marked reduction in the impact properties (over)

AD-21 275

occurred at 165°F, no. brittle zone or abrupt discontinuity in these properties was revealed. The rate of loading was more severe than that which will be applied to the fuse assembly in service. These considerations suggested that the elevated-temperature impact properties lie within an ample safety factor based on the service requirements of the fuse. (See also AD-21 274)

AD-22 068 Accession No.

Bendix Products Div., Bendix Aviation Corp., South Bend, Ind.
DEVELOPMENT AND TEST OF DIAPHRAMS FROM SYNTHETIC RUBBER MATERIALS, by Newton J. Lumm. Aug 53, 64p. incl. illus. tables (WADC Technical rept. no. 53-320)
(Contract AF 33(616)40; Continuation of Contract AF 33(038)3613) Unclassified report

Engineering subassembly tests, complete metering tests, and service tests were conducted on special and standard diaphragm materials. Test data were obtained which showed that the new ES-0613, ES-0616, and ES-0617 diaphragm materials are superior to the ES-0447 standard-material diaphragms in regard to ozone resistance and low-temperature flexibility. The new materials are equal to or better than the standard material with respect to life cycle, burst, flange crushing, flange sealing, and vibrational wear characteristics. Carburetor metering while using the new material dia- (over)

AD-22 068

phragms was satisfactory down to -65°F from the high cruise range to rated take-off powers. Minor deviations (in the rich direction) existed at the lower powers below -30°F. The tests were performed in a PR-100E3 master control without a poppet valve diaphragm. Service testing, conducted through build-up, test, setting, and engine run-in, showed the new material diaphragms to be at least equal in all respects to the current production diaphragms.

AD-23 155 Accession No.

Naval Engineering Experiment Station, Annapolis, Md.
GASKET MATERIALS FOR UNION FITTINGS OF 3000 PSI OXYGEN SYSTEMS, by Thomas H. Hake. 3 Apr 53 [8]p. incl. tables (EES rept. no. 1B066735) Unclassified report

Various grades of teflon, nylon, and synthetic-rubber gasket materials were tested to determine the most suitable material for use in union fittings of oxygen systems at 3000 psi. Circular and rectangular cross-section gaskets of the materials were subjected to oxygen aging and air line performance tests. Results indicate that longest life and most effective performance will be obtained by the use of teflon or nylon gaskets.

AD-24 488 Accession No.

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
EFFECT OF IMMERSION IN OIL AND IN WATER ON COLD COMPRESSION SETS OF OIL-RESISTANT GASKET STOCKS, by J. Oser. Technical rept. 12 Jan 53, 17p. illus. tables (Rept. no. 7-23 A) Unclassified report

(Not abstracted)

AD-25 808 Accession No.

Kraeuter-Weber Tool Co., Newark, N. J.
AN IMPROVED ROTARY, HIGH-SPEED SEAL, by A. Razdow. Quarterly rept. no. 3, 1 Mar - 19 June 53. 25 Nov 53, 1v. incl. illus.
(Contract DA 36-039-sc-5592) Unclassified report

In tests of carbon washers coated with MoS_2, the wear resistance was found to be improved. No conclusive results were obtained in test runs made with phosphate-covered steel washers. The female coupling was simplified and its weight was reduced by 0.005 oz. Special tools were designed for the assembly of the coupling to the shaft. The size of the ball bearing was decreased, and the lower bearing was replaced by a thrust bearing. Negative results were obtained in tests with stainless steel and phosphor bronze wave springs. Tests with 3 wave springs of 0.003-in. beryllium copper gave satisfactory results. The over-all weight and size of the seal were slightly reduced, and the seal housing was anodized. (See also AD-14 052)

AD-25 809 Accession No.

Kraeuter-Weber Tool Co., Newark, N. J.
AN IMPROVED HIGH-SPEED ROTARY SEAL, by A. Razdow. Interim rept. 1 Jan 52 - 28 Feb 53. 30 Apr 53, 52p. illus.
(Contract DA 36-039-sc-5592; Continuation of Contract W36-039-sc-44520) Unclassified report

An investigation was made to improve the leakproof bonding between the carbon washer and the retainer; test fixtures were built for this purpose. Tests were run with lubricating mixtures in an effort to decrease frictional heat and wear and to increase the life of the rotary seal. The coilspring was redesigned and replaced by a flat wave spring, and tests were made to determine the proper material. An important decrease was made in the over-all size and weight of the seal. Tests made on the miniaturized developmental model showed satisfactory results. Research was continued in an effort to simplify the operation and decrease the number of parts used in the diaphragm assembly and to eliminate, if (over)

AD-25 809

possible, the Teflon washer. Armstrong Cement A-6 was tested with unsatisfactory results. Adhesive Permafuse WM-35 was applied with good results. The Teflon ring could be eliminated and a larger carbon washer used which is cut down on the top in order to decrease the surface of the contact with the steel washer. The time for the manufacturing of the parts and assembly was decreased. A female coupling was simplified, and its weight was decreased. The design of the lapped steel washer was simplified, and some components were eliminated. The carbon assembly gave satisfactory results, and continuous observation of the behavior of the diaphragm is being made to study possible fatigue of the diaphragm material. (See also AD-14 052)

AD-25 907 Accession No.

Rock Island Arsenal Lab., Ill.
PNEUMATIC EQUILIBRATOR PACKINGS FOR OPERATION AT SUB-ZERO, NORMAL, AND ELEVATED TEMPERATURES, by L. R. Rassbeck. Technical rept. 7 Oct 53, 15p. illus. tables (Proj. no. TR-1031R, rept. no. 5; RIA Lab. rept. no. 53-2767) Unclassified report

Equilibrators for the 155-mm gun were tested to determine the best synthetic rubber packings and packing lubricant, as well as the best method of lubricating the packings for equilibrators which are operated at elevated, normal, and subzero temperatures. At -70°F, each configuration was gyma-cycled 20,000 c. At room temperature and 160°F, the configurations received 10,000 c. The life of O-ring packings was prolonged by continuous lubrication and the use of backup washers to prevent extrusion. Oil-saturated felt lubricators provided better packing lubrication than oil without a retaining aid. Special recoil oil, Army Spec 2-132, appeared more suitable than hydraulic Spec MIL-O-5606 (over)

AD-25 907

oil. Synthetic rubber O-rings with a Shore A durometer hardness of 75 to 80 seemed best for maximum life. Oil swelling up to 9% had no apparent effect on the performance of O-ring packings.

AD-27 587 Accession No.

Electronic Components Lab., Wright Air Development Center, Wright-Patterson Air Force Base, Ohio.
SEALING OF SYNTHETIC SAPPHIRE TO GLASS, by Emil Benz. Rept. on General Investigation of Electron Tubes. Jan 54, 6p. incl. illus. (WADC Technical rept. no. 54-18) Unclassified report

A new method of sealing synthetic sapphire to glass by using an electric arc is described. The sealing method is speedier and can be used by unskilled personnel. The process can be controlled better, which results in a more uniform sealing.

AD-27 728 Accession No.

Armour Research Foundation, Chicago, Ill.
EXPERIMENTAL INVESTIGATION OF OPTIMUM INCLOSURE FOR "O" SECTION SYNTHETIC RUBBER SEAL RING, by D. E. Gerberich, D. Kurtovich and others. Rept. for July 51-Dec 52. Aug 53, 121p. incl. illus. tables (WADC Technical rept. no. 53-319)
(Contract AF 33(038)26613) Unclassified report

For static sealing against leakage of iso-octane fuel at -65°F, 9 types of inclosures were tested by using artificially aged rings to simulate conditions after extended service. The rings were exposed to (1) extreme fluid-pressure fluctuations from 0 to 2000 psi and (2) moderate fluctuations from 0 to 100 or 1200 psi. Tests on the standard rectangular type inclosure (dimensions of Spec MIL-P-5315) show that it provides less satisfactory fuel-sealing characteristics than other types. In this inclosure the O rings appear to have insufficient squeeze (radial compression). Two inclosure types were closely equal for improved specification purposes for static (over)

AD-27 728

seals. These are a rectangular inclosure with 24% squeeze and an inclosure with inclined bottom. The latter shows better sealing properties for the severe laboratory test condition utilizing dry aged O rings, but is unsymmetrical in shape. From consideration of the laboratory test results alone, the inclined bottom type is recommended for future specification use. For testing of seals in dynamic application, a linear reciprocating motion was applied to the O ring test fixture, with a stroke length of 1/2 in. and a cyclic rate of 10 c/min. The recommended dynamic seal arrangement for best sealing properties includes chromium-plated metallic surfaces with 0.0005-in. radial clearance, and a rectangular inclosure shape with a 24% O ring squeeze value. Poor dynamic sealing properties were exhibited in the tests by all dry or presoaked aged O rings in all other inclosures at -65°F. (Extracted from report)

AD-27 886 Accession No.

Kearfott Co., Inc., Little Falls, N. J.
TYPE II, LOW-SPEED MECHANICAL-SEALED DRIVE, by Y. C. Shen. Nov 53, 16p. incl. illus. tables (WADC Technical rept. no. 53-476)
(Contract AF 33(038)3854) Unclassified report

"In an attempt to improve the efficiency of the mechanical sealed drive developed under this contract, a redesign was instituted in which the inner ball bearing races were made integral with the cranks and the outer races made integral with the housing. Straight sleeve bearings were used in place of the tapered inner bearings. In addition, the bellows were formed hydraulically. Tests on these units, however, showed them to be unsatisfactory. As a result, a second redesign was undertaken in which the sleeve bearings were replaced with ball bearings. Extensive tests on these units indicated that although they were capable of meeting the majority of the requirements of Exhibit No. MCREE-476, they failed to meet the following: (1) over-all length, (2) weight, (over)

AD-27 886

(3) torque load capacity, (4) no-load torque, (5) backlash. To meet the latter requirements it will be necessary to employ a more rugged construction and replace the cantilever loaded crank with a balance loaded oscillating crank arrangement." (Contractor's abstract)

AD-28 321 Accession No.

Battelle Memorial Inst., Columbus, Ohio.
B-36 INTEGRAL FUEL- AND OIL-TANK SEALANT. RESEARCH AND DEVELOPMENT PROGRAM, R. G. Heiligmann, P. B. Stickney and others. Final rept. 11 Jan 54, 66p. incl. illus. tables.
(Subcontract to Consolidated-Vultee Aircraft Corp., Contract AF 33(038)2182) Unclassified report

Work was initiated on the development of a nonpolysulfide sealant substitute for Thiokol-based sealants. A means for obtaining excellent bond strength was applied to conventional types of sealants, and a Pb-free cure which materially increased sealant pot-life was developed. However, no satisfactory sealant compound was developed from available polymeric materials other than the polysulfide rubbers. Copolymers of butadiene with methacrylic acid or acrylic acid when added to Thiokol-based sealants offered the greatest promise in low-temperature performance, while maintaining satisfactory resistance to aromatic fuels. Copolymers of butadiene (over)

AD-28 321

with ethyl azodicarboxylate were oily-to-waxy rather than rubbery and were not satisfactory for sealant application. Insufficient quantities of silane-polysulfide polymers were obtained because of difficulty in producing an intermediate.

AD-28 429 Accession No.

Electrical Engineering Research Lab., U. of Illinois, Urbana.
ELECTRICAL LOSSES IN CERAMIC-TO-METAL SEALS, by E. W. Ernst and O. T. Purl. Feb 54, 43p. incl. illus. tables (Technical rept. no. 2)
(Contract AF 19(604)524) Unclassified report

An investigation of the RF electrical losses in ceramic-to-metal seals was devoted to the problem of determining the loss of a base metal covered with a very thin coating of a material of unknown electrical properties. The property measured was the effective resistance of the interface for Mo-Mn, Ti hydride, and compression seals. The measurements were made in cylindrical cavities because of the advantageous seal geometry. The system designed to measure the interface loss is essentially an admittance bridge arrangement, and the RF test signal is supplied by a 2K25 klystron oscillator which is frequency modulated about the resonant frequency of the cavities. Details of the calibration of the (over)

AD-28 429

bridge are discussed, and the details of the calculation of surface resistivity are appended.

AD-28 761 Accession No.

Aeronautical Materials Lab., Naval Air Experimental Station, Philadelphia, Pa.
INVESTIGATION OF USE OF ROGERS CORP. GASKETS IN FUEL SYSTEMS, by A. Dubrow. 5 Feb 54, 8p. illus. table (Rept. no. AML NAM AE 6237.1)
Unclassified report

AD-28 862 Accession No.

Picatinny Arsenal, Dover, N. J.
DEVELOPMENT OF AN IMPROVED LUTING COMPOUND FOR USE IN REUSABLE METAL CONTAINERS, by R. H. Evans and C. C. Banta. Mar 54 [26]p. incl. illus. tables (Proj. no. CX-11, item H; Technical rept. no. 1970) Unclassified report

The luting compound specified under Spec MIL-C-20551 was found unsatisfactory for general use. Forty-three compounds were evaluated for conformance to the requirements of a proposed improved luting compound. Three compounds met all these requirements, compound 1511 submitted by the Permatex Corp. and compounds 11849AZ and 11849BA submitted by the Flintkote Co. In the evaluation the samples were troweled into simulated copper glands and subjected to storage-and cover removal tests from -65° to +160°F. Based on the satisfactory performance characteristics and chemical and physical tests a purchase description (PA-PD-398) was prepared.

AD-29 114 Accession No.

North American Aviation, Inc., Downey, Calif.
TEST OF NEOPRENE "W" HIGH-TEMPERATURE HYDRAULIC SEALS, by C. E. Hamlin. 1 Mar 54, 22p. incl. illus. (Rept. no. EM-612)
(Contract W33-038-ac-14191) Unclassified report

"Neoprene 'W' O-rings, in combination with various backup rings, were tested in a balanced actuator at 3000 psi in conjunction with an orthosilicate ester hydraulic fluid. The actuator piston was mechanically reciprocated at 2 cps, and tests were made at temperatures from 300 to 550 F for a maximum of 100,000 cycles. The neoprene 'W' O-rings, backed up by either 5R16 teflon rings or glass-loaded teflon rings were unsatisfactory at temperatures of 350 F or above, because of failure of the teflon. A split spiral aluminum bronze backup ring was developed and fabricated at North American Aviation, Inc. Neoprene 'W' O-rings backed up by these aluminum bronze rings are satisfactory for extended duty at temperatures up to 400 F, and for limited duty to 550 F." (Contractor's abstract)

AD-29 801 Accession No.

Material Lab., New York Naval Shipyard, Brooklyn.
INVESTIGATION OF THE APPLICABILITY OF THE SEALING PRESSURE TEST APPARATUS TO THE LOW TEMPERATURE EVALUATION OF NATURAL AND SYNTHETIC ELASTOMERS, by J. Z. Lichtman. Final technical rept. 1 Oct 53, 8p. illus. tables (Rept. no. 4855-11) Unclassified report

The test apparatus, which was investigated for use in evaluating the low-temperature properties of elastomeric gasket materials, was found to be unsatisfactory for determinations of either sealing pressure or follow-up upon decreasing the specimen indentation. The samples used consisted of 1 standard and 1 cold-resisting elastomer compound each of Neoprene, Buna-S, Perbunan-26, Hevea, Butyl, and Thiokol-FA; the samples were in slab form (12 x 12 x 0.5 in.). Sealing-pressure determinations were made by utilizing a source of high-pressure air which could be regulated and measured up to 500 psi. Long-time and short-time conditioning tests were conducted with 3 specimens of each elastomer.

AD-29 951 Accession No.

Material Lab., New York Naval Shipyard, Brooklyn.
INVESTIGATION OF THE APPLICABILITY OF THE SEALING PRESSURE TEST APPARATUS TO THE LOW TEMPERATURE EVALUATION OF NATURAL AND SYNTHETIC ELASTOMERS, by J. Z. Lichtman. Final technical rept. 1 Oct 53 [32]p. incl. illus. tables. (Rept. no. 4855-11) Unclassified report

(Not abstracted. See AD-29 801)

AD-30 136 Accession No.

Electronics Research, Inc., Evansville, Ind.
A SERIES OF HERMETICALLY SEALED FLUSH MOUNTED ECM ANTENNAS. Final engineering rept. Feb 54, 36p. incl. illus. (Rept. no. GAF-30/1) (Contract AF 33(600)6715) Unclassified report

Twelve flush-mounted ECM antennas covering the 300- to 11,000-mc range are described. An annular slot was utilized with 6 antennas to span the transmitting range of 200 to 3100 mc with a VSWR of 2.5:1 or less. A similar annular slot configuration was utilized with 6 receiving antennas to span the 270- to 11,000-mc range with a VSWR of 5:1 or less. The antennas which were designed for mounting in B-36 type aircraft were curved to a 75-in. radius and were of light-weight construction. A foam potting process was developed for hermetically sealing the larger cavities. The loss of the hermetic seal does not make the antenna inoperative. The hermetically sealed antennas were tested to a differential pressure equivalent to 50,000 ft.

AD-30 373 Accession No.

Rock Island Arsenal Lab., Ill.
PACKINGS AND CYLINDERS FOR HYDRO-PNEUMATIC RECOIL MECHANISMS, by C. E. Evans and C. Olsen. Technical rept. 5 May 54, 10p. illus. tables. (Proj. no. TR1-1031; RIA Lab. rept. no. 54-1649) Unclassified report

(Not abstracted)

AD-30 417 Accession No.

Power Plant Lab., Wright Air Development Center, Wright-Patterson Air Force Base, Ohio.
PRESSURIZED FLEXIBLE SELF-SEALING FUEL CELL DEVELOPMENT, by J. Weil. 1 Mar 54, 88p. incl. illus. tables (Technical note no. WCLP-54-40) Unclassified report

Includes AD-30 417(a).

An investigation was made to obtain self-sealing fuel cell installations operational at internal pressures as high as 7.5 psig and utilizing flexible cell constructions. Acceptance gunfire tests at a pressure of 2.5 psig indicated that the installation under development provided somewhat lesser protection against 50-cal AP ammunition than the conventional unpressurized system. No satisfactory protection was afforded under subjection to 20-mm AP projectiles. Support afforded the fuel cell by the backing board was of prime importance in obtaining a satisfactory installation. Acceptance gunfire tests (over)

AD-30 417

at a pressure of 5 psig were unsatisfactory. Results indicated that it will be extremely difficult to obtain a practical installation incorporating flexible fuel cells operational at pressures in excess of 2.5 psig.

AD-33 308 Accession No.

Detroit Arsenal, Center Line, Mich.
OIL SEALS, by Alex M. Calabuca. 7 Apr 54, 4p. illus. (Rept. no. 2889) Unclassified report

(Not abstracted)

AD-33 887 Accession No.

Instrumentation Lab., Mass. Inst. of Tech., Cambridge.
A STUDY OF SEAL FRICTION IN HYDRAULIC SERVOMECHANISMS, by M. S. Feldman and E. G. Wriedcn. Master's thesis. May 54, 78p. incl. illus. tables, 21 refs. (Rept. no. T-62) (Contract AF 33(616)2039) Unclassified report

The frictional characteristics of piston and rod seals used as hydraulic actuators were studied separately. The static and kinetic friction force levels were investigated as well as the effect of various seals on the dynamic characteristics of the servo. Seals of rubber, cast iron, and Teflon were tested. The 2-element Teflon piston rings exhibited the lowest static and kinetic frictional levels. The break-free force for the wide Teflon rings was approximately 1/2 that for the narrow Teflon rings. The narrow Teflon rings exhibited the lowest static and kinetic frictional levels for the rod seals tested. All of the rod seals affected the dynamic (over)

AD-33 887

response of the servo to approximately the same degree; but it was noted that a high level of static friction will strongly affect the angle resolving qualities of the servo. Consequently, the narrow Teflon rod seals gave the best results from a purely frictional standpoint. However, care must be taken in designing a servo using these seals since they may cause instability.

AD-34 171 Accession No.

Krautter-Weber Tool Co., Newark, N. J.
AN IMPROVED ROTARY, HIGH-SPEED SEAL. Final rept. 1 Jan 52-15 Dec 53. 23 Dec 53, 41p. illus. (Contract DA 36-039-sc-5592) Unclassified report

Investigations were conducted to improve the leak-proof bond between the C washer and retainer. Tests were run with lubricating mixtures in order to obtain a decrease in wear and frictional heat, and to increase the life of the seal; improvement was attained in the lubrication in the ball bearings and on the sealing contacts. The Al nut, steel washer, Teflon ring, and 1 spanner ring were eliminated in a redesign of the lapped-steel washer. The lower ball bearing was replaced by a thrust bearing in order to cushion the thrust of the seal springs against the bearings. The coil spring was replaced by a flat wave spring formed of beryllium copper. The surface of the Al housing was anodized for adequate protection, the upper spring holder was eliminated, and the lower spring holder was simplified. The female coupling was simplified, and its weight was reduced. Tests were (over)

AD-34 171

run on wave springs made of steel and phosphor bronze. A unit of 3 wave springs was finally adopted. (See also AD-25 808, AD-25 809)

AD-34 750 Accession No.

Detroit Arsenal, Center Line, Mich.
TIRE VALVE CORE TEFLON BARREL GASKET PERFORMANCE TEST, by C. F. Dandareau, Jr. Final technical rept. 25 May 54, 7p. incl. illus. (Rept. no. 2944) Unclassified report

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AD-35 302 Accession No.

General Electric Co., Owensboro, Ky.
THE DEVELOPMENT OF NEW SEALING TECHNIQUES FOR MINIATURE TUBES, by C. E. Horton. Quarterly interim technical rept. no. 4 for period ending 31 Aug 53. Dec 53, 37p. incl. illus. tables. (Contract NObsr-52534) Unclassified report

"Effort was concentrated on the metal ring phase of the project during this quarter in order to bring this technique to the same state of development as the solder frit method. Two ring designs, two sealing alloys, and a number of ring glazes have been evaluated by the process of sealing 950 type 6CB6 tubes. Mechanical and electrical tests, including life tests, have been made on these tubes. Equipment for automatically sealing tubes in quantity is nearly complete. Work continues at a reduced pace on the solder frit method. New frits were tried, and a die for an improved stem design was produced." (Contractor's abstract) (See also AD-4139)

AD-36 319 Accession No.

Armour Research Foundation, Chicago, Ill.
BONDING MATERIALS, METALLIC MATING SURFACES; LOW RF IMPEDANCE, by V. Pulsifer. Summary rept. 10 May 51 - 10 May 53. Jan 54, 57p. incl. illus. tables.
(Contract AF 33(038)23583) Unclassified report

An attempt was made to develop a gasket material for containers housing RF emitting apparatus which will prevent RF energy from escaping at joints in closed containers and also form a gas-tight joint. An excellent gasket material was made by molding, spraying, and dipping card cloth in rubber and cutting away the canvas and excess wire and rubber. Tests showed that: (1) gasket materials used for RF shielding and sealing must have unconnected paths to conduct the current across a joint; (2) paths must be evenly spaced over the surface of the gasket material so as to make contact with the gasket surfaces when pressure is applied; (3) conductors used in gaskets must be relatively stable chemically (over)

AD-36 319

and have good electrical conductance; (4) conductors perform best if they are made in a hard temper to break through the surface of the flanges; (5) the number of conducting paths per square inch is important in controlling local contact pressures; (6) there is an optimum number of conductors per square inch for every combination of gasket design and conducting material; and (7) the mechanical properties of the metal elements and the sealing agent must be such that after sufficient clamping pressure has been applied to establish electrical conducting paths, the remainder shall be sufficient to provide sealing pressure. A preformed gasket design was developed which reduced the impedance at 2.6 mc to 6 x 10^{-7} ohm at 50-psi clamping pressure and standard testing conditions. A gasket cement was developed which reduced the impedance to the order of 6 x 10^{-9} ohm under the same conditions.

AD-36 428 Accession No.

Material Labs., Puget Sound Naval Shipyard, Bremerton, Wash.
INSTALLATION OF SYNTHETIC RUBBER DECK CAULKING COMPOUND ABOARD THE USS YORKTOWN (CVA-10) by W. B. Low and J. N. Wessel. 9 Feb 53 [8]p. incl. illus. table (Technical rept. no. R-314-53) Unclassified report

AD-36 452 Accession No.

Naval Gun Factory, Washington, D. C.
SPRAYTIGHTNESS TEST OF REDESIGNED FIRING CIRCUIT TERMINAL BOX FOR ROCKET LAUNCHER MK 108 MODS, by E. L. Dickerson, Jr. and Paul D. Huffman. 24 Feb 54, 3p. (Technical rept. no. NGF-T-5-54; NAVORD no. 9902) Unclassified report

The redesigned terminal box was subjected to a 30-min spray test which simulated conditions encountered during spraying of the guide. Disassembly of the box after testing revealed about 1/2 pint of water; this presence of water was attributed to the leakage around the asbestos gasket which is used as a sealing surface between the cover and the box. Recommendations were made that (1) the cover be redesigned to form a more rigid member and (2) either a new type gasket of a softer material be provided or the present gasket be softened.

AD-36 456 Accession No.

Naval Gun Factory, Washington, D. C.
TEST OF HIGH PRESSURE HYDRAULIC PACKINGS FOR HYDROPNEUMATIC COUNTER-RECOIL SYSTEMS OF GUN MOUNTS, by W. J. Tocha and E. L. Dickerson. 27 Apr 54, 11p. illus. (Technical rept. no. NGF-T-4-54; NAVORD rept. no. 9901) Unclassified report

High-pressure hydraulic packings were tested with a 5-in/38 hydropneumatic counterrecoil cylinder Mk 1 Mod 4 in order to establish working values and friction requirements to be used in revising ordnance specifications (OS 749). A 5000-cycle leakage-endurance test, a friction-evaluation test, and a physical-properties test were conducted. Recommendations are discussed.

AD-36 564 Accession No.

Rock Island Arsenal, Ill.
PACKINGS FOR RECOIL MECHANISMS TESTS NO. 37, 37A, 37B, by Clement Olsen and Joe L. Coppersmith. Technical rept. 8 June 54, 39p. incl. illus. tables (Proj. no. TR1-1031; Rept. no. 54-2041) Unclassified report

Three 105-mm howitzer recoil mechanisms were tested to evaluate Teflon for packings and aluminum alloy cup rings for replacing leather and silver. One mechanism was subjected to 10,065 c on a shop gymnasticator, 200 c in cold room and 594 c on the powder-type testing machine. The other mechanisms were tested by firing 19,981 rounds of ammunition in one and 25,129 rounds in the other. Teflon with aluminum ring replacements showed a greater life cycle to recoil mechanisms. Teflon packing with aluminum rings was recommended to replace leather packings with silver rings on all rebuilt and new recoil mechanisms. The replacement of the non-oil-resistant rubber RIA formula 32F by RIA formula no. 441 and of the oil index rubber packing by a Teflon shroud and rubber packing was also recommended.

AD-37 003 Accession No.

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
EFFECT OF EXPOSURE TO WEATHER AT POINT BARROW, PUGET SOUND, AND MARE ISLAND ON HARDNESS AND COLD COMPRESSION SET OF GR-S, NATURAL RUBBER, AND NEOPRENE VULCANIZATES, by A. E. Barrett. Technical rept. 31 July 53, 17p. illus. tables (Rept. no. 69-40)Unclassified report

(Not abstracted)

AD-37 096 Accession No.

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
DEVELOPMENT OF A HYCAR 1014 STOCK FOR COLD-RESISTANT OIL-RESISTANT GASKETS, by J. Oser. Technical rept. 18 Sep 52, 8p. illus. tables. (Rept. no. 7-20) Unclassified report

"A cold-resistant oil-resistant Hycar 1014 stock, designated by the code number E-9-518, was developed by the Rubber Laboratory. This stock met the requirements of specification MIL-R-900A for hardness and compression set at -35°F; all the requirements of specification MIL-G-2765 except that pertaining to ultimate elongation; and the requirement, agreed upon by the Bureau, of a minimum tensile strength of 1000 psi and a minimum ultimate elongation of 300%. A trial run of mixing, extruding and curing of this stock was conducted in the factory of the Rubbercraft Corporation, Torrance, California. This trial proved the suitability of the stock for industrial production." (RL abstract)

AD-37 100 Accession No.

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
TEST FOR EVALUATING THE TENDENCY OF SYNTHETIC RUBBER GASKET MATERIALS TO LOSE PLASTICIZERS BY VOLATILIZATION, by J. Oser. Technical rept. 31 Aug 53 [26]p. incl. illus. tables. (Rept. no. 167-3) Unclassified report

A volatilization apparatus for determining the evaporation of plasticizers from gasket materials consisted of a 6-ft-long, 1-in.-diam steel air duct in which the specimens were exposed to a warm air current. Air velocity and temperature in the apparatus, as well as the size, thickness, and number of specimens which were exposed simultaneously in the apparatus, were found to affect the rate of plasticizer evaporation. Reproducible heat-loss determinations (to about 1 mg per specimen) were obtained with the apparatus when these factors were controlled. A fair correlation was shown between the plasticizer loss from GR-S gasket stocks during 60 days of (over)

AD-37 100

weather aging and the plasticizer loss during exposure in the apparatus at 260°F for 30 min for for 3 hours. On the basis of these results, a test for evaluating the tendency of vulcanizates to lose plasticizers by volatilization is recommended for inclusion in Spec MIL-R-900A for medium-soft synthetic-rubber gaskets.

AD-37 218
(Unannounced report) Accession No.

National Advisory Committee for Aeronautics, Washington, D.C.
COMPATIBILITY OF PENTABORANE WITH MATERIALS USED FOR SEALS, GASKETS, AND CONSTRUCTION, by Samuel Kaye and Frank V. Sordyl. 19 July 54, 16p. incl. illus. tables (Research memo. RM E54E12) Confidential report

"The compatibility of pentaborane with 24S-T and 17S-T aluminum, aluminum foil, and 50-50 tin-lead solder was tested in the temperature range -30° to -50° F, at 90° F, and at 70° F. The compatibility with Buna, butyl, GR-S, neoprene, silicone, Thiokol, and natural rubbers, and also asbestos paper, graphite asbestos, Vellumoid, and graphite gasket materials was tested. The metals, Teflon, and Kel-F were not attacked under these conditions. The gasket materials also appeared satisfactory, but pentaborane extracted some binder material from graphite packing; the rubbers deteriorated." (NACA abstract)

AD-37 218

Classification Changed UNCLASSIFIED
Authority: NASA DTD 29 Sept 59
BY M. Stephens 1/8/60
NAME OF INDIVIDUAL MAKING CHANGE DATE

AD-37 705 Accession No.

Forest Products Lab., Madison, Wis.
PACKAGING REQUIREMENTS FOR BEARINGS. PART 6. THE PERFORMANCE OF THREE TYPES OF SEALS FOR VARIOUS INTIMATE WRAPS WITH MIL-G-3278 GREASES (SERIES E) by M. A. Taras. Mar 54, 26p. incl. illus. tables (WADC Technical rept. no. 53-38, pt. 6)
(Contract AF 18(600)103) Unclassified report

"A study of the performance of three methods of closing various intimate wraps containing ferrous bearing parts coated with grease complying with Specification MIL-G-3278 was conducted by the Forest Products Laboratory. The three methods for closing the intimate wraps included the drugstore fold, heat sealing, and the use of an aluminum band. All specimens were enclosed in water-vaporproof pouches complying with Specification MIL-B-131A and were exposed to a 3-week cycle of temperatures ranging from -65° to 160°F. Following the cyclic exposure, the specimens were stored for 60 (over)

AD-37 705

days at 160°F. and 92 percent relative humidity. Results indicate that the effectiveness of the three sealing methods investigated is largely dependent upon the compatibility of the intimate wraps and the greases used. When both greases are considered, the three sealing methods appear to be equally effective. Corrosion when present, however, was more severe in packages using the drugstore fold method. In general, the greases used hardened upon exposure. Nylon, however, was the only intimate wrap that prevented any of the greases from hardening. No reason for this reaction was apparent. The data also indicate that one of the greases was less deleterious to the vinyl coating of the pouch material used in this test than the other." (Contractor's abstract)
(See also AD-27 685, AD-27 931)

AD-38 529 Accession No.

Industrial Lab., Mare Island Naval Shipyard, Vallejo, Calif.
DEVELOPMENT OF SPECIFICATION FOR LOW TEMPERATURE OIL-RESISTANT SYNTHETIC RUBBER GASKET MATERIAL. EVALUATION OF PIONEER RUBBER STOCK NO. 44'5627. MATERIAL EXHIBITED BY PIONEER RUBBER MILLS, SAN FRANCISCO, CALIFORNIA, by Y. D. Izu. 12 Mar 53 [23]p. incl. illus. tables (Rept. no. 5421-52)
Unclassified report

AD-41 362 Accession No.

Bjorksten Research Labs., Inc., Madison, Wis.
DETERMINATION OF LEAKAGE VALUES OF SEALS, by Earl A. Meyer, Robert J. Roth and others. Nov 53, 151p. incl. illus. tables (WADC technical rept. no. 54-16)
(Contract AF 33(038)15981) Unclassified report

"The rate of air leakage through various seals in aluminum, brass, and steel specimens was determined quantitatively with one side of the seal evacuated and the other side at one atmosphere. Fusion, adhesive, gasket and lapped type seals on 449 specimens were tested. Leakage rate measurements were made in both directions through the seals before and after various environmental tests of pressure, humidity, vibration and temperature cycling. The various types of seals were evaluated on the basis of their leakage both before and after specified environmental exposures. For these measurements two leakage-rate meters were designed and constructed to evaluate leakage rates of 1 to 3,000,000 standard cc. of air/year. Both meters are (over)

AD-41 362

fully described and drawings for their construction are included. An RCA-1945 Hydrogen Sensitive Ion Gauge was used to measure the smaller leakage rates; and the rate of movement of a water plug in a calibrated capillary was used for the larger leaks. The non-destructive testing of electronic components on the production line prior to a simple sealing procedure is described and a number of components supplied by WADC were tested." (Contractor's abstract)

AD-41 652 Accession No.

Naval Gun Factory, Washington, D. C.
SPRAYTIGHTNESS TEST OF FIRING CIRCUIT TERMINAL BOX FOR ROCKET LAUNCHER MK 108 MODS. 13 Sep 54, 4p. illus. (Technical rept. no. NGF-T-67-54; NAVORD rept. no. 4800)
Unclassified report

A comparison was made between the relative sealing properties of 2 gasket materials, Silastic 80 and Silastic 675, when used on the original and the redesigned firing circuit terminal boxes for the Mk 108 rocket launcher. A series of spraytightness tests were performed on 2 boxes containing 12 and 8 hold down bolts, respectively. Both of the materials formed an effective seal against water leakage. The Silastic 675 gasket was used in spraytightness tests on a redesigned firing box with 8 bolts; no leakage of water was evident. After exposure to 400°F, submersion and spraytightness tests were also conducted on the gaskets. As a result of the tests, the recommendation was made that the (over)

AD-41 652

original firing circuit terminal box and new gasket material, Silastic 675, be used on the Mk 108 rocket launcher.

AD-41 961 Accession No.
(Unannounced report)

Naval Engineering Experiment Station, Annapolis, Md.
VALVE-STEM PACKINGS FOR HIGH PRESSURE AND HIGH TEMPERATURE STEAM SERVICE, by R. F. Stone. 3 Aug 53, 1p. (EES rept. no. 010126)
Unclassified report

AD-42 033 Accession No.

Wichita U. School of Engineering, Kans.
TESTS OF A LABYRINTH-SEAL MOCKUP, by Richard E. Wallace and Paul Morris. July 54, 22p. incl. illus. (Engineering rept. no. 151)
(Contract AF 33(616)2330) Unclassified report

The labyrinth-seal mock-up was made of wood with the labyrinth elements cut from 1/8-in.-thick bonding plywood. One side of the seal was made variable so that the overlap could be decreased from 2.25 in. to 0.75 in. in 3/8-in. increments corresponding to the element gap. The number of elements was varied by removing downstream plywood elements, which were made tight slip fits to the side-piece slots. Figures are presented of the weight-flow variation as a function of the overlap-to-gap ratio with various numbers of elements and for gage pressures from 2 to 11 in. of mercury. The curves are all similar and indicate that an overlap-to-gap ratio of 4 provides the optimum operating condition for the seal. Experiments revealed (over)

AD-42 033

that the weight flow and power consumed in the seal will permit satisfactory operation of the unit to critical pressure ratios. The mock-up experiments only qualitatively indicate the basis of a prototype labyrinth seal.
(See also AD-18 162)

AD-42 477 Accession No.

Massachusetts Inst. of Tech., Div. of Industrial Cooperation, Cambridge.
ANALYSIS OF THE ROTARY REGENERATOR FOR GAS-TURBINE-CYCLE APPLICATIONS; AND AN ANALYSIS AND INVESTIGATION OF ROTARY REGENERATOR SEAL LEAKAGE, by D. B. Harper. Aug 54, 43p. illus. 66 refs (Technical rept. no. 5) (Contract N5ori-7862) Unclassified report

An analysis is presented of the effect of the rotary regenerator on the gas-turbine cycle, and results are reported of an experimental investigation of seal leakage flow. The analysis shows that even though a definite small leakage between high- and low-pressure sides must exist, this condition will not detract from the size advantages of this heat-exchanger type. A rotary regenerator was sized for a specific gas turbine-plant requirement, and the effect of choice of pressure drop and leakage rate are shown by design-point curves. The best over-all rotary regenerator arrangement has a (over)

AD-42 477

rotor containing the matrix and a system of rotating compartmented ducts leading out to small seal shoes. The seal shoes can be pressure-balanced by a diaphragm or piston which can be pressurized by a port in the seal shoe. The location of the port along the pressure gradient depends on the seal and piston areas. The loading force should be just sufficient to keep the seal in close contact with the rotor compartments without causing excessive friction.

AD-42 776 Accession No.

Detroit U., Mich.
STRESS RELAXATION OF GASKET MATERIALS AND EVALUATION OF A TEST FIXTURE, by L. S. Kowalczyk. 1954, 117p. illus. 48 refs. (Contract DA 20-089-ORD-36608) Unclassified report

A literature survey was made of methods for measuring the stress relaxation of gasket materials. Stress-relaxation values were obtained and plotted for 22 commerical gasket materials using the Farnam-Cole relaxometer, type RT, model D. All the materials gave smooth stress-relaxation curves, with the rates of stress relaxation appearing very large at the beginning of the test. The conditioning of the gasket material generally affected the curves. Study was made of the influence of the following factors on stress relaxation: temperature, major load, sample thickness, time of relaxation, time of first reading after compression, machine operator, and reloading and locking up of the sample in the capsule. The Farnam-Cole relaxometer (over)

AD-42 776

yielded reproducible results, was applicable to all nonmetallic gasket materials, and was recommended for stress-relaxation tests of gasket materials. A recommended procedure for laboratory relaxation tests is presented.

AD-43 419 Accession No.
(Unannounced report)

Material Lab., New York Naval Shipyard, Brooklyn.
TYPE APPROVAL EVALUATION OF SYNTHETIC RUBBER GASKET MATERIAL MANUFACTURED BY THE GARLOCK PACKING COMPANY, PALMYRA, NEW YORK. Final technical rept. 27 Mar 53 [5]p. incl. table (Rept. no. 4508-241) Unclassified report

AD-43 502 Accession No.
(Unannounced report)

Material Lab., New York Naval Shipyard, Brooklyn.
EVALUATION OF GASKET MATERIAL, STOCK NO. E-9-480 DEVELOPED BY THE RUBBER LABORATORY, MARE ISLAND NAVAL SHIPYARD, VALLEJO, CALIFORNIA. Final technical rept. 8 Apr 53 [6]p. incl. tables (Lab. proj. no. 5451) Unclassified report

AD-43 690 Accession No.

Naval Engineering Experiment Station, Annapolis, Md.
GARLOCK TYPE CA51A-2 MECHANICAL SEAL FOR GASOLINE-PUMP SERVICE SUBMITTED BY GARLOCK PACKING COMPANY, PALMYRA, NEW YORK, by R. F. Stone. 21 May 54, 2p. illus. (EES rept. no. 010067) Unclassified report

"Type CA51A-2 mechanical seal, manufactured by the Garlock Packing Company, was tested to determine its suitability for use in gasoline pumps operating at stuffing-box pressures between 5' and 25' head of aviation gasoline. The seal was found to be suitable for this service." (NEES abstract)

AD-46 159 Accession No.

Johns Hopkins U. School of Engineering, Baltimore, Md.
LEAKAGE TESTS FOR OPTICAL INSTRUMENTS AND LEAKAGE TESTS FOR ELECTRICAL AND ELECTRONIC INSTRUMENTS, by H. E. Hoelscher. Final rept. Sep 54, 34p. illus. (Contract DA 36-038-ORD-5309) Unclassified report

The techniques, devices, and methods under test included the corona discharge method, the immersion technique, the Magna Flux Corp. method, the gelatin bubble method, and the agar-agar film technique. Of the film formers studied as possible strippable coatings, the gelatin solutions were the most satisfactory. Gelatin was used by adding Dow-Corning Antifoam A to reduce the amount of foaming and the formation of spurious air bubbles which could not be eliminated otherwise. Gelatin at 12.5% by weight in H_2O provided the optimum concentration; Antifoam A was added in an amount between 0.5 and 1% of the total solution by weight. The (over)

AD-46 159

gelatin film technique was useful only on flat aluminum plates; it was not useful on curved surfaces. The gelatin technique was not reproducible owing to the effect of the gelatin film on the leak pore itself and the sensitivity of the kinetics of bubble formation to film properties; the technique was useful as a primary calibration method. The agar-agar film was successful for detecting the existence of leak points in a wide variety of apparatus. The test is effected by coating the test piece as rapidly as possible with the agar-agar solution, admitting CO_2 pressure to the inside of the test piece, adjusting the pressure to 15 psi gage, and allowing the test piece to remain under pressure for the development time. This test is rapid, simple, and easy to apply, and requires no specialized apparatus nor highly skilled personnel. The test can be used as a qualitative check, and with additional effort applied to the calibration through laboratory studies can be used as a quantitative method with equal facility.

AD-47 411 Accession No.
(Unannounced report)

North American Aviation, Inc., Los Angeles, Calif.
RECOMMENDED DIMENSIONS FOR "O" RING TYPE METAL-FACED FUEL CELL FITTINGS. 31 July 52, 6p. incl. illus. (Rept. no. NA-52-793) Unclassified report

AD-47 412 Accession No.

Society of Automotive Engineers, Inc., New York.
O-RING GROOVE CONFIGURATION FOR USE WITH MS29513 O-RINGS AS STATIC FACE SEALS, by Philip H. Jones. Rept. on Aircraft Fuel and Oil Systems and Equipment. 28 Apr 54, 4p. illus. (Rept. no. SAE-A18-54-1) Unclassified report

AD-47 671 Accession No.

Consolidated Vultee Aircraft Corp., Fort Worth, Tex.
EVALUATION OF B-36 INTEGRAL FUEL AND OIL TANK DESEALANTS, CLEANERS AND SEALANTS.
Final rept. Vols. 1 and 2, Oct 52 - Apr 54. 28 Apr 54, 2 vols. incl. illus. tables (Rept. no. FZA-36-273)
(Contract AF 33(038)2182) Unclassified report

"All leading materials available at the present time for use in descaling, cleaning, and sealing aircraft integral fuel and oil tanks were evaluated in Convair's Engineering Chemical Laboratory and their respective properties were obtained. In addition, improved materials of this same type, obtained from Turco Products, Inc., Minnesota Mining and Manufacturing Company, and Battelle Memorial Institute, were compared with standard products. These three companies were under subcontract to Convair for development of improved tank desealing, cleaning, and sealing products. As a result of this program it is concluded: (1) that satisfactory desealing and cleaning materials have been developed, (over)

AD-47 671

and (2) that integral tank sealants for use in B-36 airplanes were considerably improved over those previously available. Leak-proofing qualities and service-life performance of the newly-developed sealants are expected to exceed by a considerable amount those available with current sealants. The findings of this program may also prove valuable in sealing integral tanks of all types of Air Force aircraft." (Contractor's abstract)

AD-47 739 Accession No.

Applied Physics Lab., U. of Washington [Seattle]
AN INVESTIGATION OF CABLE SEALS I, by J. B. Morrison. 1 Mar 54, 87p. incl. illus. tables, 11 refs. (Technical rept. no. APL/UW/TE/54-41)
(Contract NOrd-12937) Unclassified report

An investigation was conducted on the fundamental mechanics of cable seals employing flexible packings. Investigation revealed that the mechanical properties of rubber make it a highly desirable packing material for cable glands. Results of testing indicate that the use of radial interference between the packing and the cable and between the packing and the gland wall brings about automatic sealing action of a rubber packing, whereas if clearance is provided the sealing action depends upon the correct initial tightening of the gland nut. No valid justification was found for the use of rubber packings of trapezoidal cross section as against those having a rectangular cross section. There is less possibility of leakage in the case where the downstream end face of (over)

AD-47 739

the packing is supported directly by the gland body than in the case where a separate washer or bushing is used between these surfaces. Substitution of interference packings of rectangular cross section improved the effectiveness of existing cable glands. Tentative recommendations are given for the design of interference-type packing glands.

AD-48 310 Accession No.

Naval Proving Ground, Dahlgren, Va.
GUNFIRE TESTS OF MODEL A3D-1 AIRPLANE FUSELAGE SELF SEALING FUEL CELL INSTALLATIONS, by H. P. Branscome. Final rept. 18 Dec 54, 1v. incl. illus. tables (NPG rept. no. 1314)
Unclassified report

Gunfire resistance tests were conducted on the sections of the production fuselage of the model A3D-1 aircraft which contain the forward and after fuel-cell installations, to determine (1) whether the self-sealing performance of the fuel cells and the integrity of the cell liners, seams, and joints would satisfactorily conform to the requirements established by Spec. MIL-T-5578A when the installation is subjected to 0.50-cal AP and 20-mm practice projectile impacts; and (2) whether the supporting and surrounding structure for the fuel cells would satisfactorily resist excessive structural damage resulting from the hydraulic surge of fuel incident to projectile impacts. Each of the 17 qualifying wounds in (over)

AD-48 310

the forward fuel cell and the 20 qualifying wounds in the after cell exhibited a satisfactory self-sealing performance. The forward fuselage satisfactorily resisted excessive structural damage and the forward bulkhead of the after fuselage installation suffered excessive damage as a result of the hydraulic surge of fuel incident to projectile impacts. The conclusions reached are that (1) the forward and after self-sealing fuel cells satisfactorily conformed to the requirements of Spec. MIL-T-5578A; (2) the forward fuselage section conformed to the requirements of Spec. SR-112A; (3) the structural integrity of the forward bulkhead of the after fuselage installation was unsatisfactory; and (4) the Swedlow Plastics Co. support panels employed in the installation performed satisfactorily.

AD-48 377 Accession No.

Syracuse U. Research Inst., N. Y.
DEVELOPMENT OF ROTARY AND TRANSLATORY SEALS. Final rept. 1 July 52 - 31 Aug 54. 89p. incl. illus.
(Contract DA 30-115-ORD-363, Proj. TR5-5059T)
Unclassified report

Injection, mechanical, and stuffing-box seals for aircraft fire-control mechanisms and similar apparatus were studied. The leakage rates of all seals were reduced to very low values, in the neighborhood of 0.1 cu in./year (psi). A modified injection seal (I-30) which uses a felt packing impregnated with lubricating oil had the longest seal life, 3000 hr. The mechanism of leakage was shown to be similar to the viscous flow of gases through a capillary tube. The leakage rate or the volume (at 75°F and 1 atm)/unit time with nitrogen gas was directly proportional to the pressure drop across the seal. A rubber-base, oil-blended compound (FRL-3) was used to obtain the best design of straight (over)

AD-48 377

injection seal and to establish a viscosity-extrusion criterion for seals. The mechanical and stuffing-box seals generally do not give satisfactory reproducible leakage rates.

Uniterms (cont'd)

Supporting
Gas
Flow
Vacuum
Data
Viscosity
Felt
Labyrinth
Knudsens-T-Factor
Fire
Control
Systems

AD-48 399 Accession No.

Rock Island Arsenal Lab., Ill.
OBSERVATION OF OIL LEAKS OCCURRING IN RECOIL MECHANISMS IN STORAGE, by Harry W. Brennan. Technical rept. 9 Nov 54, 13p. incl. illus. tables (Proj. no. TB5-5010B, rept. no. 1; RIA Lab. rept. no. 54-3813) Unclassified report

An inspection was made of the 105-mm howitzer recoil mechanisms stored at Rock Island Arsenal to determine the oil leakage at purge plugs, stuffing boxes, and oil heads. Results indicated that the volume of oil lost is slight and not serious in nature. Leakage rate varied from zero to approximately one reserve (4 oz)/yr.

AD-48 807 Accession No.

Naval Gun Factory, Washington, D. C.
EVALUATION OF OPTICAL SEALS AND OPTICAL SEALING METHODS. CONVENTIONAL TYPE STATIC SEALS (Unclassified title). 18 Oct 54, 7p. illus. (Technical rept. no. NGF-T-71-54; NAVORD rept. no. 4814) Unclassified report

An evaluation was made of 13 static seals and sealing methods for use in optical equipment. Six experimental test fixtures were made in 3 sizes with one each of brass and aluminum. The seals used in these fixtures were O-ring gaskets, flat gaskets of buna N rubber, flat gaskets of neoprene rubber, flat gaskets of silicone rubber, triangular molded gaskets of Hycar rubber, triangular molded gaskets of silicone rubber, triangular molded gaskets of neoprene rubber, and lead and wax. The sealing situations were subjected to a fire hose spray test, a 24-hr test at -20°F, a 24-hr test at 165°F, a 72-hr weathering test, a 150 ft-lb shock test, and a vibration test. Results indicated that the most (over)

AD-48 807

effective seals were the molded triangular gasket type and the O-ring type. Only one failure was attributed directly to the seal. This seal was made of 70 durometer hardness silicone rubber in flat gasket form. The 3 remaining failures were due to the porousness of the brass test fixtures. Of the 9 situations that passed all tests, 4 used flat gasket seals, 2 used molded triangular gasket seals, 2 used O-ring seals, and one used a lead and wax seal. (See also AD-48 808)

AD-48 808 Accession No.

Naval Gun Factory, Washington, D. C.
EVALUATION OF OPTICAL SEALS AND OPTICAL SEALING METHODS. ROTARY TYPE SEALS. 18 Oct 54, 7p. illus. (Technical rept. no. NGF-T-72-54; NAVORD rept. no. 4815) Unclassified report

Fourteen combinations of situations were tested in an effort to find an adequate rotary seal. The word "situation" was defined as a typical test fixture in which a bronze shaft rotates within a cylindrical bronze housing with a chamber for air. Between the shaft and housing is a ring of sealing material which seals by touching both shaft and housing while rotating against one of them. Situation 3, which used 3 Hycar rings sealing on a shaft having a 32-RMS finish, passed all tests satisfactorily. (See also AD-48 807)

AD-49 845 Accession No.

Battelle Memorial Inst., Columbus, Ohio.
B-36 INTEGRAL FUEL- AND OIL-TANK SEALANT RESEARCH AND DEVELOPMENT PROGRAM, by R. G. Heiligmann, P. D. Stickney and others. 15 Oct 54, 48p. incl. illus. tables (Suppl. to Final rept. AD-28 321)
(Subcontract to Consolidated Vultee Aircraft Corp., Contract AF 33(600)5793; Continuation of Subcontract to Consolidated Vultee Aircraft Corp., Contract AF 33(038)2182) Unclassified report

"Work has been directed toward improvement of integral fuel-tank sealants for Consolidated Vultee Aircraft Corporation by Battelle Memorial Institute. Base polymers for sealants have been investigated and new materials have been found which are advantageous as modifiers. Optimum formulations have been established for these sealants. In addition, an organic curing system has been developed which imparts excellent physical properties to Thiokol-based sealants. It has been con- (over)

AD-49 845

cluded that properly compounded sealants based on Thiokol LP-32 offer the greatest benefit in terms of sealant strength and elongation." (Contractor's abstract)

AD-50 824 Accession No.

General Electric Co., Schenectady, N. Y.
PROPELLANT STATIC SEAL DEVELOPMENT IN PROJECT HERMES, by T. C. Jenkins and S. E. Logan. Summary rept. on Proj. Hermes. Nov 54, 41p. incl. illus. (Rept. no. R54A0562)
(Contract DA 30-115-ORD-23, Proj. TU1-2000) Unclassified report

Methods of improving propellant sealing devices were investigated for guided missile use. The 2-in.-diam AN type flared tube joints were replaced by the bolted joints. Pipe threads were eliminated in favor of the straight threaded fitting or a bolted-type joint. Propellant static seals under consideration included initially loaded seals, pressure sensitive seals, and temperature compensating seals. Teflon, Kel-F, and Pb failed by cold flow and contraction when used as seals in liquid O carrying components. Spiral wound gaskets of the stainless steel-Teflon variety were satisfactory in a joint which remained assembled. With repeated disassembly, the hard surfaces of the gasket scored (over)

AD-50 824

or grooved the soft surfaces of the Al flange faces which resulted in leaks. However, these materials were considered undesirable for missile use. For temperature compensating seals, an O-ring (AN Spec 6230) proved to be satisfactory for ambient temperature liquids. The O-rings served as an adequate high-pressure liquid O static seal, but required high flange loading. Various elastomers were tested in contact with liquid O to determine their shock sensitivity to detonation. Of the compounds tested, type W neoprene with a P33 C black filler was the least susceptible to detonation.

AD-50 800 Accession No.

Vitro Corp. of America, West Orange, N. J.
INTEGRAL FUEL TANK LEAK DETECTOR, by E. W. Ruhl. Progress rept. on Phase 1, 8 July-7 Nov 54. 40p. incl. illus. (KLX-1733)
(Contract AF 33(600)27479) Unclassified report

"During this report period, all existing methods of determining the true location of integral fuel tank leakage have been studied and evaluated. Several new techniques were investigated experimentally and leak parameters relating to detector sensitivity have been determined. In addition, recommendations have been prepared on two methods (a Freon ionization detector and a color reaction method) to be developed during the next phase of the contract." (Contractor's abstract)

AD-51 200 Accession No.

Pitman-Dunn Labs., Frankford Arsenal, Philadelphia, Pa.
A DRY VACUUM TEST FOR AIR LEAKS IN CAD CARTRIDGES, by R. Feder and M. Mooney. Oct 54 [19]p. incl. illus. (Proj. no. TS1-15-C44; Memo. rept. no. MR-592) Unclassified report

A method of determining air leaks in cartridges is described which includes the use of a vacuum technique. A cartridge seal may be checked in a vacuum equivalent to an altitude of over 80,000 ft, i.e., the pressure differential can approach the limit of 14.7 psi. Data are tabulated on various types of cartridge cases tested for leakage. A leaky and a non-leaky cartridge can be used to check the apparatus each day.

-[illegible] 775 Accession No.
[illegible]announced report)

Frankford Arsenal, Philadelphia, Pa.
[illegible] FOR THE INITIATOR T4, SEAR ASSEMBLY, by R. Garsdia, Technical rept. 12 July 54, 3p. (Proj. no. TS1-15-C11; Rept. No. S-40d2) Unclassified report

AD-54 206 Accession No.

Kearfott Co., Inc., Little Falls, N. J.
A HERMETICALLY SEALED METAL-TO-METAL MECHANICAL DRIVE, by Y. C. Shen. Sep 53, 16p. incl. illus. (WADC Technical rept. no. 53-391)
(Contract AF 33(038)23311) Unclassified report

AD-55 132 Accession No.

Fairchild Camera and Instrument Corp., Hicksville, N. Y.
HIGH TEMPERATURE, NOBLE METAL FILM TYPE, SEALED VARIABLE RESISTORS, by Lawrence B. Krauss. Quarterly progress rept. 1 Aug - 1 Nov 54. 18p. incl. tables (Rept. no. 8623-1)
(Contract DA 36-039-sc-64500; [Continuation of Contract DA 36-039-sc-5568]) Unclassified report

"The mechanical design of the variable resistor, (Experimental type), was modified to include a hermetic shaft seal of proprietary design. A simulated hermetically sealed panel was devised and tested to be used for the testing phase of the contract in determining the characteristics of hermetically sealed variable resistors. The mechanical parts and materials were ordered and received with the exception of some parts whose deliveries were delayed due to manufacturing difficulties. However, sufficient testing was accomplished to verify the success of the hermetic shaft seal adaptation to the resistor. A resistance control film masking material (over)

AD-55 132

was selected to solve the problem of removing evaporated resistance control [illegible] material from areas of the substrate where it was undesirable, such as between the contact bars at the ends of the control film. The resistance element materials were ordered and received. Test runs were made of the required resistance values. Twenty-six of the forty-five resistance elements were made. All of the test equipment and fixtures of Schedule A of the contract were ordered, received and tested." (Contractor's abstract) (See also AD-23 075)

AD-55 695 Accession No.

Thiokol Chemical Corp., Trenton, N. J.
FUEL CELL SEALANT COMPOUNDS, by Earl H. Sorg, John P. Bowen and others. Rept. on Compounding of Elastomers. Oct 54, 109p. incl. illus. tables (WADC Technical rept. no. 54-10)
(Contract AF 33(038)30523) Unclassified report

"This work was undertaken by the Thiokol Chemical Corporation to develop integral fuel tank sealant compounds with improved properties. The hexamethylene terpolymer sealant compound, prepared and submitted to Wright Field at the end of the first year's work on this contract showed better heat resistance coupled with greatly improved low temperature properties than a comparable IP-2 compound. The 0.5 mole % of TCP and 2 mole % of TCP cross-linked pentamethylene formal copolymers showed better heat and aromatic fuel resistance than the 2 mole % of TCP cross-linked copolymer previously submitted. Compounding studies showed that IP-32 should be more suitable than IP-2 compounds (over)

AD-55 695

since lower moduli and higher ultimate elongation before and after heat aging and after immersion in aromatic fuel were attained. Five two-package mix experimental translucent sealant compounds prepared from the liquid polysulfide/epoxy resin system and submitted to Wright Field were tough and resistant to jet fuel but did not pass all of the Air Force specifications. However, the compounds displayed promising potentialities. The IP-2 base experimental translucent sealant compound submitted to Wright Field for evaluation showed good potentialities as an improved sealant and was more transparent during and after application and cure than an IP-2 base commercial sealant compound." (Contractor's abstract)

AD- 56 036 Accession No.

New York U., Coll. of Engineering, N. Y.
SEALING TECHNIQUES FOR WAVEGUIDE AND COAXIAL ROTARY JOINTS, by Michael D. Maccarone. Quarterly progress rept. no. 1, 1 Aug - 30 Nov 54. 28p. illus.
(Contract DA 36-039-sc-64476) Unclassified report

"A general approach to the development investigation will be along the following lines. The design of the magnetic seal basically will consist of a chamber containing a magnetic slurry in the presence of a magnetic field. The design of the magnetic chamber and the type of magnetic slurry that will be used will be determined empirically through experimental tests that will be conducted in the initial part of the investigation. Once the various parameters can be established and individualized as extensively as is needed a basic magnetic seal design will have been determined. This seal will necessarily have physical and mechanical problems which will have to be eliminated. The refinement of the basic ex-
(over)

AD-56 036

perimental seal will establish the necessary design information which can later be applied in a practical method to the design of magnetically sealed rotary joints for the R. G. 48/U and R. G. 51/U waveguides." (Contractor's abstract) (See also AD-25 809)

AD- 56 708 Accession No.

Naval Engineering Experiment Station, Annapolis, Md.
EFFECT OF LUBRICATING OIL ON ELECTRICAL RESISTANCE OF GASKETED JOINTS, by F. C. Sernatinger, Jr. Research and development rept. 21 Feb 55, 3p. incl. tables (EES rept. no. 070233) Unclassified report

"A series of test runs was made to determine the effect of various types of lubricating oils in lowering the electrical resistance between gasket-insulated simulated sections of diesel engine crankcases used in mine sweepers. It was concluded that the use of new, used, or contaminated lubricating oils should have no serious effect on the electrical resistance between the sections." (NEES abstract)

AD- 58 142 Accession No.

Power Plant Lab., Wright Air Development Center, Wright-Patterson Air Force Base, Ohio.
FINAL QUALIFICATION TESTS CONDUCTED ON UNITED STATES RUBBER CO. SELF-SEALING OIL CELLS CONSTRUCTION US 221-1 FOR THE C-119F AIRPLANE, by J. Weil. 11 Jan 54, 19p. incl. illus. (Technical note no. WCLP-54-1) Unclassified report

Minimum processing - no abstract

AD- 58 978 Accession No.

Royal Aircraft Establishment (Gt. Brit.).
AN EXPERIMENTAL APPARATUS FOR THE MEASUREMENT OF AIR LEAKAGE RATES OVER A WIDE TEMPERATURE RANGE, by M. R. Mead. Nov 54, 7p. illus (Technical note no. RAD 585; [CRB ref. no. 55/110; Encl. 10 to Air Attache, London, rept. no. IR-288-55]) Unclassified report

Minimum processing - no abstract

AD- 59 125 Accession No.

National Gas Turbine Establishment (Gt. Brit.).
THE REGENERATIVE HEAT EXCHANGER FOR GAS TURBINE POWER PLANT. PART VI. MECHANICAL AND SEALING TESTS ON A DISC TYPE ROTARY REGENERATOR, by M. Cox and R. K. P. Stevens. June 54, 18p. illus. (Rept. no. R-159)
UNCLASSIFIED report

Minimum processing - no abstract

CLASSIFICATION
CHANGED TO UNCLASSIFIED
AUTH. CITED IN ASTIA R/B # 63A
DATED [illegible]
[illegible] 30 AUG 1961
(Signature & Date)

AD-60 141 Accession No.

ASTIA Reference Center, Library of Congress, Washington, D. C.
SEALS AND GASKETS. A SELECTED REPORT BIBLIOGRAPHY. Feb 55, 47p. 382 repts. (Rept. no. ARC 1839) Unclassified report

Minimum processing - no abstract

AD- 61 987 Accession No.

Armour Research Foundation, Chicago, Ill.
BONDING MATERIALS: METALLIC MATING SURFACES: LOW RF IMPEDANCE, by V. Pulsifer. Final rept. Nov 54, 75p. incl. illus. tables.
(Contract AF 33(038)23583) Unclassified report

Minimum processing - no abstract

AD- 63 064 Accession No.
(Unannounced report)

Ryan Aeronautical Co., San Diego, Calif.
INTEGRAL FUEL TANK SEALING SPECIFICATION FOR THE RYAN MODEL 61 AERIAL DRONE. USAF MODEL Q-2, by D. W. Floersch. 30 Apr 54, rev. 27 Jan 55, 12p. incl. illus. (Rept. no. 6129-8)
(Contract AF 33(600)27100) Unclassified report

Minimum processing - no abstract

AD-63 258 Accession No.

Material Lab., New York Naval Shipyard, Brooklyn.
QUALIFICATION TEST OF OIL RESISTANT SYNTHETIC RUBBER GASKET MATERIAL - SUBMITTED BY QUAKER RUBBER CORP. Final technical rept. 21 Apr 55, 4p. tables (Lab. proj. no. 5542-20) Unclassified report

Minimum processing - no abstract

AD- 63 260 Accession No.

Material Lab., New York Naval Shipyard, Brooklyn.
QUALIFICATION TEST OF SYNTHETIC RUBBER GASKET MATERIAL- SUBMITTED BY THE DAVIDSON RUBBER COMPANY. Final technical rept. 6 Apr 55, 4p. tables (Lab. proj. no. 5542-9) Unclassified report

Minimum processing - no abstract

AD-64 334 Accession No.
(Unannounced report)

Material Lab., New York Naval Shipyard, Brooklyn.
QUALIFICATION TEST OF SYNTHETIC RUBBER GASKET MATERIAL - SUBMITTED BY HEWITT-ROBINS, INC. Final technical rept. 13 Apr 55, 4p. tables (Lab. proj. 5542-11) Unclassified report

Minimum processing - no abstract

AD-64 335 Accession No.
(Unannounced report)

Material Lab., New York Naval Shipyard, Brooklyn.
QUALIFICATION TEST OF OIL RESISTANT SYNTHETIC RUBBER GASKET MATERIAL - SUBMITTED BY B. F. GOODRICH COMPANY. Final technical rept. 28 Apr 55, 3p. tables (Lab. proj no. 5542-18) Unclassified report

Minimum processing - no abstract

AD-65 209 Accession No.
(Unannounced report)

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
EVALUATION OF GOODYEAR TIRE AND RUBBER COMPANY RECIPES AND DEVELOPMENT OF OTHER RECIPES FOR SPECIFICATION MIL-R-2765A, by J. W. Hollister. 12 May 55, 7p. tables (Rept. no. 7-34) Unclassified report

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AD-67 223 Accession No.

Armour Research Foundation, Chicago, Ill.
EVALUATION OF HERMETIC SEALING TERMINALS, by George A. Forster. Phase rept. no. 1, 15 July 54-25 Apr 55 on Improved Electronic Components. May 55, 25p. incl. illus. tables, 15 refs. (Contract AF 33(600)27674) Unclassified report

"The first phase of this project was to determine all necessary requirements of hermetic sealing terminals for use in airborne electronic equipment and to devise tests for them. The sources of information were manufacturers and users of hermetic terminals, government agencies particularly the Air Force, and technical literature including government standards and specifications. The information was analyzed and arranged into a set of requirements with tests covering each electrical and environmental condition. The important electrical characteristics are dielectric strength, corona voltage, current capacity, and insulation resistance. The environmental conditions include temperature (over)

AD-67 223

aging, temperature cycling, temperature shock, vibration, physical shock, terminal strength, salt spray and gas leakage. Limiting requirements have been set up within a system of grades for the environmental conditions and insulation resistance. A series of simulated environmental tests were designed covering each condition. The second phase of the project will be the testing of all typical types and manufacture of hermetic terminals to determine if they meet the requirements or in what respects they fail." (Contractor's abstract)

AD-67 333 Accession No.

Samuel Feltman Ammunition Labs., Picatinny Arsenal, Dover, N. J.
THERMOPLASTIC FILM AS A SEALER FOR WATERPROOF CASE LINERS, by Richard J. Chabot. July 55, 17p. incl. illus. tables (Proj. no. TB4-672; Technical rept no. 2215) Unclassified report

"Summary: Thermoplastic film-sealed case liners and conventional adhesive-sealed liners, were tested for effects of rough handling, aging, water penetration, and water-vapor penetration. In addition, both types of sealed seams were tested for compliance with the seam strength requirements of Specification MIL-A-140A 'Adhesive, Water Resistant, Waterproof Barrier Material'. Conclusions: (a) Thermoplastic-sealed case liners are approximately equal in performance to adhesive-sealed liners. (b) In the laboratory tests, case liners sealed with thermoplastic film required less fabrication time than liners sealed with adhesives. In actual assembly plant procedure, however, fabrication (over)

AD-67 333

time would be about the same for both methods (See paragraph 10). (c) After heat aging, thermoplastic sealed seams are approximately as strong as adhesive sealed seams. (d) After water immersion, thermoplastic sealed seams are stronger than adhesive sealed seams (e) The adhesives and thermoplastic films used in these evaluations are approximately equal in cost, according to current market prices. (f) Thermoplastic film seals waterproof canvas and various cloth materials effectively. This information may be useful in specific packaging applications." (SFAL summary and conclusions).

AD-67 362 Accession No.

Aircraft Lab., Wright Air Development Center, Wright-Patterson Air Force Base, Ohio.
A METHOD FOR MORE ACCURATE DESIGN ESTIMATION OF AIR LEAKAGE FROM CABINS AND COCKPITS OF PRESSURIZED AIRCRAFT, by A. B. Nutt. Rept. on Construction Techniques and Application of New Materials and Canopy Door and Window Seal Development. Sep 54, 23p. incl. illus. tables, 25 refs. (WADC Technical rept. no. 54-446) Unclassified report

"A review of current methods of determining air leakage from pressurized aircraft cockpits and cabins is made. An empirical method for more accurately calculating and predicting such leakage is presented. A bibliography is incorporated as a source of more detailed information on current leakage amounts and methods of determination." (AL abstract)

AD-67 600 Accession No.

Samuel Feltman Ammunition Labs., Picatinny Arsenal, Dover, N. J.
LOW TEMPERATURE SEAM STRENGTH OF BARRIER MATERIALS, by Richard H. DeVore. July 55, 15p. incl. illus. tables (Proj. no. TB4-672; Technical rept. no. 2211) Unclassified report

Investigation was made to determine: (1) the practicability of incorporating requirements for seam tests of low temperatures in military specifications for barrier materials; (2) the low temperate extremes that can be satisfactorily tolerated by seams made from currently available commercially manufactured barrier materials; and (3) the effects of low temperatures (as extreme as -65°F) on the seam strength characteristics of barrier materials. Testing at low temperatures was accomplished by attaching static loads to the test specimens and suspending them so that the load was applied directly on the seams fabricated from the test materials. Some of the specimens were cut from the materials as received; others were aged, using the accelerated (over)

AD-67 600

aging process of Specification MIL-B-131A. All specimens were cut so as to produce a seam 1 in. wide and 3 in. long. Cylinders of 2 weights, 10 oz and 2 lb, were suspended from the seam specimens by means of clamps during the low-temperature tests. Most of the types of barrier materials currently used for military packaging can be heat sealed so as to meet the low-temperature test requirements. Saran-faced Kraft paper was, however, satisfactory at -65°F. Sealing conditions which are satisfactory for unaged materials are not always adequate for the same materials when they have been subjected to accelerated aging. Results show that polyethylene heat seal-faced materials which seal satisfactorily when unaged (using a given temperature, dwell, and pressure) often require a higher temperature, longer dwell, or different pressure to seal satisfactorily after being subjected to accelerated aging. Heat sealing conditions for vinyl heat seal-faced sheets were more constant.

AD-68 405 Accession No.

Naval Ordnance Test Station, Inyokern, China Lake, Calif.
LABORATORY TESTS OF HIGH-SPEED AND HIGH-PRESSURE ROTARY O-RING SHAFT SEALS, by L. E. Van Hise and F. H. Waterhouse, Jr. 21 Mar 55, [14]p. incl. illus. tables (NOTS 1067; NAVORD rept. no. 3464) Unclassified report

"A test fixture was designed and built for laboratory evaluation of an O-ring seal for a rotating shaft operating at 500 psi and at 2,000 rpm and above. A housing incorporating a spring-type retaining ring ('Spirolox,' Ramsey Corporation, St. Louis, Mo.) to position a 1/8-inch O-ring (Buna N) and a hard chrome-plated shaft (4 1/2 to 5 1/4 microfinish) proved successful in operating under the prescribed conditions." (NOTS abstract)

AD-68 873 Accession No.

Naval Gun Factory, Washington, D. C.
DEVELOPMENT OF AIR LEAKAGE TESTS FOR AMMUNITION BOXES, CONTAINERS AND TANKS. 27 June 55 [10]p. incl. illus. tables (Rept. no. NGF-T-31-55; NAVORD rept. no. 5083) Unclassified report

The following containers (in empty and in loaded condition) were submerged in water 90° warmer than the ambient temperature: (1) MK 91, Mod 0, fuze container; (2) MK 3, Mod 3, 20-mm ammunition box; (3) MK 2, Mod 0, ammunition component box; (4) MK 6, Mod 1, 5-in/38 cartridge tank; (5) MK 5, Mod 1, 3-in/50 cartridge tank; (6) MK 1, Mod 0, small arms ammunition box; and (7) MK 4, Mod 0, 16-in powder tank. Average size empty containers (approximately 100 to 2000 cu in) were easily tested in hot water; small, very large, and loaded containers were difficult to test for air leakage by submersion in hot water. A partial vacuum around the submerged container was recommended especially for testing small and medium (over)

AD-68 873

size containers. The incorporation of air valves was suggested in pressure testing large containers.

AD-70 011 Accession No.

Douglas Aircraft Co., Inc., Santa Monica, Calif. EVALUATION OF HIGH TEMPERATURE HYDRAULIC SEALS, by Edward A. Webster. Rept. on Hydraulic Seal Development. Jan 55, 96p. incl. illus. tables, 38 refs. (Rept. no. DEV-1792; WADC Technical rept. no. 55-120)
(Contract AF 33(616)236) Unclassified report

An investigation was conducted in order to develop an interim seal suitable for aircraft and missile systems utilizing 3000-psi MIL-O-5606 hydraulic fluid from -65° to +300°F. This investigation included determination of temperature limitations of approved AN6227 O-rings and AN6246 leather back-up rings, comparative qualification tests of all currently available experimental high-temperature O-ring materials, evaluations of various vendor and contractor-furnished experimental seal configurations designed for both standard and nonstandard packing glands, and tests of back-up materials and configurations suitable for high-tempera- (over)

AD-70 011

ture use. Testing results listed the upper temperature limit of AN6227 packings at approximately 200°F. Specific compounds within this group have widely divergent heat resistances, but none should be used for extended periods of time above 250°F. Of the O-ring compounds tested, only one (compound F-1) had satisfactory -65°F sealing characteristics. None were comparable at 300°F to AN6227 compounds at 120°F. None of the O-ring compounds may be used for extended periods of time at 300°F. Spiral Teflon back-ups and metallic O-rings may be safely used with certain reservations at 300°F. Leather back-up rings exhibited marginal qualities at 200°F. Currently approved AN6230 and AN6290 static seals and gaskets are not satisfactory for use at 300°F.

AD-72 231 Accession No.
(Unannounced report)

Material Lab., New York Naval Shipyard, Brooklyn. QUALIFICATION TEST OF SYNTHETIC RUBBER GASKET MATERIAL, SUBMITTED BY THE MARTIN RUBBER COMPANY, LONG BRANCH, NEW JERSEY. Final technical rept. 6 July 55 [7]p. incl. tables (Lab. proj. no. 5542-12) Unclassified report

AD-72 233 Accession No.
(Unannounced report)

Material Lab., New York Naval Shipyard, Brooklyn. QUALIFICATION TEST OF SYNTHETIC RUBBER GASKET MATERIAL, SUBMITTED BY CONTINENTAL RUBBER WORKS, ERIE, PENNSYLVANIA. Final technical rept. 6 July 55 [7]p. incl. tables (Lab. proj. no. 5542-8) Unclassified report

AD-72 287 Accession No.
(Unannounced report)

Material Lab., New York Naval Shipyard, Brooklyn. QUALIFICATION TEST OF OIL RESISTANT SYNTHETIC RUBBER GASKET MATERIAL, SUBMITTED BY THE GARLOCK PACKING COMPANY, PALMYRA, NEW YORK. Final technical rept. 17 May 55 [5]p. incl. table (Lab. Proj. no. 5542-17) Unclassified report

AD-72 288 Accession No.
(Unannounced report)

Material Lab., New York Naval Shipyard, Brooklyn. QUALIFICATION TEST OF SYNTHETIC RUBBER GASKET MATERIAL, SUBMITTED BY UNITED STATES RUBBER COMPANY, PASSAIC, NEW JERSEY. Final technical rept. 14 July 55 [6]p. incl. table (Lab. proj. no. 5542-15) Unclassified report

AD-72 260 Accession No.
(Unannounced report)

Material Lab., New York Naval Shipyard, Brooklyn. QUALIFICATION TEST OF SYNTHETIC RUBBER GASKET MATERIAL, SUBMITTED BY QUAKER RUBBER CORPORATION, PHILADELPHIA, PENNSYLVANIA. Final technical rept. 7 July 55. [6]p. incl. table (Lab. proj. no. 5542-13) Unclassified report

AD-72 503 Accession No.

Midwest Research Inst., Kansas City, Mo. TESTING OF SEAL DESIGNS AND MATERIALS, by Carl C. Perkins, Jr. Final rept. 25 Feb 55, 39p. incl. illus. tables (Rept. no. AEDC TR-55-10)
(Contract AF 18(600)1122) Unclassified report

In an attempt to develop seals for the flow converter of a propulsion wind tunnel, 3 silicone rubber materials were tested in several seal configurations. Test fixtures were designed to provide full-scale duplication of each type of seal junction found in the flow converter. Data were obtained on the leakage rates and the effects of abrasion at temperatures up to 650°F. The test results indicated that Dow Corning Silastic no. 80 was the best suited material tested and that seal configuration no. 4 (a seal lug with additional seals in its side) was the most flexible and the most easily replaced design. At 650°F, the service life of the seal can be expected to be about 1 to 3 hr; at 550°F, the service life should be increased substantially, and at 500°F, the seals should (over)

AD-72 503

operate indefinitely. Seals joined at 90° angles to each other were bonded at the 45° intersection with satisfactory results.

AD-73 961 Accession No.

Midwest Research Inst., Kansas City, Mo. TESTING OF SEAL DESIGNS AND MATERIALS, by Carl C. Perkins, Jr. 31 May 55, 19p. incl. illus. (Suppl. to Final technical rept. [Rept. no. AEDC-TR-55-18] AD-72 503)
(Contract AF 18(600)1122) Unclassified report

Seals made from laminated strips of phosphorous bronze were tested against an Ni-plated plate, a stainless steel plate, and a plate coated with a heat-resistant paint. The leakage rate for this type of seal and the leakage rate of seal junctions and seal corners was established, and the abrasion effect of the seal on the various plates was determined. The results indicated that the spring-loaded metallic seal was feasible and desirable from the standpoint of maintenance in the flow converter of the propulsion wind tunnel. The abrasion test results indicated that at 650°F, the Ni-plated material provided the best results in preventing scoring. The painted surface was unsatisfactory, and the stainless steel galled during a 4-hr test at 650°F.

AD-74 089 Accession No.

Aircraft Lab., Wright Air Development Center, Wright-Patterson Air Force Base, Ohio. PERFORMANCE CYCLING TESTS OF HIGH TEMPERATURE HYDRAULIC O-RING PACKINGS, by Frank R. Straus. Jan 55, 21p. incl. illus. tables (Technical note no. WCLS-54-71) Unclassified report

The methods used to determine the mechanical performance of synthetic rubber O-ring packings in hydraulic fluids at temperatures up to +550°F are described. A total of 40,000 cycles were conducted at 300°F in MLO 8200 fluid on a commercial neoprene WHT compound with spiral Teflon back-up rings. No leakage occurred at -40°F for a 20-hr period at 10 psi. At 3000 psi and -40°F, the leakage was 25 cc in 1 hr. Subjecting a standard AN6227 O-ring compound to 50,000 cycles at room temperature in MIL-O-5606 fluid resulted in a 38-cc leakage from the right-hand seal and 5 cc from the left-hand seal. An acrylate compound used with spiral Teflon back-up rings was subjected to 5000 cycles (over)

AD-74 089

at room temperature and 6200 cycles between 350° to 425°F; the test was stopped when fluid poured from the right-hand seal. Test results indicated that spiral Teflon back-up rings can be used for at least 2 hr up to 550°F at 3000 psi without extruding. Teflon back-up rings are desirable on both sides of the O ring regardless of the pressure. The use of brass washers to maintain close clearances was satisfactory, although some failures could be attributed to high rate of wear on the washer. Compound 453-25C functioned satisfactorily in a -65° to +400°F range. Compound 453-9A functioned satisfactorily at 500°F for more than 5 hr. The lives of compounds 453-9A and 1203-70 at 550°F were about 1.5 hr. The removal of the initial stretch or tension on an O-ring compound resulted in longer life of the packing.

AD-74 322 Accession No.

Aircraft Equipment Testing Co., Baltimore, Md.
TESTING OF METAL BOSS SEALS, by Harry P. Kupiec. Rept. on Aircraft Hydraulic Systems. Apr 55, 61p. incl. illus. tables (WADC Technical rept. no. 55-163)
(Contract AF 33(600)26548) Unclassified report

"The metal boss seal was conceived by Wright Air Development Center to meet the requirements of hydraulic and pneumatic systems with operating pressures up to 5,000 psi, and temperatures as low as -100°F. and as high as 600°F. Basic development on the seal was accomplished by Wright Air Development Center. Further development work and testing of the seal were performed by the Aircraft Equipment Testing Company. The application of the metal boss seal involves the use of deformable metal ring in conjunction with standard AN hydraulic fittings with AND10056 or AND10057 fitting ends in standard AND10050 bosses. It is concluded that the metal boss seal possesses the desirable characteristics for a boss seal as indicated

AD-74 322

by tests conducted on size -5, -8, and -12. The metal boss seal is considered relatively simple and reliable. With proper choice of material, it is possible that this seal design may be suitable for operating temperatures above 600°F. This seal is being considered as a replacement for the current standard AN 6290 synthetic rubber gasket." (Contractor's abstract)

AD-75 568 Accession No.

General Hermetic Sealing Corp., Valley Stream, N. Y.
REUSABLE SEALS FOR ELECTRONIC EQUIPMENT, by William Brown and A. Razdow. Quarterly rept. no. 1, 1 June-31 Aug 55. 37p. incl. illus.
(Contract AF 33(616)2994) . Unclassified report

Progress is reported on the investigation of reusable seals and the preparation of necessary test procedures and equipment. Preliminary tests disclosed that brass as a material for the enclosures gives the best initial results. QQ-S-633 steel was found to be practical if the proper plating is applied. Heat-transfer analysis showed that nitrogen should be used as the filler gas; a small percentage of helium must be added for leak-rate testing. Studies of solders and fluxes showed that the temperature at which the solder softens must be held over 150°C. To study the heat distribution over a greater period of time and the effect of the heat on the components in the enclosure, the temperature-measure-
(over)

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ment cycle had to be increased to more than 4 minutes. Preliminary tests disclosed that the proper use of known aluminum soldering and plating methods was necessary. With correct plating, efficient leak-proof, reusable enclosures can be manufactured.

AD-77 374. Accession No.

New York U. Coll. of Engineering, N. Y.
SEALING TECHNIQUES FOR WAVEGUIDE AND COAXIAL ROTARY JOINTS, by Michael B. Maccarone. Quarterly progress rept. no. 4, 1 June-31 Aug 55. 21p. incl. illus.
(Contract DA 36-039-sc-64476) Unclassified report

Evaluation was made of an experimental mechanical rotary seal. The rotary seal unit consists of a stationary upper component which houses 2 new departure ball bearings and a lower portion which houses the rotary seal components. The stationary component contains a C-graphite sealing ring (mating sealing surface) which is mounted in a steel housing. The rotating component consists of a small cylindrical permanent magnet of Alnico V and two O rings. One component is attached to the internal rotating member of the joint and the other is coupled to the brass housing by two O rings. A steel shaft at the top portion drives the internal member of the rotary joint and the rotating component of the seal.
(over)

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An aluminum disk attached to the shaft covers the ball bearings. In the side of the lower housing is the secondary pressure inlet and in the bottom of the housing is the primary or internal pressure inlet. The rotary seal unit utilizes applied pressure in the secondary chamber to create the force necessary to keep the sealing surfaces in contact; while the magnet keeps the sealing surfaces in contact when the unit is running unpressurized. A maximum torque of 15 in-lb was obtained with the unit running at 2200 rpm and at 90 to 100 psi (a development of about 1/2 hp). A large portion of this horsepower was developed at the sealing surfaces. The unit performance was considered satisfactory except for the heat developed at the sealing surfaces. A small amount of leakage was attributed to seal construction. The C-graphite sealing ring showed little wear after 5 hr of testing. (See also AD-69 748)

AD-78 224 Accession No.

Bjorksten Research Labs., Inc., Madison, Wis.
INVESTIGATION OF SEALANTS FOR AIRCRAFT INTEGRAL FUEL TANKS, by R. P. Lappala, S. E. Rolowetz, and W. Stabenau. Final rept. 15 Jan 54-9 Aug 55. 79p. incl. illus. tables.
(Contract AF 33(600)26380) . Unclassified report

"Solvent-based sealant systems were formulated and screened to select sealants suitable for fill and drain application to aircraft integral wing tanks and for protection of adhesive bonds in fuel tanks. The screening tests simulated actual service conditions, including prolonged exposure to fuel vapor (Hydrocarbon Type III, MIL-H-3136). Polyurethane Y-573 (Bjorksten Research Laboratories) was selected as the best sealant on the basis of its excellent fuel and water resistance, adhesion to Alclad aluminum, and protection of Metlbond 402 adhesive bonds. This sealant was prepared from meta-tolylene diisocyanate and polyethylene glycol 200 and catalyzed with Triol 230. The second choice was FM-47 adhesive (Bloomingdale Rubber Co.), which,
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AD-78 224

however, was more sensitive to water contamination in the fuel. An attempt was made to seal prototype tanks with the Y-573 and FM-47 sealants by the fill and drain method. Uniform coatings were not produced due to difficulty in obtaining proper drainage of the tanks, and minor leaks resulted. In view of the superiority of these sealants in other respects, however, they merit further investigation of application techniques. Two-coat applications of the Y-573 and FM-47 sealants are recommended, with separate curing operations for each coat. Solvent degreasing is the only cleaning procedure required before application of the sealants."
(Contractor's abstract)

AD-78 234 Accession No.
(Unannounced report)

Midwest Research Inst., Kansas City, Mo.
DESIGN AND TESTING OF THE CONVERGENCE BLOCK AND FLEXIBLE PLATE SEALS OF FLOW CONVERTER "B". Final rept. 15 June 54, 61p. incl. illus. tables [Rept. no. AEDC TR-54-38]
(Subcontract to Sverdrup and Parcel, Inc., [Contract AF 33(038)9928]) Unclassified report

AD-78 688 Accession No.
(Unannounced report)

Frankford Arsenal, Philadelphia, Pa.
SEALING OF FIRE CONTROL INSTRUMENTS. DEVELOPMENT OF A SEAL FOR RETICLE ILLUMINATION WINDOWS, by John P. Yannello. 28 Oct 55, 6p. incl. illus. table (Proj. no. TR5-5059T; Technical memo. rept. no. M55-6-1) Unclassified report

AD-78 750 Accession No.
(Unannounced report)

Naval Engineering Experiment Station, Annapolis, Md.
SEALING OF SUPERHEATER TUBE SEATS BY HIGH TEMPERATURE BRAZING, by W. E. Clautice. 5 Mar 54, 4p. illus. (EES rept. no. 060080A)
Unclassified report

"High temperature brazing of superheater tube seats was investigated, using the high frequency induction and electric resistance heating methods. The development of special gas burners for operation inside of the superheater headers was also attempted. The results of these tests indicate that high temperature brazing of tube seats in air by the methods investigated and the materials used in the present state of development, is not applicable for sealing superheater tube seats."
(NEES abstract)

AD-80 814 Accession No.

North American Aviation, Inc., Downey, Calif.
HIGH TEMPERATURE INFLATABLE GASKETS, FABRICATION PROCEDURE, by F. J. French and F. S. Spanninger. 28 July 55, 13p. incl. illus. (Rept. no. AL-2170) Unclassified report

A discussion is presented on the development of a DC-152 silicone rubber inflatable gasket for short time service at 500°F. Procedures described for fabrication of the gasket include extrusion, autoclave cure, air inlet stem and splicing installation, and step and final cures. A cap is applied to the air inlet stem area for protection against rough handling and damage from cutting.

AD-80 889 Accession No.

Industrial Lab., Mare Island Naval Shipyard, Vallejo, Calif.
DEVELOPMENT OF ELASTOMERIC COMPOUNDS FOR USE WITH NON-FLAMMABLE HYDRAULIC FLUIDS, by Arthur W. Scott. 20 July 55 [28]p. incl. illus. tables (Development rept. no. 5238-54)
Unclassified report

AD-82 044 Accession No.
(Unannounced report)

Naval Gun Factory, Washington, D. C.
EVALUATION OF OPTICAL SEALS AND OPTICAL SEALING METHODS. CONVENTIONAL TYPE STATIC SEALS. 7 Dec 55 [5]p. incl. table (Rept. no. NGF-T-68-55; NAVORD rept. no. 5114) Unclassified report

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AD-82 045 Accession No.
(Unannounced report)

Naval Gun Factory, Washington, D. C.
EVALUATION OF OPTICAL SEALS AND OPTICAL SEALING METHODS ROTARY TYPE SEALS, 19 Dec 55 [10]p. incl. illus. (Rept. no. NGF-T-69-55; NAVORD rept. no. 5116) Unclassified report

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AD-82 306 Accession No.
(Unannounced report)

American Designers, Inc.
THE DESIGN OF O-RING SEALS FOR ROCKETS. 17 Mar 52, 22p. incl. illus. tables (NOTS Technical memo. no. 1046) (In cooperation with Naval Ordnance Test Station)
(Contract N123s-82333) Unclassified report

AD-82 914 Accession No.

Ontario Research Foundation (Canada).
INVESTIGATION INTO THE SEALING OF QUARTZ CRYSTAL HOLDERS TYPE HC-6/U, by B. W. Schumacher. Quarterly rept. no. 2 for Apr 55. 31 May 55, 31p. incl. illus. tables (Serial no. 2-P-4-998)
Unclassified report

Solder sealing is being investigated for hermetically sealing quartz crystal units in the crystal holder type HC-6/U. Studies will be made of (1) the effects of external conditions such as temperature, humidity, and salt spray on the sealing; (2) refinements in the technique of solder sealing; and (3) a reliable method of production testing of the crystal seals. A survey was made of available literature on quartz crystals. Efforts were concentrated on the development of a leak testing method for solder seals. A pressure-rise method which utilizes a carrousel-type vacuum lock device was used to check leaks in the mass-spectrometer range. A total of 87 units were tested for leaks from a batch of
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AD-82 914

103 HC-6/U units. Only 37 showed no leak. The units were tested for their electrical behavior in the temperature cycling test, especially for dipping failures. Of the 50 units with leaks, 35 were also dippers. Of the 37 units which were hermetically sealed, 19 dipped.

AD-82 915 Accession No.

Ontario Research Foundation (Canada).
INVESTIGATION INTO THE SEALING OF QUARTZ CRYSTAL HOLDERS TYPE HC-6/U, by B. W. Schumacher. Quarterly rept. no. 3 for period ending July 55. [9]p. incl. tables (Serial no. 2-P-4-998)
Unclassified report

Efforts were concentrated on the development of a leak testing method for solder seals. Calculations showed that leaks which caused dipping failures are detectable by a pressure-rise method if each unit is put into a vacuum chamber which is evacuated to a final pressure of about 10^{-6} mm Hg. Of 5300 crystal units in HC-6/U holders, 364 were tested to obtain statistical information concerning the quality of the seals of a larger number of crystal units which had been in stock for several years. About 63% of the oldest units in stock were leakers. The ratio of bubblers to small leakers was about 1:1. (See also AD-82 914)

AD-82 916 Accession No.

Ontario Research Foundation (Canada).
INVESTIGATION INTO THE SEALING OF QUARTZ CRYSTAL HOLDERS TYPE HC-6/U, by B. W. Schumacher. Quarterly rept. no. 1. Jan 55. [24]p. incl. illus. tables.
(Contract P 69-10-116) Unclassified report

Considerations are presented concerning the detection of small leaks in hermetic seals of small containers. The purpose of hermetically sealed containers for electric circuit components is to protect the sealed-in components from contact with deleterious gases or vapors. Hg arc rectifiers must be protected against the entrance of any gas, Ag contacts of a switch from corrosive gases such as SO_2, and transformers and piezoelectric quartz crystals from the entrance of moisture. The basic requirement in classifying leaks and sealed containers for a hermetic seal is given by the circuit element to be protected. Classification of sealed containers is outlined, and a discussion of leak
(over)

AD-82 916

detection methods relevant to small-sealed containers is presented. (See also AD-82 914)

AD-83 653 Accession No.
(Unannounced report)

Aberdeen Proving Ground, Md.
TEST OF OLDHAM COUPLING DUST SEAL, by T. R. McConnell. Partial rept. no. 3 (Final) on Production Engineering Test for Tank, 90mm Gun, M48. Oct 55, 8p. incl. illus. (Proj. no. 1T2-760, Rept. no. 4)
Unclassified report

AD-83 716 Accession No.

Naval Ordnance Test Station, Inyokern, China Lake, Calif.
INJECTED METAL SEALS, by Guinn E. Metzger and Robert A. Weinhardt. 28 Oct 55, 18p. incl. illus. tables (NOTS 1254; NAVORD rept. no. 4944)
Unclassified report

"A series of tests on the injection of molten metal into a groove between two mating parts to determine the relationship of injection pressure and temperature to groove fill was performed at the U. S. Naval Ordnance Test Station. The effect of injection pressure and temperature on the extent of metal flash beyond the groove confines was also established. This relationship was established with tubular specimens varying in diameter from 1-inch to 5-inches, with nominal joint clearances of 0.5 to 4.5 mils, and with two injected metals (Cerrobase and Cerromatrix). Injection pressures varied from 80 to 1600 psi. Injection temperatures varied from 280 to 480°F for Cerrobase and from 540 to 700°F for
(over)

AD-83 716

Cerromatrix. The production of joints with injected Cerrobase or Cerromatrix appears to be feasible for the range of sizes and clearances and the joint design investigated. The evaluation of the mechanical strength and sealing effectiveness of such joints will be done at a later date." (NOTS abstract)

AD-85 114 Accession No.
(Unannounced report)

National Advisory Committee for Aeronautics, Washington, D. C.
WEAR OF TYPICAL CARBON-BASE SLIDING SEAL MATERIALS AT TEMPERATURES TO 700°F, by Robert L. Johnson, Max A. Swikert, and John M. Bailey. Feb 56, 22p. incl. illus. table (Technical note TN 3595) Unclassified report

"Wear and friction studies were made with carbon-type seal materials sliding against tool steel, stainless steel, and chromium-plated steel at temperatures to 700°F and a sliding velocity of 10,000 feet per minute. Wear was several times greater at 500°F and above than with no heat added for all carbon materials. The mating material had more effect on wear and surface failure than did the use of various impregnants in the carbon. Chromium plate gave less wear than the other mating materials under all conditions of the experiment." (NACA abstract)

AD-85 318 Accession No.

Institution of Mechanical Engineers (Gt. Brit.).
THE FLOW OF AIR THROUGH RADIAL LABYRINTH GLANDS, by W. J. Kearton. 19 Oct 54, 14p. incl. illus. tables. Unclassified report

"A theory for the flow of air through a radial gland has been worked out for both outward and inward flow. Expressions are derived for the pressure distribution in each kind of flow, and it is shown that the critical pressure ratio can be reached only in the final constriction. Experiments were made on a gland having a single ring and also on a 20-ring gland of the staggered type. The discharge coefficients in the latter gland were lower than those in the single-ring gland, possibly due to the different approach conditions. The observed pressure distribution in the multi-ring gland agreed well with the theoretical value. Finally, some experiments were made with unstaggered radial glands under various conditions. Discharge coefficients greater than unity were measured." (IME abstract)

AD-85 568 Accession No.

Aero Medical Lab., Wright Air Development Center, Wright-Patterson Air Force Base, Ohio.
LONG-TERM EMERGENCY PRESSURIZATION WITH NYLON INFLATABLE BULKHEADS, by Elizabeth Comfort. June 55, 5p. incl. illus. table. (WADC Technical note no. 55-296) Unclassified report

"An investigation has been made of the feasibility of using a nylon inflatable material known as Rigidaire to re-establish and maintain a pressure differential of 6 psi in a damaged pressure cabin aircraft. During tests in a low pressure chamber the material was found to be capable of sealing this differential over rectangle orifices 24 x 40 inches and 18 x 24 inches for prolonged periods of time." (AML abstract)

AD-86 205 Accession No.

Horizons, Inc., Cleveland, Ohio.
DEVELOPMENT OF FRICTION-SEAL MATERIALS FOR HIGH TEMPERATURE-HIGH SPEED OPERATION, by F. C. Wagner. Quarterly progress rept. no. 2. 23 Feb 56, 5p. incl. illus. table.
(Contract AF 33(616)3124) Unclassified report

Research was initiated to develop materials which will perform satisfactorily as friction seals under conditions of high temperature and high rubbing speeds. desired operating conditions are: (1) 30,000-fpm surface speed; (2) 1000°F ambient temperature; (3) 200-psi air pressure differential across the seal; (4) 2 to 5 in. of oil pressure across the seal; and (5) maximum internal oil temperature of 500°F. The desired performance of such a seal is: (1) minimum wear life of 1000 hr; (2) maximum oil leakage of 1 cc/min; and (3) maximum air leakage of 1 lb/hr/6-in. diameter. A wear test machine was constructed consisting of a rotating shaft with one component of the potential seal (over)

AD-86 205

attached to the end. Specimen preparation is proceeding; 4 grades of graphite base material (Graphitar) no. 39, 67, 75, and 84) were selected. Fabrication of specimen disks from powdered metals was begun. The first specimens will consist of sintered Ni powder with additives of up to 10% CrB and 5% graphite, with approximately 30% Ag added by infiltration. Other seal materials being considered are WC-Ag and Ni-Ag.

AD-87 221 Accession No.
(Unannounced report)

Kraettler-Weber Tool Co., Newark, N.J.
DRIVE, MECHANICAL, ROTARY SEAL (HIGH SPEED) by A. Randow. Quarterly rept. no. 8, 7 Apr 51-15 Mar 52. [23]p. incl. illus.
(Contract W36-039-sc-44620) Unclassified report

AD-88 545 Accession No.

Varian Associates, Palo Alto, Calif.
VA-86 KLYSTRON AMPLIFIER ENGINEERING SERVICES PROGRAM, by Robert C. Schmidt. Final rept., 1 June 54-30 Sep 55, 24p. incl. illus. (Engineering rept. no. 155-15; [RADC TR-56-28])
(Contract AF 19(604)1190) Unclassified report

Research was conducted on high-power output windows, other ceramic-to-metal seals, air-cooling, the design and fabrication of cathodes, the use of various exhaust techniques, permanent magnet fields, and the examination of metallic bellows; these phases were evaluated with respect to the design and development of high-power, 4-cavity, tunable klystron amplifiers. Window development resulted in a design consisting of a circular ceramic window of a diameter almost as large as the wide dimension of the waveguide, which is placed in the guide at an acute angle. An RF input seal was devised which employs a ceramic cylinder with metallizing and brazed joints at the ends and uses some parts (over)

AD-88 545

of a standard type-N cable connector. A complete redesign of the mechanical structure of the cathode HV insulator appeared necessary for the best use of the properties of the ceramic seals. An analysis of air cooling of high-power klystrons vs liquid cooling revealed that the total power requirement of the prime mover of an air-cooling system would be over 3000 w; the requirement for an equivalent water-cooling system would be 1250 w. The metallic bellows for tuning the cavities of high-power klystrons was investigated. A bellows made of a nonmagnetic grade of stainless steel exhibited a minimum life of over 1500 c when cycled; individual tests yielded a life as high as 11,500 c. An exhaust technique was studied which accomplishes tube outgassing by employing a high exhaust bakeout temperature (600° to 650°C) and special handling of the cathode breakdown. (See also AD-86 410)

AD-89 072 Accession No.

Stanford Research Inst., Menlo Park, Calif.
INVESTIGATION OF NONRUBBER MATERIALS FOR O-RING SEALS (Unclassified title) by E. E. Ryskiewicz. Final rept. 21 Sep 55, 17p. incl. illus. table. Confidential report

AD-89 193 Accession No.
(Unannounced report)

Douglas Aircraft Co., Inc., Santa Monica, Calif.
INVESTIGATION OF 10° UNDERCUT GROOVE FOR "O" RING PACKING, by B. H. Vennum. 10 Dec 45, 7p. incl. illus. (Rept. no. SM-20063) Unclassified report

AD-89 198 Accession No.
(Unannounced report)

Douglas Aircraft Co., Inc., Santa Monica, Calif.
PERFORMANCE OF HOUGHTON #2216 BACK-UP RINGS WITH "O" RINGS ON LARGE STRUTS AT 3000 PSI PRESSURE, by E. R. Broadbent. 1 Jan 46, 15p. incl. illus. tables (Rept. no. SM-11155) Unclassified report

AD-89 199 Accession No.
(Unannounced report)

Douglas Aircraft Co., Inc., Santa Monica, Calif.
PERFORMANCE OF LINEAR LT2-70 HYDRAULIC "O" RING PACKING USED WITH CHROME RETANNED LEATHER BACK-UP RINGS AT 3000 PSI PRESSURE, by E. R. Broadbent. 4 Apr 46, 7p. incl. illus. (Rept. no. SM-11809) Unclassified report

AD-89 767 Accession No.
(Unannounced report)

Electrical Engineering Research Lab., U. of Illinois, Urbana.
A CRITICAL SURVEY OF METHODS OF MAKING METAL-TO-CERAMIC SEALS, by R. Wellinger. 30 Apr 49 [12]p. incl. illus.
(Contract W33-038-ac-14742) Unclassified report

AD-89 828 Accession No.

Consolidated Vultee Aircraft Corp., Fort Worth, Tex.
EVALUATION OF B-36 INTEGRAL FUEL AND OIL TANK SEALANTS. Final rept. 18 Jan 55, 65p. incl. illus. tables, (Rept. no. FZA-36-314 suppl. to Rept. no. FZA-36-273, AD-47 671)
(Contract AF 33(600)5793; [Continuation of Contract AF 33(038)2182]) Unclassified report

"A representative group of commercially available sealing materials for integral fuel and oil tanks were evaluated in Convair's Engineering Chemical Laboratory, and their respective properties were obtained. In addition, improved sealants from Battelle Memorial Institute and Commercial products from this country and England were included in the program. Battelle Memorial Institute was under subcontract to Convair for development of improved sealants during the period extending from 15 March 1954 to 15 October 1954. As a result of this program it is concluded that integral tank sealants for use in B-36 airplanes are considerably
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AD-89 828

improved over those previously available. Leak-proofing qualities and service-life performance of the newly-developed sealants are expected to exceed by a considerable amount those available with current sealants. The findings of this program may also prove valuable in sealing integral tanks of all types of Air Force aircraft." (Contractor's abstract) (See also AD-47 671)

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design. The existing type of sealing gasket between the receptacle and housing will not provide an efficient moisture seal. A design for the sealing of the trailer-receptacle cable entrance eliminates the need for extra sealing accessories. The trailer-plug design consists of a modification of the weatherproof crimp-splice method for connecting wires. Test procedures were designed for detecting incipient failure before, during, and after aging in various environments. Protection is not complete primarily around electrical contacts and terminals, particularly when frequent disassembly is necessary. Variables which cause failure include tolerances and clearances which affect the confining volume/sealing material volume ratio; cold flow of resilient sealing materials with time and pressure; contraction and hardening of sealing materials because of low temperatures; improper material selections; and human error in assembly and maintenance. Neoprene materials had their greatest failure at -37° to -40°F.

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AD-91 555 Accession No.
(Unannounced report)

Material Lab., New York Naval Shipyard, Brooklyn.
EVALUATION OF SYNTHETIC RUBBER DRUM GASKETS SUBMITTED BY VARIOUS COMPANIES, by W. K. Gondek and M. Greenberg. Final technical rept. 3 Feb 56 [8]p. incl. tables (Lab. proj. no. 5094-32) Unclassified report

AD-92 941 Accession No.

General Hermetic Sealing Corp., Valley Stream, N. Y.
REUSABLE SEALS FOR ELECTRONIC EQUIPMENT, by William Brown and A. Razdow. Scientific quarterly rept. no. 2, 1 Sep - 30 Nov 55. 30p. incl. illus.
(Contract AF 33(616)2994) Unclassified report

Progress is reported on the continuation of tests on various enclosures, on the band sealers, and the heat reflecting baffles. The electronic multilocation temperature indicating device and 2 switching thermocouple systems were completed and calibrated. The heat gradient in and on the cans was determined on a series of 50 cans for different joints on the round and rectangular brass and steel enclosures. Joints extending out from the enclosures showed lower temperature gradients. The temperature in the steel containers were generally lower than in the brass containers. The square shaped can showed lower temperatures than the round can. Comparative tests of heat-transfer methods
(over)

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showed the band sealer and the inductive heating method to be more efficient than the soldering iron and the torch. Temperatures were higher during the unsealing phase. The heat distribution within the enclosure showed that the temperature increased toward the cover plate and close to the side walls of the can. A simple approach for effective protection of heat sensitive components within the can was developed. A semiautomatic leakage test device determined the leakage rate of each enclosure. Test results show that the smaller heat reflecting baffle is more efficient. (See also AD-75 568)

AD-94 444 Accession No.
(Unannounced report)

Detroit Arsenal, Center Line, Mich.
TEST OF COMPENSATING IDLER DOUBLE LIP SEALS IN THE HEAVY RECOVERY VEHICLE, M51, by William L. Harrison. Final rept. 29 Feb 56 [11]p. incl. illus. (Rept. no. 3462) Unclassified report

AD-94 590 Accession No.

North Electric Mfg. Co., Galion, Ohio.
DEVELOPMENT OF A HERMETICALLY SEALED TOGGLE SWITCH, by W. H. Blashfield and W. A. Roberts. Final rept. Oct-Dec 55. Nov 55, 5p.
(Contract AF 33(600)22898) Unclassified report

Sample headers were prepared with maximum glass insulation around the terminal sleeves. Tests for dielectric strength, at 70,000 ft altitude (1.3 in. Hg) and at 50,000 ft altitude (3.4 in. Hg), substantiated the ability of the design to meet the modified requirements of 500 v rms, 60 c at 50,000 ft. Tests results taken on 12 terminals to ground checked as follows: 550 v rms at 60 c for 50,000-ft altitude and 350 v rms at 60 c for 70,000-ft altitude. Investigations were made to determine the material and method for sealing the terminals into the sleeves in the header. Test results indicated that a 97½% Pb, 2½% Ag, 581°F solder is suitable for the sealing. A 220°F temperature differential existed between the material melting temperature and the solder
(over)

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used to seal the containers or to attach exterior leads to the terminals. The terminals are sealed into the header sleeves by using a controlled atmosphere oven to apply the required temperatures. The switches delivered were all of DPDT characteristic and may be tested as a DPDT, as a SPDT, and as a SPST. The design objective was accomplished by providing a switch wherein the power to operate the contacts and hold them operated was obtained with the use of permanent magnets arranged in a manner that would permit hermetic sealing of the contacts separate from the operating lever and thereby eliminate the use of resilient diaphragm seals to maintain the required resistance to infiltration of ambient conditions into the contact cavity. (See also AD-66 633, AD-75 909)

AD-94 674 Accession No.
(Unannounced report)

Naval Engineering Experiment Station, Annapolis, Md.
CARBON PACKING MATERIAL, by Walter E. Grant. Specification-development rept. 20 Dec 55 [9]p. incl. illus. tables (EES rept. no. 010023) Unclassified report

AD-94 749 Accession No.
(Unannounced report)

Boeing Airplane Co., Seattle, Wash.
DESIGN DATA FOR O-RINGS AND SIMILAR ELASTIC SEALS, by R.J. Dunsmoor, E.J. Quesenson, and others. Progress rept. 15 Oct 55-15 Jan 56. 35p. incl. illus. tables, 13 refs. (Document no. D-17416-1)
(Contract AF 33(616)2867) Unclassified report

AD-94 807 Accession No.

New York U. Coll. of Engineering, N. Y.
SEALING TECHNIQUES FOR WAVEGUIDE AND COAXIAL ROTARY JOINTS, by Milton M. Sackman, Michael Maccarone, and Fred H. Posser. Quarterly progress rept. no. 5, 1 Sep-30 Nov 55. [36]p. incl. illus.
(Contract DA 36-039-sc-64476) Unclassified report

Tests are reported on original and modified experimental models of a mechanical rotary seal. Initial test series on the original carbon-graphite sealing ring showed that at high pressures the temperatures rose above safe operating limits; at lower pressures, the unit ran within the safe operating temperatures. Subsequent test series were run with a sealing area about 1/2 the original and by regulating the secondary pressure separate from the internal pressure. In these tests, with the internal pressure creating a sealing force, the temperature rose too high when the secondary pressure was above 30 psi; normal torque values (over)

AD-94 807

went as high as 13 in.-lb, with infrequent fluctuations up to 26 in.-lb. In the third series of tests, the unit was modified so that the only sealing force would be that caused by the secondary pressure; the same carbon ring was used as in the second series of tests. During these tests, the internal pressure was kept at 60 psi and the unit was run at 900, 1400, and 2100 rpm. Results showed that the torque values and leakage rates remained constant regardless of speed, while the temperature increased with the speed. (See also AD-77 374)

AD-94 809 Accession No.

New York U. Coll. of Engineering, N. Y.
SEALING TECHNIQUES FOR WAVEGUIDE AND COAXIAL ROTARY JOINTS, by Milton M. Sackman, Michael B. Maccarone, and Fred H. Posser. Quarterly progress rept. no. 6, 1 Dec 55-29 Feb 56. [24]p. incl. illus.
(Contract DA 36-039-sc-64476) Unclassified report

Lubrication tests were conducted on the old experimental model of the mechanical rotary seal (brass experimental model). Socony-Vacuum gargoyle grease BRB no. 2, General Electric silicone grease 81482, and PD 603 grease meeting the requirements of MIL-G-3278 were each tested for 1 week with daily testing times of up to 8 hr of continuous running; maximum speeds of the tests reached 2100 rpm. These initial tests suggested that the gargoyle grease is unsatisfactory, and that more and longer tests should be conducted on the other 2 lubricants. Construction is reported on an experimental model of an RG-52 rotary joint size designed with consideration for microwave transmission needs. (over)

AD-94 809

This unit will accomplish a seal by using a nonrotating carbon surface moving against a rotating hardened steel surface. Except for the O-rings, bearings, the valve mechanism for the secondary pressure inlet, and sealing surfaces, the unit will be made of Al. Detailed design and operation characteristics, and possible alternate design modifications of the RG-52 rotary joint unit are described. (See also AD-94 807)

AD-94 922 Accession No.

Sciaky Bros., Inc., Los Angeles, Calif.
WELDING DEVELOPMENT PROGRAM FOR MUNITIONS CLOSURES, by S. M. Robelotto. Rept. no. 7, 1-30 Apr 56. 1 May 56, 5p. incl. illus. tables [CRL rept. no. ETF 420.6-2/7]
(Contract DA 18-108-cml-5860) Unclassified report

Thin-wall munition seals: The PMCO5STK welding machine for welding thin-wall munition closure caps was installed. Heavy-wall munition seals: A total of 40 specimens consisting of AISI C-1033 tubes and AISI C-1048 shells were welded. All Specimens showed leak tightness before and after 3-cycle drop tests from 12 ft nose down on a 1/2-in-thick steel plate resting on concrete. Eight of the 40 specimens were subjected to pushout tests which consisted of ramming the tube out of the shell with a 120,000 lb testing machine; the load to rupture varied from 51,000 lb to no failure at 100,000 lb. Of the 40 specimens, 25 were cross-sectioned, polished, and examined; cracking in some areas of the weld and (over)

AD-94 922

in the zone adjacent to the weld was noted in most of the 25 specimens. Basic welding-machine settings are tabulated for the 40 specimens. Thirteen of the specimens were welded without using postheat, and 16 were welded with a postheat. Postheat treatment did not adequately prevent cracking. Thirteen specimens were welded under a high forge force. (See also AD-84 396, AD-91 885)

AD-97 258 Div. 1/2, 14/3, 26/8

Parker Appliance Co., Cleveland, Ohio.
HYDRAULIC ACTUATOR ROD SEAL FOR HIGH TEMPERATURE OPERATION, by R. B. Resek. Rept. on Aircraft Hydraulic Systems. July 56. 27p. incl. illus. tables (WADC Technical rept. no. 56-288)
(Contract AF 33(600)29703) Unclassified report

A rod seal was required for hydraulic actuators which would operate with fluid and ambient temperature as high as 700°F. A test actuator and test stand were designed and built for testing sample seals from -65°F to 400°F (limited to 400°F by temperature limitations of fluids available at the time this program was initiated). Both static and cycling tests were run with metallic foil seals, tubular metal seals, and several types of asbestos seals. The fluid was OS-45 and pressures were from 10 psi to 3000 psi. A molded scarf-cut packing ring of asbestos with a metal mesh core was the only seal tested which met all of the specified requirements. This should provide a high (over)

AD-97 258

temperature hydraulic actuator rod seal suitable for aircraft use at high temperatures, although leakage may be slightly greater than is normally permitted. (Author)

AD-97 454 Accession No.

Bendix Aviation Corp., Eatontown, N. J.
PRODUCTION REFINEMENT CONTRACT ON SD-1064B TUBE (RTMA 6094) by R. H. Kuhnapfel. Final rept. vol. 1. 22 July 54, 1v. incl. illus. tables.
(Contract AF 33(600)21623) Unclassified report

The necessary refinement of tube design for production and the additional features of mount structure necessary for tubes which are to survive repetitive high level shock (50 g) are outlined. The final tube design of the 6094 incorporated ceramic element spacers, a skip-space heater, and minor dimensional shifts. The use of ceramics instead of micas, the use of nonex glass, and the use of techniques to produce tubes capable of prolonged operation at 300°C are discussed. The results of the use of ceramics in the finished tubes seemed acceptable. Tubes made with ceramics passed the 17 volume unit AF noise test limit which was current on the prototype tubes. A discussion is given of the equipment provided and the production rates. Results of the pilot run (over)

AD-97 454

showed the equipment is capable of producing a tube superior to the prototype currently available. It was concluded that the basic equipment was capable, with some duplication and supplementation, of production in the 100,000 per month category. Manufacturing, equipment and processing specifications are appended. An improved military control type of acceptance specification is proposed for the final tube, RTMA 6094. (See also AD-46 078)

AD- 97 547 Accession No.

General Hermetic Sealing Corp., Valley Stream, N. Y.
REUSABLE SEALS FOR ELECTRONIC EQUIPMENT, by William Brown and A. Razdow. Quarterly scientific rept. no. 3, 1 Dec 55-29 Feb 56. 7p. incl. illus. (Contract AF 33(616)2994) Unclassified report

Further investigations were conducted on heat-transfer methods. The tests with resistance wire imbedded in the cover of the can showed that special heat-deflecting devices were needed for the reduction of heat transmission. The results obtained for the electrical resistance heating method showed that variation of the surface of the enclosures made the method impractical. Band-heater tests were conducted with overlapped flanged covers. The solder was set up over the joint, and the heating element was wrapped around the joint. Further investigation of the application of the bandheater or preform heater showed asbestos and Fiberglas to be inadequate as insulating materials because of the high (over)

AD- 97 547

temperatures required. Heating elements with ceramic beads and a heat-reflecting clamp around the can were successfully used. Heat-transfer tests with induction heating devices showed good results for the steel container, and on the average, sealing was completed in 50 sec. Brass and aluminum containers did not work out as well. New methods of soldering and plating of aluminum were tested and evaluated. No conclusive results were obtained. A general procedure for sealing and unsealing of hermetically sealed enclosures was worked out. The oscillographic temperature indicator was further improved by the application of base line clamping diodes. (See also AD-92 941)

AD- 97 649 Accession No.

Chemical and Radiological Labs., Army Chemical Center, Md.
INVESTIGATION OF SOME SEALING METHODS FOR POLYETHYLENE SHEET AND FILM, by John R. Curen and Warren L. Price. 4 Oct 55, 8p. incl. table. (Special rept. no. CRLR 481) Unclassified report

By employing special techniques and procedures, high-frequency heating permits efficient and economical sealing of polyethylene, even in those instances where material thickness and complex item design present unique problems. (CRL conclusions)

AD- 97 767 Accession No.

Battelle Memorial Inst., Columbus, Ohio.
INVESTIGATION OF ENCAPSULENTS FOR GROUND ELECTRONIC EQUIPMENT. Final engineering rept. July 55-June 56. 1v. incl. illus. tables [Rept. no. RADC TR-56-102])
(Contract AF 30(635)2919) Unclassified report

An evaluation was made of encapsulating materials to determine the effect of encapsulation on electronic components at high frequencies. Three types of encapsulents were selected for extensive study: epoxides, polyesters, and foam materials. During the screening program, the following results were noted: (1) anhydride-hardened epoxides have better resiliency than the amine-hardened epoxides, and are generally favored for encapsulation purposes; (2) moderate temperature cures are essential to develop optimum physical and electrical properties; (3) plasticizers and inorganic fillers tend to degrade the electrical properties of encapsulents; (4) the electrical properties of the polyamide-epoxy alloys (over)

AD- 97 767

are undesirable at elevated temperatures; (5) the electrical grade polyesters are generally rigid and possess undesirable physical properties as encapsulents; and (6) the pure hydrocarbon materials are desirable dielectrics because of the absence of polar groups. Solid resins are too lossy for desirable encapsulation. Foam materials are considered more desirable for electronic equipment than the solid encapsulents because of their low dielectric constant and dissipation factor. For high-frequency encapsulation, a resin modified with urea formaldehyde microballoons proved superior to solid encapsulents. A series of resistors, capacitors, and inductors were evaluated before and after encapsulation in 3 types of resin: Castiplast, Epocast, and Eccofoam. Results indicate that foam encapsulents are the most desirable. In the other materials, a considerable change was noted in the resistance of resistors over the 5- to 100-mc range. The effect of encapsulation on capacitors was negligible above 10$\mu\mu$f. No measurable effects were noted following encapsulation of the inductors normally used for video or wide band amplifier peaking.

AD-98 106 Accession No.

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
DEVELOPMENT OF O-RING AND GLAND FOR SEALING RG-14A/U CABLE IN UG-665C/U CONNECTOR-RECEPTACLE, by J. Oser. Technical rept. 29 Feb 56 [10]p. incl. illus. (Rept. no. 67-13) Unclassified report

"An O-ring and a gland developed by the Rubber Laboratory for UG-665C/U connector-receptacle and RG-14A/U cable are described. The sealing capability of the O-ring was tested at room temperature at a pressure of 1000 psi. The O-ring maintained a watertight seal against RG-14A/U cable even after the cable and the O-ring assembly had been intermittently oven aged at 158°F, with no pressure applied, for a period totaling 64 hours. The Laboratory recommends the adoption of the O-ring and the gland for use in UG-665C/U connector-receptacle with RG-14A/U cable. The use of these two parts will greatly simplify the design of the connector-receptacle by replacing 7 of its present components." (RL abstract)

AD- 98 205 Accession No.

Frankford Arsenal, Philadelphia, Pa.
MODIFICATION OF BOXES, AMMUNITION, CALIBER .30, M19A1 AND CALIBER .50, M2A1, by H. L. Berkowitz. 25 Apr 56 [27]p. incl. illus. (Rept. no. 6; Memo. for record no. 1726) Unclassified report

Modifications in the design characteristics and specification requirements for 0.30-cal M19A1 and 0.50-cal M2A1 ammunition boxes were investigated to improve quality and facilitate manufacturing. The grooved-type gasket is superior to the solid-type gasket in meeting the performance requirements of the ammunition box. A greater load is required to compress a solid-type gasket than a grooved-type gasket. A lesser degree of convexity of box cover would result from the use of a grooved-type gasket. A greater loss in compression is obtained when using a solid-type gasket, since it affords a greater distortion of the box because of the higher reactive load exerted. A direct correlation exists between the load required to close the M2A1 ammunition (over)

AD- 98 205

box (equal to the load exerted by the gasket on the box) and the loss in gasket compression. Ammunition boxes fitted with both the grooved- and solid-type gaskets satisfactorily withstood environmental storage of 24 hours at 165 and -70°F. Reversal of the body seam from the hinge end to the latch end of the box and redesign of the body seam minimized the scrubbing action at the hinge end of the box and eliminated cut gaskets and the localized excessive compression caused by sharp protrusions at the top of the body seam. Modification of the body and cover hinge would have an adverse effect on the functioning of the box after rough handling. The type of paint for ammunition boxes as currently specified affords adequate coverage and protection.

AD- 98 226 Accession No.

Naval Ordnance Lab., White Oak, Md.
PROBLEMS ENCOUNTERED IN THE DEVELOPMENT OF POTTED ELECTRONIC DEVICES FOR A SPECIFIC ORDNANCE APPLICATION, by R. H. Flack and J. O. Mobley. 14 May 56, 22p. incl. illus. tables. (NAVORD rept. no. 4248) Unclassified report

"The development of three devices at the Naval Ordnance Laboratory required encapsulation of electronic assemblies for protection against shock and temperature-humidity effects. Partly, at least, due to the fact that these assemblies were somewhat larger than many encapsulated units, the original potting material cracked or caused destruction of the embedded components when subjected to the JAN Temperature-Humidity Test. The 28 day duration of this test and the necessity for providing a satisfactory material in a short time led to a program of simultaneous evaluation of a number of materials which appeared promising in generally similar applications at NOL and elsewhere. Each material (over)

AD- 98 226

was checked for its ability to withstand shock, vibration and temperature-humidity tests. Insulation resistance was measured both before and after temperature-humidity testing. The program eventually encompassed 27 material combinations, two of which proved to have exceptionally good properties for the application at hand. Additional experimentation was necessary to determine methods of applying pre-potting protective coatings to certain components, and to select components which would provide suitable ruggedness and stability. The case history of the program depicts the all-out effort which often must be expended in solving problems of this type in connection with accelerated development projects, and points out the need for long range accumulation of practical data on encapsulation to facilitate their solution." (NOL abstract)

AD-98 646 Accession No.
(Unannounced report)

White Sands Proving Ground, N. Mex.
NIKE. INVESTIGATION OF LEAKAGE OF NIKE FUEL-PIPE ASSEMBLY (NIKE TEST PLAN NO. 63). Oct 55 [16]p. incl. illus. tables (Technical memo. no. 265). Unclassified report

AD-99 469 Accession No.

Dugway Proving Ground, Utah.
AGENT LEAKAGE FROM THE E101R3 CLUSTER ADAPTER CONTAINING LEAKING E54R6 BOMBLETS. 23 Mar 56, 30p. incl. table (DPGR rept. no. 170; CW 27-54) Unclassified report

Tests were conducted to determine if leaking E54R6 bomblets would cause the entire E101R3 cluster adapter to be unsafe for normal storage, transportation, or handling. Eight specially constructed containers which underwent a 5-mo open-storage test were divided into 2 groups: group 1 contained complete E101R3 clusters in which 1 or more of the E54R6 bomblets leaked, and group 2 contained only the E101R3 cluster adapters with a container holding 500 cc of agent welded on the inside of the nose plate to simulate a leaking bomblet. Leaking bomblets in the E101R3 clusters are safe for normal storage.

AD-99 495 Accession No.
(Unannounced report)

Engineer Research and Development Labs., Fort Belvoir, Va.
TWELVE-INCH, BENCH-TYPE, ELECTRIC-MOTOR-DRIVEN SEAL LAPPING MACHINE AND ACCESSORIES, by Henry Wilson. 6 Aug 53, 43p. incl. illus. tables (Rept. no. 1310) Unclassified report

AD-99 706 Accession No.

Naval Ordnance Test Station, Inyokern, China Lake, Calif.
DETECTION OF O-RING ON IGNITER CUP OF 2.0-INCH GIMLET ROCKET MOTOR, by J. I. Bujes and D. C. Harms. 23 Apr 56, 13p. incl. illus. (NOTS 1440; NAVORD rept. no. 5241) Unclassified report

"A method of detecting O-rings in assembled rocket motors is described. The detection is performed by the application of cadmium sulfide or cadmium selenide crystals in the presence of X-rays. For field inspection, a 'Go no-go' gage was developed which might be combined with an automatic inspection system for ejecting the defective motor from the conveyor line." (NOTS abstract)

AD-99 741 Accession No.
(Unannounced report)

Material Lab., New York Naval Shipyard, Brooklyn.
QUALIFICATION TEST OF SYNTHETIC RUBBER GASKET MATERIAL, SUBMITTED BY BOSTON WOVEN HOSE AND RUBBER COMPANY, BOSTON, MASSACHUSETTS. Final technical rept. 16 Aug 56, [6]p. incl. table (Lab. proj. no. 5542-16)
Unclassified report

AD-100 053 Accession No.
(Unannounced report)

Naval Gun Factory, Washington, D.C.
EVALUATION OF OPTICAL SEALS AND OPTICAL SEALING METHODS: ROTARY TYPE SEALS.
7 May 56 [12]p. incl. illus. table (Rept. no. NGF-T-16-56; NAVORD rept. no. 5160) Unclassified report

AD-100 300 Accession No.
(Unannounced report)

Army Medical Research Lab., Fort Knox, Ky.
THE EFFECT OF LEAKAGE FROM CLOSURES UPON THERMAL PROTECTION, by Steven M. Horvath, John E. Wagar, and Howard Golden. Rept. on Studies of Cold Weather Clothing. 14 June 44 [22]p. incl. illus. tables (Sub-proj. no. 20-1) Unclassified report

AD-100 488 Accession No.

Quartermaster Food and Container Inst. for the Armed Forces, Chicago, Ill.
VACUUM REQUIREMENTS FOR CANNED TOMATOES. Jan 56, 64p. incl. illus. tables. Unclassified report

Symposium Sponsored by the Quartermaster Food and Container Inst., 18 Feb 55.

Contents:
Introduction
Purpose of the symposium
Importance of vacuum in canned foods
Procurement and storage aspects of the problem
Results of survey of current commercial practice
Procurement of canned tomatoes
The problem of vacuum as it affects depot operations
Further considerations in attaining prolonged shelf life
Relationship of vacuum to shelf life
Extending shelf life by means of low temperature storage

(over)

AD-100 488

Minimum vacuum requirements - recommendations and decisions

AD-100 836 Accession No.
(Unannounced report)

Krautter-Weber Tool Co., Newark, N. J.
DRIVE, MECHANICAL, ROTARY SEAL (HIGH SPEED) by A. Raszowitz. Monthly letter rept. no. 11, 7 Oct-6 Nov 50, 15 Nov 50, 5p. (Rept. no. 16) (Contract W36-039-sc-44630) Unclassified report

AD-101 003 Accession No.
(Unannounced report)

Lincoln Lab., Mass. Inst. of Tech., Cambridge.
HIGH-PRESSURE LEAK DETECTOR, by William Paul and D. M. Warschauer. 12 May 55, 1p. [Reprint no. JA-347] (In cooperation with Harvard U., Div. of Applied Science)
[Contract AF 19(122)458, Proj. Lincoln]
Unclassified report

Reprint from Review of Scientific Instruments 26:731, July 55.

AD-101 096 Accession No.
(Unannounced report)

Power Plant Lab., Wright Air Development Center, Wright-Patterson Air Force Base, Ohio.
QUALIFICATION TESTS ON PARKER APPLIANCE CO. "O" RINGS, COMPOUND 1011-10, MIL-P-5315A, by C. D. Cannon. 11 Oct 54, 3p. incl. table (Technical note no. WCLP-54-183) Unclassified report

AD-101 097 Accession No.
(Unannounced report)

Power Plant Lab., Wright Air Development Center, Wright-Patterson Air Force Base, Ohio.
EVALUATION OF TESTS OF FUEL "O"-RINGS MANUFACTURED FROM PLASTIC AND RUBBER PRODUCTS COMPANY COMPOUND NOS. 457-60 AND 458-60 TO SPECIFICATION MIL-P-5315A, by E. J. Hildebrand. 24 Sep 53, 3p. incl. table (Technical note no. WCLP-53-266) Unclassified report

AD-101 098 Accession No.
(Unannounced report)

Power Plant Lab., Wright Air Development Center, Wright-Patterson Air Force Base, Ohio.
EVALUATION OF TESTS OF FUEL "O" -RINGS MANUFACTURED OF LINEAR INCORPORATED COMPOUND NO. LEX-60 TO SPECIFICATION MIL-P-5315A, by E. J. Hildebrand. 11 Mar 53, 3p. incl. table (Technical note no. WCLP-53-75)
Unclassified report

AD-101 099 Accession No.
(Unannounced report)

Power Plant Lab., Wright Air Development Center, Wright-Patterson Air Force Base, Ohio.
EVALUATION OF TESTS OF FUEL O-RINGS MANUFACTURED FROM ALBERT TROSTEL AND SONS COMPANY COMPOUND C-901 TO SPECIFICATION MIL-P-5315A, by E. J. Hildebrand. 10 Feb 53, 2p. (Technical note no. WCLP-53-40) Unclassified report

AD-101 331 Accession No.

Dugway Proving Ground, Utah.
EVALUATION OF THE PRESS-FIT CLOSURE OF THE 155MM M121 AND 105MM M360 SHELL. 24 Feb 56, 272p. incl. illus. tables (Rept. no. ETLR-6-55)
Unclassified report

A total of 3378 M121 and 1582 M360 GB-filled shell were subjected to various sequences of being exposed to temperature extremes (-80° and 160°F), being dropped twice from a truck moving at 30 mph, being dropped thrice from a height of 5 ft, and being vibrated at 1 g for 10 hours. Two of the M360 shell and none of the M121 shell developed agent leakage. It is concluded that: (1) the M121 shell with the current press-fit closure design can be subjected to rough handling and environmental extremes and still remain leakproof to toxic agent. This is also anticipated to be true for the M360 shell, pending final evaluation of the cause of the two agent leakers; (2) with the retention of the 0.0030- to 0.0060-in. interference fit range specification, it can be anticipated
(over)

AD-101 331

that the components of regular production will exhibit the same characteristics as the shell tested at Dugway Proving Ground; (3) there is no apparent difference in respect to agent leakage between shell deviating from specification requirements and those meeting the requirement of Class A shell, as defined in this report; and (4) within the scope of these tests, the press-fit closure is satisfactory with respect to storage, transportation, handling, and safety. (Extracted from report)

AD-101 653 (Unannounced report) Accession No.

Becco Chemical Div., Food Machinery and Chemical Corp., Buffalo, N. Y.
STATIC SEALS FOR CONCENTRATED HYDROGEN PEROXIDE. [1954] 10p. Unclassified report

Presented at Symposium on Aircraft High Temperature Static Seals for All Media, Cleveland, Ohio, 18-19 May 54.

AD-102 156 Accession No.

Pacific Div., Bendix Aviation Corp., North Hollywood, Calif.
FRICTION STUDY OF HYDRAULIC SEALS, by Richard E. Tisch. Rept. on Aircraft Hydraulic Systems and Hydraulic Seal Development. June 55, 63p. incl. illus. tables (WADC Technical rept. no. 55-228) (Contract AF 33(616)183) Unclassified report

"Design of a machine to measure hydraulic packing breakout and running loads is described. Test conditions, as specified by Exhibit 'A' of Contract No. AF 33(616)-183, are described. The method of establishing the final test procedure is described. Gross loads measured in test runs are tabulated. Results of tare-load lap seal runs are tabulated, and graphs show the method of arriving at probable tare loads. Resultant net friction loads are charted to show the relative influence of controlled variables. In order to comply with the intent of the contract it was adjudged necessary to go beyond the letter of the contract. The rea-
(over)

AD-102 156

sons for and results of such additional tests are described. Significant characteristics of packing friction conditions are discussed. It is concluded that data secured in these tests, when combined with existing data on the friction of packings of various diameters, cross-sections, materials and shapes, will enable the designers of airplane and missile hydraulic systems to determine more accurately the actual breakout and running loads. Recommendations are made for additional tests of similar and related nature." (Contractor's abstract)

AD-102 222 Accession No.
(Unannounced report)

Aberdeen Proving Ground, Md.
DEVELOPMENT OF A PROCUREMENT SPECIFICATION FOR ANTILEAK, COMPOUND, RADIATOR, by Virgil O. Hatch. 28 Dec 55, 25p. incl. illus. tables. (Proj. no. TB5-8010A rept. no. 20; Engineering Lab. rept. no. 24) Unclassified report

"Seven commercially available antileak compounds, from six manufacturers were investigated and evaluated for their ability to block 1/32 inch diameter holes and hold under pressure of 15 p.s.i. when employing water or ethylene glycol coolants. A simple leak test unit simulating a vehicle cooling system was constructed for the evaluation. A procurement specification was drafted, based on the physical and chemical properties of the commercial brand compounds. All seven commercial antileak compounds were found to effectively plug 1/32 inch diameter holes and hold under a pressure of 15 p.s.i. when employing water as a coolant.
(over)

AD-102 222

Only three of the compounds were found to effectively plug 1/32 inch diameter holes and hold under a pressure of 15 p.s.i. when employing ethylene glycol antifreeze solutions." (APG summary)

AD-103 928 Accession No.
(Unannounced report)

Naval Ordnance Lab., White Oak, Md.
KINETIC AND STATIC FRICTION OF THE "O" RING TYPE GASKET, by L. E. Hoblia, S. A. Humphrey, and T. E. Dinsmoor. 25 Mar 52, 26p. incl. illus. (NAVORD rept. no. 2281) Unclassified report

"Military applications of small shaft seals are common in underwater ordnance devices. Not infrequently, installations are required where the driving power available to overcome shaft seal friction losses is small, and the design engineer must be able to predict these losses with some degree of precision. Available data from which such predictions can be made is limited. Therefore to provide supplementary data on the character of the frictional forces to expect when employing 'O' ring type gaskets, this survey was undertaken by the Torpedo Engineering Division of the Naval Ordnance Laboratory and the results are presented herein in the form of design work sheets." (NOL abstract)

AD-105 235 Accession No.
(Unannounced report)

Boeing Airplane Co., Wichita, Kans.
SEALING DOCUMENT FOR B-47B AND B-47E AIRPLANES. 16 May 49. rev. 17 Feb 56 [402]p. incl. illus. (Document no. D-9840) Unclassified report

AD-105 415 Accession No.

Rock Island Arsenal Lab., Ill.
THE DETECTION OF NITROGEN LEAKAGE IN A HYDROPNEUMATIC RECOIL MECHANISM BY MEANS OF A TRACER, by Harry C. Muffley. Technical rept. 18 July 56, 14p. incl. illus. tables. (Proj. no. TB5-5010B; rept. no. 3; RIA Lab. rept. no. 56-2077) Unclassified report

A technique was developed for determining the amount of N that leaks past the floating piston of a hydropneumatic recoil mechanism into the recoil oil. This value would be used to establish the serviceability of the mechanism. The leak detector is a GE type H-1 which operates on the basic principle of the halide torch commonly used in refrigerator work and is sensitive to halogen but not to N. The instrument will detect a leak so small that in 1 yr one-hundredth of an ounce of Freon would pass the opening. The leak detector was limited in establishing the magnitude of the leak, but it did indicate qualitatively that gas does (over)

AD-105 415

pass the floating piston of a hydropneumatic recoil mechanism into the coil.

AD-106 671 Accession No.

Naval Air Test Center, Patuxent River, Md.
EGLIN AIR FORCE BASE, CLIMATIC LABORATORY COLD WEATHER TEST OF TEFLON BACK-UP RINGS AS INSTALLED IN THE HYDRAULIC AND PNEUMATIC SYSTEMS OF THE F7U-3 AIRPLANE. Rept. no. 1 (Final) 1 Aug 56 [28]p. incl. illus. tables. (Proj. TED no. PTR-AC-1301. 13. 1; Serial no. ST37-222) Unclassified report

This report covers the cold-weather evaluation of the Teflon back-up ring installed in lieu of the AN 6291 leather back-up ring in the universal fittings of the pneumatic and hydraulic systems of the F7U-3 airplane. The test program consisted of general operational tests of those systems down through -65°F. It is concluded that: (a) the Teflon back-up ring is superior to the AN 6291 leather back-up ring as used in the universal type fitting, (b) the MS "flareless" type fitting is the best fitting tested for use in 3000-psi hydraulic and pneumatic systems, (c) operation of the inverter (P/N (over)

AD-105 671

NE5109-2) is unacceptable at cold soaked temperatures below 0°F, and (d) the Bendix selector valves are unsatisfactory for operation at 0°F and below. (Extracted from report)

AD-105 754 Accession No.

Kentucky Research Foundation, Lexington.
AN INVESTIGATION OF SOME PROPERTIES OF SOLUTIONS OF DIPOLAR IONS IN SEVERAL NON-AQUEOUS SOLVENTS, by Paul G. Sears. Final rept. for Sep 55-Aug 56. 6 Sep 56, 4p. (Contract DA 33-008-ORD-1153) Unclassified report

The dielectric properties of solutions of dipolar ions in nonaqueous solvents: The dielectric increment per mole of solute per 1000 g of solvent was determined for (1) sulfamic acid (I) in N, N-dimethylpropionamide(II), N, N-dimethylbutyramide (III) and several dimethyl sulfoxide (IV) and N, N-dimethylacetamide (V) mixtures at 25° and in IV and V at 25°, 35°, 45°, and 55°C; (2) sulfanilamide in N, N-dimethylformamide, II, acetone, 2-butanone, and 3-pentanone at 25°, in N-methylacetamide (VI) at 33°C and in V and IV at 25°, 35°, 45°, and 55°C; and (3) several other sulfa drugs in IV at 25°C and in VI at 33°C. The dielectric increment for I in each of the amides was about 13 dielectric (over)

AD-105 754

units/mol of solute/1000 g of solvent at 25°C but for I in IV was less than 8. At 25°C, the dielectric increment/mole of sulfanilamide/1000 g of solvent was greater than 13 in each of the amides and between 9 and 11 in the 3 ketones. Viscosities of solutions of dipolar ions and other substances in nonaqueous solutions: Absolute viscosities were determined at 25°C for (1) I, p-aminobenzoic acid, benzenesulfonamide, sulfanilamide, and urea in IV; (2) I and KSCN in V; and (3) I in II. For IV solutions, the increase in viscosity with increasing molal concentration was sulfanilamide > I > p-aminobenzoic acid > benzenesulfonamide > urea. For V solutions, the increase in viscosity with increasing molal concentration is greater for I than for KSCN. The increase of viscosity with concentration of I for II solutions is about the same as for V but is greater than that for IV. Densities and apparent molal volumes: The densities of solution of I in II, III, IV, V and H_2O were determined at 25°. The apparent molal volume of I in each solvent was calculated to be less than the theoretical value and was almost constant for the 3 amides as solvents.

AD-106 511 Accession No.

Naval Gun Factory, Washington, D. C.
IMPREGNATION OF POROUS METAL CASTINGS, by L. F. Hatakeyama. Progress rept. 19 July 56, 5p. (Technical rept. no. NGF-T-41-56; NAVORD rept. no. 5189) Unclassified report

A study is being conducted on impregnants and impregnating methods to utilize the improvements brought about by the continuing development of plastomeric materials and processes in industry. Tentatively, it is hoped that the results of the study will be used to consolidate and modernize government specifications. A review of current government specifications or impregnants and impregnating methods showed that there is a notable absence of specifications governing the impregnation of ferrous metal castings. A survey showed that the drying oils and sodium silicate solutions are being displaced by the plastomeric sealants which are mostly phenolformaldehyde and polyester-styrene types. Mention is made of a new application of plasto- (over)

AD-106 511

meric materials (expoxide resins) to impregnate, encapsulate, and pot electrical components by methods similar to those used in the impregnation of castings. Except for mention of Cu interference with polymerization of the obsolete, unstabilized polyester-styrene sealants, it was believed that there is no condition that would prevent the impregnation of any particular casting alloy by any of the existing plastomeric sealants. A vacuum-pressure apparatus was set up to impregnate porous test specimens. The impregnation procedure to be followed is roughly outlined in MIL-STD-276 (impregnation of nonferrous metal casting), and inorganic and tung oil impregnants will be used to permit a comparison. The apparatus consists of a 2-gal impregnating tank, a Victor A-375 compressed air regulator, a compound gage, a Cenco rotary vane pump, and the necessary fittings.

AD-106 719 Accession No.

Consolidated Vultee Aircraft Corp., Fort Worth, Tex.
POD, INTEGRAL FUEL TANK, SEALING SYSTEM FUEL TIGHTNESS AND STRUCTURAL EVALUATION TEST PROGRAM FOR B-58 PODS, by C. E. Sisler. 19 Dec 55, rev. 20 June 56, 5p. incl. illus. (Rept. no. FZM-4-395; Supersedes rept. no. FZM-4-395, AD-95 154) (Contract AF 33(038)21250) Unclassified report

AD-106 754 Accession No.

Armour Research Foundation, Chicago, Ill.
DEVELOPMENT OF HIGH-LOW TEMPERATURE AIRCRAFT CANOPY SEALS. PART I. NEW MATERIAL SEALS OF CURRENT DESIGN, by Joseph S. Islinger. June 56, 44p. incl. illus. tables. (WADC Technical note no. 55-441, pt. 1) (Contract AF 33(600)27588) Unclassified report

"Silicone rubber seals were evaluated in Part I of a program to develop high-low temperature cabin pressurization and rain seals to be used with present and future aircraft canopies. Three seal designs considered to be representative of diaphragm, inflatable and non-inflatable canopy seals on current aircraft were evaluated. The seals fabricated from recently developed silicone elastomers, were to be capable of functional operation at extreme temperatures of -85°F and +200°F. The evaluation tests were conducted with each seal installed in a test fixture simulating a 'half-size' aircraft canopy. The seals were subjected to ambient tempera- (over)

AD-106 754

tures of +75°F, -85°F, and +200°F and were checked for leakage while the canopy was pressurized. The seals were also subjected to 25 open-close pressure cycles at a temperature of -65°F and 1000 open-close pressure cycles at room temperature. The inflatable seal generally appeared to be more effective than the non-inflatable one in sealing the canopy fixture against a maximum differential pressure of 12 psi. The diaphragm seal was not evaluated because a satisfactory installation could not be obtained. All designs are capable of preventing moisture from entering the pilot enclosure when the canopy is closed. However, the inflatable seal must be inflated in order to do so." (Contractor's abstract)

AD-106 777 Accession No.

New York U. Coll. of Engineering, N. Y.
SEALING TECHNIQUES FOR WAVEGUIDE AND COAXIAL ROTARY JOINTS, by Milton M. Sackman, Michael B. Maccarone, and Fred H. Posser. Quarterly progress rept. no. 7, 1 Mar-31 May 56. 25p. incl. illus. table.
(Contract DA 36-039-sc-64476) Unclassified report

An experimental model of an RG-52 rotary seal (AD-94 809) was subjected to 400 hr of testing. The nitrogen supply tank, the pressure regulator(s) and the values, and routing assemblages of the test set-ups are described. Results from these tests indicate the soundness of the design of this type of a rotary seal. The torque values obtained during normal running were no higher than 1 3/4 in.-lb. The temperature rise was such that there was less than 30°F differential between room temperature and the rotary seal outer wall temperature. The leakage rate amounted to 126 x 10^6 cc/yr based on 1 atm and a working pressure of 60 and 59 psi. (over)

AD-106 777

The grease used in the bearing was supplied by the Signal Corps (PD-603) and the grease used at the seal surfaces was manufactured by Lehigh Chemical Co and is designated L-793 synthetic instrument grease. Fabrication is reported of 2 additional RG-52 rotary seals.

AD-107 274 Accession No.

Armour Research Foundation, Chicago, Ill.
HERMETIC SEALING TERMINALS, by Geroge A. Forster. Rept. for 15 July 54-30 Apr 56 on Improved Electronic Components. 46p. incl. illus. tables.
(WADC Technical rept. no. 56-251)
(Contract AF 33(600)27674) Unclassified report

"The purpose of this project was determine the requirements for hermetic sealing terminals in military airborne electronic equipment and to compare these requirements with the capabilities of present terminals. Information from the Air Force, manufacturers, and users of hermetic terminals was analyzed to determine requirements and tests suitable for terminals. The important electric characteristics are dielectric strength, corona voltage, current capacity, and insulation resistance. Tests related to environmental conditions include temperature aging, temperature cycling, temperature shock, vibration, mechanical shock, moisture resistance, terminal strength, salt spray and gas leakage. Approximately 100 groups of 30 each, of (over)

AD-107 274

all typical types and manufacture of hermetic terminals, were tested. About 70 per cent of the groups failed in one or more respects. One cause of high failure rate was the high initial temperature rating of some groups. If re-tested at a lower temperature rating these groups may pass. It was found that vibration and shock tests were not necessary to evaluate hermetic terminals. Allowable gas leakage should be less than 10^{-9} cubic centimeters per second at a differential pressure of one atmosphere and a temperature of 25°C. More leakage than this indicates a seal which will deteriorate to complete failure. Surface cracks in vitreous insulations are few and do not constitute a major problem. The moisture-resistance test caused the most failures. These failures were perhaps due to the accentuating of incipient failures from previous tests. No one type or style of terminal is markedly superior nor is classed as general purpose." (Contractor's abstract)

AD-107 713 Accession No.

Naval Ordnance Lab., White Oak, Md.
EVALUATION OF SEALING METHODS FOR BASE FUZE, MK 28, by A. R. Timmins. 9 May 56 [27]p. incl. illus. tables, 17 refs. (NAVORD rept. no. 3946) Unclassified report

"An evaluation program was conducted on two sealing methods used on the Base Detonating Fuze Mk 28. Approximately 250, 200, and 60 fuzes respectively were used in the tests on (a) metal injection sealed fuzes, (b) seam welded fuzes, and (c) regular production fuzes. Leak tests were conducted using a Mass Spectrometer Leak Detector on fuzes as received and after various types of conditioning tests. In addition, detonator safety and functioning tests (gun fired and initiated by water impact in tests conducted at the Naval Proving Ground, Dahlgren, Virginia) were made. Fuzes sealed with the metal injection method and those sealed with the seam weld method had excellent seals which were not affected by adverse environmental conditioning nor (over)

AD-107 713

by mechanical rough handling, transportation vibration, etc." (NOL abstract)

AD-107 802 Accession No.
(Unannounced report)

Aerojet-General Corp., Azusa, Calif.
DEVELOPMENT OF AN IMPROVED BOMB SEALANT, by J. M. Wrightson. Informal progress rept. no. 1, 10 May-9 June 56. 21 June 56, 3p. (Rept. no. L 1911-1)
(Contract DA 28-017-501-ORD-2115)
Unclassified report

"The project has been initiated and preliminary studies have been made to determine the most feasible methods to be followed in developing suitable materials for use as a bomb sealant. Procurement of raw materials and equipment is in process, and it is expected that formulation and testing work can be started in the next report period. The initial emphasis will be placed on development and evaluation of foamed-in-place polyurethane compositions." (Contractor's summary)

AD-107 960 Accession No.

Army Ballistic Missile Agency, Huntsville, Ala.
TESTS WITH INFLATABLE SEALS USING SURGICAL RUBBER TUBING, by Oscar C. Holderer and Robert A. Winkless. 10 July 56, 23p. incl. illus. (Research rept. no. 1R15) Unclassified report

"The design of the tubing connectors, the geometry of the tubing and the tubing retaining groove were found to be satisfactory for the intended application. It was observed that after deflation the tubing will retract properly into the groove to prevent damage by chafing or pinching. The relationship between gap size, inflating pressure, and pressure differential across the seal has been established; empirical equations have been presented. The following trends are deduced:
(a) Each gap size is associated with a definite upper limit for the inflating pressure before extrusion occurs. This relationship can be approximated by an exponential function where the gap size appears as the exponent.
(b) The minimum inflating pressure, necessary to seal (over)

AD-107 960

against a differential pressure, is equal to this pressure differential plus a fixed value which is essentially independent of the gap size and the pressure differential to be sealed. (c) The upper limit of pressures for which inflatable seals of this type may be used is dictated mainly by the size of the gap. At higher pressures and if gas is the medium which is to be sealed the tubing must be inflated before the pressure differential is established. Otherwise the rubber tubing will become damaged by 'burning' just prior to sealing." (Contractor's conclusions)

AD-108 632 Accession No.
(Unannounced report)

Krautter-Weber Tool Co., Newark, N. J.
DRIVE, MECHANICAL, ROTARY SEAL (HIGH-SPEED) by A. Razdowitz. Monthly letter no. 6, 7 Feb-6 Mar 50. 10 Mar 50, 5p. (Rept. no. 8)
(Contract W36-039-sc-44520) Unclassified report

AD-108 720 Accession No.

Ontario Research Foundation (Canada).
INVESTIGATION INTO THE SEALING OF QUARTZ CRYSTAL HOLDERS TYPE HC-6/U, by B. W. Schumacher. Final rept., Nov 54-Mar 56. 1v. incl. illus. tables.
(Contract P 69-10-116) Unclassified report

Solder sealing as a satisfactory method of hermetically sealing quartz crystal units in the crystal holder type HC-6/U was investigated. The investigation included (1) studies of the effects of external conditions such as temperature, humidity, and salt spray on the sealing to be followed by recommendations for refinements in the technique of solder sealing; and (2) development of a reliable method of production testing of the crystal seals. A high percentage of crystal units which were stored for some period of time showed a failure called dipping. The reason for the dips is probably the condensation of water vapor on the crystal faces; other volatile materials are also considered such as residues (over)

AD-108 720

of the soldering flux. Destructive tests showed that leaks were present; they were attributed to poor soldering. (See also AD-82 917)

AD-109 660 Accession No.
(Unannounced report)

Raytheon Mfg. Co., Waltham, Mass.
CERAMIC-METAL SEAL OUTGASSING STUDY, by R. J. Dyer and M. L. Weiss. Quarterly scientific rept. no. 1, 1 Apr - 30 June 56. 6p. incl. illus. (Rept. no. QK-608-51; [AFCRC TN-56-757])
(Contract AF 19(604)1826) Unclassified report

AD-110 287 Div. 8, 14, 26
(Unannounced report)

Raytheon Mfg. Co., Waltham, Mass.
CERAMIC-METAL SEAL OUTGASSING STUDY, by R. J. Dyer and M. Weiss. Quarterly scientific rept. no. 3 on QK608, Oct-Dec 56. 3p. (AFCRC-TN-57-153)
(Contract AF 19(604)1826) Unclassified report

AD-110 390 Div. 30/3

Maryland U., College Park.
VARIABLE MICROLEAK, by John L. Davis. 15 Nov 56 [4]p. incl. illus. (Technical rept. no. 59; OSR TN-56-569)
(Contract AF 18(600)1582) Unclassified report

A satisfactory method of testing helium sensitive mass spectrograph leak detectors is presented. The detector is evacuated to a pressure within its operating range and then the pressure is raised to a standard amount by admitting a gas mixture containing a known concentration of helium. The resulting indication is a direct measure of the sensitivity. A valve or leak that can be varied is essential to raise the pressure the known amount. A valve which is easily made and fulfills all the requirements for this method of testing is described as well as an apparatus for preparing a known concentration of helium. The soft brass valve body is made on a lathe and the seat is formed by reaming with a 5/0 standard taper pin reamer. It is recommended that the
(over)

AD-110 390

reaming be done on a lathe to insure accurate centering. The threads on the stem were cut after the body was threaded, and the body was used as a gage for the diameter of the stem threads. The leakage rate was variable from 0 to 4×10^{-7} standard liters/sec with about 1/2 turn of the stem. It was possible to control the pressure in the leak detector to within $\pm 3 \times 10^{-7}$ mm Hg. With an Octoil Manometer, it is possible to measure a concentration of helium in air of about 1 part in 3000 to about 10%.

AD-110 636 Accession No.

Koppers Co., Inc. [Baltimore, Md.]
EVALUATION OF DESIGNS AND MATERIALS FOR HIGH SPEED-HIGH TEMPERATURE SHAFT SEALS FOR TURBOJET ENGINE APPLICATIONS, by Ernest J. Taschenberg. Rept. on Gas Turbine Technology. 15 May 56, 66p. incl. illus. tables (WADC Technical rept. no. 56-287)
(Contract AF 33(616)3004) Unclassified report

Includes AD-110 636(a)

"With the constant upward trend in operating conditions in aircraft turbojet engines, it has been necessary to re-design engine rubbing shaft seals and find new materials to meet these conditions. This program was initiated to facilitate the development of a shaft seal which would operate satisfactorily under extended conditions of speed, temperature and pressure. Since shaft seals necessarily involve rubbing surfaces, a large number of materials were investigated to de-
(over)

AD-110 636

termine those having promising wear characteristics at room temperature and at high temperature. The subject of shaft seal design was reviewed with respect to the adaptability of such new materials to the several designs incorporating these materials." (Contractor's abstract)

AD-111 249 Accession No.

Aberdeen Proving Ground, Md.
TEST OF T41E1 RUBBER BLADDER FUEL CELLS, by D. A. J. Misiora and J. R. Fraylick. Final rept. 30 Mar 56 [26]p. incl. illus. (Proj. no. TT1-649A, rept. no. 2) Unclassified report

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AD-112 953 Accession No.

Sciaky Bros., Inc., Los Angeles, Calif.
WELDING DEVELOPMENT PROGRAM FOR MUNITIONS CLOSURES, by S. Robelotto. [Monthly progress] rept. no. 12, 1-30 Sep 56. 1 Oct 56, 42p. incl. illus. tables [CWL rept. no. 420:6-2/12]
(Contract DA 18-108-cml-5860) Unclassified report

The 4 thin-wall munition geometries were successfully sealed with each closure design (a 45° double bevel, a flat disk, and a 7/8-and a 1 3/8-in. Welch plug). The 4 shell-blank geometries may be welded on the PMCO2STK Spotwelder, but the geometries with hole diameters of 1.25 in. require almost maximum current output and long welding times. The 45° double-bevel and flat-disk closure caps were the most reliable and satisfactory closures. A method of measuring current is described for use on the weld of heavy wall munition seals. The 3 primary leads are inserted in the current transformer so that their currents induce a voltage in the current transformer. An ammeter is connected to
(over)

AD-112 953

the current transformer, and the primary and secondary current of the welding transformer are calculated by the use of the transformer ratio. (See also AD-107 534)

AD-113 178 Accession No.
(Unannounced report)

Engineer Research and Development Labs., Fort Belvoir, Va.
ADAPTION OF COMMERCIAL VACUUM-SEAM-TESTER FOR BOLTED STORAGE TANKS, by W. L. Nelson. 1 Sep 55, 28p. incl. illus. (Rept. no. 1424) Unclassified report

"This report covers the work done to adapt a commercial vacuum-seam-tester (normally used in locating leaks in the bottom of welded storage-tanks) for use in locating leaks in the bottom of bolted storage-tanks. The end result of this investigation was the preparation of drawings and specifications of a vacuum-seam-tester for bolted storage-tanks. The drawings appear in this report as Appendix B. The specifications are in preparation. Two commercial seam-testers were tested; both identical except that the vacuum pump of one is powered by an electric motor and that of the other by a gasoline engine. All the component parts of the com-
(over)

AD-113 178

mercial model proved satisfactory except the inspection-box, the edges of which would not fit between the protruding bolts on the tank bottom. An inspection-box having dimensions that allowed the edge of the box to fit between the tank bolts was then designed and fabricated at ERDL. Service tests made on this design showed that the seam-tester will satisfactorily locate seam leaks in the bottom of bolted tanks." (Extracted from report)

AD-113 822 Accession No.

Material Lab., New York Naval Shipyard, Brooklyn.
QUALIFICATION OF SYNTHETIC RUBBER CALKING COMPOUND NO. 3C-1200 AND PRIMER NO. 3C-1200 SUBMITTED BY CHURCHILL CHEMICAL CORPORATION, LOS ANGELES, CALIFORNIA. Final technical rept. 25 June 56 [8]p. incl. tables (Lab. proj. no. 5708-3) Unclassified report

"The 3C-1200 calking compound failed to conform to the requirements of Mil Spec MIL-C-18255 (SHIPS) with respect to: (a) tensile strength (initial) of molded specimens, (b) hardness (aged) of molded specimens, and (c) average extensibility of adhesion specimens after 7 days oven aging." (ML conclusions)

AD-113 830 Accession No.

Material Lab., New York Naval Shipyard, Brooklyn. QUALIFICATION OF SYNTHETIC RUBBER CALKING COMPOUND NO. 14012 AND PRIMER NO. 21011, MANUFACTURED BY ARMSTRONG CORK COMPANY, LANCASTER, PENNSYLVANIA. Supplementary technical rept. 7 Aug 56 [6]p. (Lab. proj. no. 5708-2) Unclassified report

"The No. 14012 calking compound failed to conform to the specifications with respect to: (a) tensile strength (initial) of molded specimens, (b) hardness (initial) of molded specimens, and (c) average extensibility of adhesion specimens after 7 days oven aging at 158°F." (ML conclusions)

AD-116 389 Div. 11/12, 13/1, 14/1, 14/9, 30/3, 30/5

Southern Research Inst., Birmingham, Ala. DESIGN AND CONSTRUCTION OF TEST EQUIPMENT FOR EVALUATING SEALING CHARACTERISTICS OF NON-METALLIC GASKETS, by Jack S. Barr and Sibert Oglesby, Jr. Final progress rept. 5 Nov 56, 27p. incl. illus. tables, 18 refs. (Rept. no. 2905-738-8)
(Contract DA 01-009-ORD-468) Unclassified report

A test facility was sought for measuring the sealing characteristics of gasket materials ranging from cork to compressed asbestos. The tester was to be suitable for testing under such conditions as -65° to 500°F and 3 to 4000 psi. Tester traps the fluid that leaks past a gasket and measures the volume of this fluid. The loading frame for the tester consists of a manually operated jack capable of a 20,000-lb load. Tests were made on 10 representative gasket materials at room temperature; one sample was tested at 215°F against nitrogen, and another was tested at room temperature against a (over)

AD-116 389

silicone base oil. The range of characteristics of the materials tested is demonstrated by the shapes of the leakage rate curves. The harder materials show leakage of gas at very low locked-up pressures. The softer materials do not show appreciable leakage until much higher locked-up pressures are reached. An apparent discontinuity in the curves plotted with nitrogen as the test fluid indicates that no leakage of gas occurs up to a clearly defined pressure. Once this pressure is exceeded, the leakage rate increases at a rapid rate. The apparent critical pressure can be used as an index of the sealibility of the gasket material. The apparatus provided gives a sensitive and reproducible method for measuring leakage of both gases and liquids past a gasket. The sensitivity obtained is about 0.02 ml/min of fluid past the gasket with a 1-min sampling time.

AD-116 938 Div. 4/1, 14/9

Atomic Energy Research Establishment (Gt. Brit.). HIGH VACUUM SHAFT SEALS, FLANGED JOINTS AND THE GASSING AND THE PERMEABILITY OF RUBBER-LIKE MATERIALS, by R. H. V. M. Dawton. July 56 [39]p. incl. illus. tables (AERE rept. no. GP/R 1875; HD-2318) Unclassified report

High-vacuum measurements are presented on: (1) vacuum shaft seals at 20°C and 120°C; (2) the gassing and the permeability of rubber and rubber-like substances; and (3) the effect of flanged joints (including a silastic gasket joint operating at 300°C) in a high vacuum system. (Author)

AD-117 021 Div. 8/4, 26/1

Raytheon Mfg. Co., Waltham, Mass. CERAMIC-METAL SEAL OUTGASSING STUDY, by R. J. Dyer and M. Weiss. Final rept. 1 Apr-31 Dec 56. 10p. illus. tables (Rept. no. QK608; AFCRC TR-57-162)
(Contract AF 19(604)1826) Unclassified report

The total amount of each gas evolved and individual gas evolution rates were determined for test assemblies brazed with the following solders: Ag-Cu eutectic, 80% Au-20% Cu, Ag, and pure Cu. Attempts to braze with Incosil-15 (Ag-Cu-In alloy) solder resulted in leaky seals. Evolved gases were analyzed with a mass spectrometer. The test assemblies were baked out for about 6 hr, during which time the gas evolved was collected and analyzed. The gases evolved were H, CO, water vapor, and CO_2. The maximum rate of H evolution ranged from 2.3×10^3 to 1.1×10^4 cc μ/min at an average temperature of 400°C; the rate decreased to an average of 31 cc μ/min at the end of the 6-hr heating (over)

AD-117 021

period. The average maximum rate of CO was 70 cc μ/min at temperatures ranging from 651° to 702°C; the rate decreased to an average of 36 cc μ/min at the end of bakeout period. Water-vapor evolution decreased to a negligible value at the end of about 3 hr of bakeout at about 650°C. The CO_2 average maximum rate of evolution reached about 46 cc μ/min at temperatures ranging from 256° to 495°C. The CO_2 evolution rate decreased to a negligible value after about 155 min of bakeout from 651° to 704°C. (See also AD-132 254)

AD-118 047 14/1, 2, 8 17/2 30/3, 5

Horizons, Inc., Cleveland, Ohio
DEVELOPMENT OF FRICTION-SEAL MATERIALS FOR HIGH TEMPERATURE-HIGH SPEED OPERATION, by Peter F. Mataich. Rept. for July 55-July 56, on Metallic Materials. Feb 57, 18p. incl. illus. tables (WADC Technical rept. no. 56-579)
(Contract AF 33 (616)3134) Unclassified report

A study was made in an attempt to find materials suitable for use as friction seals under conditions of high temperature and high rubbing speeds. A special test rig for screening the materials under simulated operating conditions was designed and constructed. It was found that the group of materials evaluated in the present program showed excessive wear under the severe operating conditions required. This excessive wear was attributed to the excessive temperatures reached at the mating surfaces when the lubricating oil film broke down, and it is suggested that this condition could be remedied by incorporating a [illegible]able (over)

AD-118 047

solid lubricant in the seal material itself. (Author)

AD-118 213 Div. 14/1, 14/8

Hydra-Power Corp., New Rochelle, N. Y. INVESTIGATION OF HYDRAULIC LOW FRICTION TEFLON CAP SEALS, by John O. Bruno. Rept. on Aircraft Hydraulic Systems. Dec 57, 56p. incl. illus. tables (WADC Technical rept. no. 57-163)
(Contract AF 33(600)28637) Unclassified report

Test apparatus was designed and built, and a series of tests were conducted, to provide design data covering a low friction rod seal composed of a standard AN6227 O ring seal and a Teflon cap which fits around the inside periphery of the O ring. Basic requirements included the use of Mil-O-5606 hydraulic oil, a 1-in. rod (0.997 ± 0.001), a rod stroke of 4 in. minimum, and impulse pressures of zero to 3000 psi applied while the rod was cycling at 30 ± 5 cycles/min; consideration was given to temperatures in excess of 160°F. During the tests, the dimensions of the Teflon cap were varied as well as the surface finish, packing squeeze, side loads, (over)

AD-118 213

and extrusion gap. The results of tests for the 0.031-in. Teflon cap indicate that (1) the thicker section of the cap facilitates manufacture and installation, (2) the leakage characteristics of the seal are unaffected by O-ring squeeze (0.015 to 0.028 in.), (3) the surface finish of the Teflon cap and rod may be as high as 18 rms, but the better finish should lower the break-out friction, (4) the Teflon cap must be about 0.007 in. smaller in inside diameter than the rod, (5) a 0.003-in. radial extrusion gap may be used, (6) a 55 Rockwell C hardness for the rod is suggested, (7) nominal side loads from about 2° misalignment does not affect leakage, (8) no leakage change because of temperature was indicated, (9) the impulse breakdown of the O ring as a static seal seems to limit the life of the O ring and cap, and (10) leakage with rod travel in the same direction as applied pressure was below the 1 cc/1000 cycles allowable.

AD-118 597 Accession No.

[Material Lab.] New York Naval Shipyard, Brooklyn. VARIABLE PITCH PROPELLER SEALS; CHEMICAL ANALYSIS OF. Final rept. 21 Nov 56, 3p. (Lab. proj. no. 5901-1) Unclassified report

AD-119 330 Div. 11/2, 14/1, 14/6, 14/9, 26/6, 27/1

Material Lab., New York Naval Shipyard, Brooklyn.
O-RINGS USED TO SEAL CYLINDER LINERS OF DIESEL ENGINES, by S. A. Eller and A. A. Stein. Final technical rept. 12 Dec 56, 1v. incl. illus. tables. (Lab. proj. no. 5354-9) Unclassified report

Available to U. S. Military Organizations only

An investigation was conducted to determine the suitability of Neoprene, Buna N and silicone O-ring gaskets for use in cylinder liners of diesel engines. Ten O-ring gaskets obtained from commercial sources and from stock were subjected to simulated service conditions. Following each test condition the back load, which is a measure of the sealability of the O-ring, samples, was measured with the Material Laboratory compression stress relaxation apparatus. In addition, the overall height and hardness of the samples were measured. The percent load relaxation of the samples after aging in a Geer oven for 2 weeks at 194±2F and after immersion (over)

AD-119 330

in distilled water for 2 weeks at 194±2F while in a compressed condition were calculated. Based on these results recommendations were made regarding requirements for inclusion in a specification for this type of O-ring. (Author)

AD-119 437 (Unannounced report) Div. 27/2

National Advisory Committee for Aeronautics, Washington, D. C.
A BALANCED-PRESSURE SLIDING SEAL FOR TRANSFER OF PRESSURIZED AIR BETWEEN STATIONARY AND ROTATING PARTS, by Arthur N. Curren and Reeves P. Cochran. 9 Jan 57, 17p. incl. illus. (Research memo. RM E56I11)
Unclassified report

Available to U. S. Military Organizations. Others direct requests to NACA, Wash., D. C.

A balanced-pressure sliding seal was developed to satisfy the need for a seal capable of transferring high-pressure air from stationary to rotating parts with a minimum of leakage. This seal was designed in connection with experimental investigations of air-cooled turbine rotor blades in turbojet engines where high sliding velocities and high ambient seal operating temperatures are encountered. The seal consists of two (over)

AD-119 437

stationary concentric carbon-base seal rings sliding on a rotating chromium-plated disk. Blade cooling air passes through the center of the rings and the disk, and pressure-balancing air is introduced into the annular space between the two rings. The seal was investigated experimentally to determine the amount of cooling-air leakage, the quantity of pressure-balancing air required, the temperatures of the sliding surfaces, and the wear of various seal parts. At sliding velocities up to 10,000 feet per minute and cooling-air pressures up to 38.3 pounds per square inch absolute, no leakage of cooling air occurred. The quantity of balance air required to operate the seal was about one-fourth to one-eighth the quantity of cooling air lost in leakage with the labyrinth seals used in this application under the same operating conditions. External cooling of the sliding surfaces was required to keep the temperatures below the oxidizing temperatures of carbon-base seal rings. The rate of wear on the seal rings was unmeasurable on the inner ring and about 0.0005 inch per hour on the outer ring. (Author)

AD-119 763 Div. 22/1, 26/1

Sciaky Bros., Inc., Los Angeles, Calif.
WELDING DEVELOPMENT PROGRAM FOR MUNITIONS CLOSURES, by S. Robelotto. Progress rept. no. 14, 1-30 Nov 56. 1 Dec 56, 29p. incl. illus. tables [CRL rept. no. ETF 420.6-2/13]
(Contract DA 18-108-cml-5860) Unclassified report

Techniques for the welding of thin-wall munitions seals were investigated. A total of 330 specimens, representing various combinations of munition blanks and closure caps were welded. The welding equipment consisted of a RMCO 5STK Spotwelder (220,000 amp maximum secondary current and 18,500 lb maximum welding pressure), a RMCO 2STK Spotwelder, and a RMCO 2ST Spotwelder (70,000 amp maximum secondary current and 4500 lb maximum welding pressure). Methods of reducing weld phase shifts and the verification of tentative weld schedules were studied, and the effectiveness of various electrode face diameters on specific specimen combinations were investigated. These weld- (over)

AD-119 763

ing schedules resulted in leaktight specimens of good weld strength. (See also AD-117 992)

AD-120 620 Accession No.

American Lava Corp., Chattanooga, Tenn.
A STUDY AND EVALUATION OF METHODS OF PRODUCING A CERAMIC-TO-METAL SEAL BY PRESSED POWDER TECHNIQUES FOR AUTOMATIC MASS PRODUCTION, by Harry C. Dunegan. Final rept. 16 Aug-30 Nov 56. 82p. incl. illus. tables, 13 refs.
(Contract AF 33(600)27329) Unclassified report

"A method was developed for producing vacuum tight ceramic to metal seals by means of a pressed powder technique. In this method, layers of ceramic and of metal powders were fabricated in one pressing operation. A single firing cycle matured the ceramic, sintered the metal and developed the bond. The difficulties in overcoming fabricating and firing problems, in selecting compatible material systems and in determining optimum bonding conditions were described. The results of over ninety-different metal, alloy and metal oxide mixtures in combination with approximately thirty ceramic formulations were listed (over)

AD-120 620

Phase changes in the metallic layers close to the interfaces were used to explain certain types of bonding." (Contractor's abstract) (See also AD-87 677)

Uniterms (cont'd)

Thermal-expansion
Polishing
Rupture

AD-120 853 Div. 8/2

Ontario Research Foundation (Canada).
INVESTIGATION INTO THE SEALING OF QUARTZ CRYSTAL HOLDERS TYPE HC-6/U, by B. W. Schumacher. Quarterly progress rept. no. 1 on phase 2, 1 Apr-31 July 56. 28 Sep 56, 6p. incl. table.
Unclassified report

A study is underway to find a reliable, solder-sealed metal enclosure which can withstand exposure to low temperature without cracking of the seal. Tests were conducted to determine the significance of internal stresses and surface irregularities on HC-6/U nickel-silver crystal holder cans. Mercurous nitrate tests showed no cracks, even by force, for 12 cans of pre-tinned edge. For 12 untinned cans, no spontaneous cracks were observed; 1 can was cracked by force. For comparison, a group of older CCCN units (AD-108 720) were desoldered and tested. All (8) cans of the formerly leaking units cracked spontaneously. For 20 cans of the formerly tight units, 9 showed no cracks, 7 (over)

AD-120 853

showed 1 crack only, and 4 showed from 2 to 4 cracks. Six cans of the new group were subjected to 12 cold cycles on dry-ice slabs. A mercurous nitrate test showed no cracks for 5 cans and a very fine crack on the sixth. X-ray back-reflection pictures were used to establish the following relationships for the stresses in a can in an effort to find a nondestructive method for determining stress. If cracks occur in the mercurous nitrate test, the diffraction rings of Cu Kα radiation are not resolved into the Kα1 and Kα2 doublet. If no cracks occur, at least the first doublet is resolved (second doublet or second ring on the picture not resolved). Results are tabulated for 58 units (37 tight and 21 leaking) which were X-ray tested for stresses. No correlation was found between the stresses and the percentage of leaking cans.

AD-121 112 Div. 11/2, 11/5, 26/6

Aberdeen Proving Ground, Md.
BENDIX SEALED BRAKES ON TRUCK, UTILITY, 1/4 TON, 4X4, M38, by Paul Laichinger, Jr. Partial rept. no. 1. Nov 56, 1v. incl. illus. (Proj. no. TT3-T32, rept. no. 8) Unclassified report

One design of lip seals and six designs of face seals for brakes were tested by installing them on a Truck, Utility, 1/4 Ton, 4x4, M38 and driving the truck in mud. Each seal design failed. After each failure, the manufacturer developed another design which, in turn, was tested. This report discusses six different tests. None of the seals tested were capable of satisfactorily keeping mud out of the brakes. The seals failed because of the inability of the seal stator to follow eccentricities of the rotor, wiping of mud into the seal by eccentric motion, jamming of the seal mechanism by mud, mechanical failure of the seal mechanism, and/or high wear rate of the seal face material. (Author)

AD-121 294 Div. 22/12

Aberdeen Proving Ground, Md.
DOUBLE LIP SEAL FOR COMPENSATING IDLER, by L. G. Bach. Final rept. Partial rept. no. 10 on Production Engineering of Components for Tank, 90mm Gun, M48. Jan 57 [17]p. incl. illus. (Proj. no. 251-037-03, rept. no. 1) Unclassified report

Available to U. S. Military Organizations. Others direct requests to Office, Chief of Ordnance, Wash., D. C., Attn: ORDTX-AR

Twelve double lip seal installations were tested to determine seal performance by using light oil (OE 10) and operating on paved, gravel, and cross-country courses. Five M48 tanks were used. Eight seals were used for 2000 miles with small additions of oil; the remainder failed at points between 620 and 1847 mi. Seal deficiencies included cutting and folding under of the outer lip and cutting of the inner lip by the spindle. The double lip seal is superior to the standard single (over)

AD-121 294

lip seal but performance is dependent on a complex installation procedure. The felt ring used in the standard seal installation provides an effective barrier against foreign materials when used in conjunction with a double lip seal. (Author)

AD-121 339 Div. 14/1, 14/9, 30/3

Laboratories for Research and Development, Franklin Inst., Philadelphia, Pa.
GASKET SEALS, by C. Ross Bloomquist. Final rept. 1 July 55-30 Sep 56. 25p. incl. illus. tables (Rept. no. F-A-1858)
(Contract DA 36-034-ORD-2083, Proj. TR 5-5059T) Unclassified report

The sealing efficiency of Buna N and cork-rubber rectangular gaskets sealed in aluminum cells with an air pressure difference of 15 psi was determined. Correlations were attempted between sealing efficiency, gasket materials and properties, and seal design parameters. It was found that the ambient temperature, type of gasket material and its compression-set characteristics probably establish the level of sealing efficiency. Generally, seals made with rectangular Buna N gaskets leaked less than comparable ones made with rectangular cork-rubber gaskets. A limited number of leak rate measurements made on rectangular Thiokol gaskets (over)

AD-121 339

show, superficially, better sealing characteristics than similar Buna N gaskets; however, this performance is believed due to creep of the Thiokol while under compression. Pyramid-shaped Buna N gaskets were found to seal at least as well as rectangular Buna N gaskets at room temperature. (Author)

AD-121 461 Div. 4/4, 20/5
(Unannounced report)

Atomics International, Canoga Park, Calif.
THE BEHAVIOR OF TETRALIN IN LIQUID SODIUM, by R. L. McKisson and K. E. Horton. 1 Feb 57, 19p. incl. illus. tables (NAA-SR-1771)
(Contract AT (04-3)49) Unclassified report

A simplified, noncquilibrium experiment designed to qualitatively study the behavior of tetralin i.e., rate of reaction or decomposition in the presence of sodium, at temperatures up to 950°F has been performed to establish the probable consequences of a tetralin leak in the Sodium Reactor Experiment. It was found that tetralin decomposition with rapidity in the presence of hot liquid sodium. The ultimate decomposition products are hydrogen, carbon, and low molecular weight aliphatic compounds. The hydrogen is rapidly absorbed by the hot sodium up to approximately 750°F. At higher temperatures, sodium tends to release hydrogen. Zirconium metal submerged in sodium irreversibly absorbs hydrogen, at least up to 950°F, so that (over)

AD-121 461

most of the hydrogen resulting from tetralin decomposition in the SRE ultimately would be absorbed in the zirconium moderator can sheaths. The carbon would circulate in the sodium, so that very slight carburization of the stainless might be expected with operation at 950°F. The hydrocarbon residues will collect in the gas space over the sodium pool and eventually decompose. (Author)

AD-121 502 Accession No.

Kidde, Walter, and Co., Inc., Bellville, N. J.
A SUMMATION OF THE PNEUMATIC HIGH TEMPERATURE SEAL EVALUATION PROGRAM, by H. E. Heigis, K. F. Peter, and J. A. Neigel. 12 June 56, 16p. incl. illus. tables. Unclassified report

Information is presented on materials, lubricants, surface requirements, and installation of seals for pneumatic equipment to guide those who are concerned with their selection. Effort was concentrated largely upon static seals which operate in the temperature range of -65° to 225°F.

AD-122 254 Div. 8/4, 14/1, 14/2

Raytheon Mfg. Co., Waltham, Mass.
CERAMIC-METAL SEAL OUTGASSING STUDY, by R. J. Dyer and R. M. Adams. QK 608 Quarterly scientific rept. no. 2, 1 July-1 Oct 56. [6]p. incl. illus. (Rept. no. QK-608-S2; [AFCRC TN-56-965])
(Contract AF 19(604)1826) Unclassified report

A study is being made of the gases which are evolved from Raytheon ceramic-to-metal seal assemblies at temperatures up to 700°C, and the rate at which such gases are produced. The test assemblies consist of a Raytheon cylindrical HV bushing, on the ends of which are applied 1/4-in. metal bands, and Kovar cups. The wall thickness of the top and bottom Kovar cups is 0.030 inch. The top cup is equipped with a Kovar exhaust stem, and a glass tube with a standard 12/30 taper joint is attached to the Kovar stem to facilitate a union with the mass spectrometer manifold. Two assemblies were made with each of the following solders: pure silver, silver-copper eutectic; 80% gold -20% copper al- (over)

AD-122 254

loy, and pure copper. Brazing was accomplished in an atmosphere of pure hydrogen by using a 20-kw RF induction heater. After brazing, the test assemblies were attached to the mass spectrometer manifold. Apeizon W wax was used as a sealing medium. The bakeout oven was raised around the assembly after a pumping time of 1 hr; the pressure in the assembly was about 2×10^{-5} mm Hg. Four assemblies were baked out and the evolved gases were analyzed. Curves are given which show the relationship between bakeout time, temperature, and individual gas evolution rates for a test assembly brazed with pure silver. (See also AD-109 660)

AD-123 436 Div. 8/2

Defence Research Telecommunications Establishment (Canada).
PACKAGING A 30 Mc AMPLIFIER FOR AIRBORNE USE, by G. W. Williams. July 56, 19p. incl. illus. (EL rept. no. 5046-1; [Encl. 3 to Air Attache, Ottawa, rept. no. TL 21-57]) Unclassified report

Two 30 Mc amplifiers have been packaged for airborne use in the form of sealed plug-in units. These utilize small components and subminiature tubes to achieve a gain of 50 to 60 db in units which measure 3 in. x 3 in. x 1 1/2 in. or smaller. Foamed and solid thermosetting plastics are used to protect the circuit against environmental stresses. The tubes are replaceable by removing sealed tube shields. The completed units have been subjected to environmental testing. (Author)

AD-123 604 Div. 14
(Revised 6-59)
Directorate of Materials and Explosives Research and Development (Gt. Brit.).
NOTES ON A DEMONSTRATION OF R.D. 1286, A THREAD-SEALING CEMENT, by R. W. Bryant, W. A. Dukes, and J. Gooding. Dec 56 [19]p. incl. illus. tables (Technical note no. DMXRD/SEAL/56/4) Unclassified report

AD-123 771 Div. 10/2, 14/1

Naval Engineering Experiment Station, Annapolis, Md.
CHEMISEAL MECHANICAL SEAL FOR GASOLINE PUMP SERVICE SUBMITTED BY UNITED STATES GASKET COMPANY CAMDEN, NEW JERSEY, by R. F. Stone. (Qualification rept. 15 Aug 56, 4p. incl. illus. tables (EES rept. no. 010249) Unclassified report

Available to U. S. Military Organizations only

Two types of Chemiseal mechanical seals, manufactured by United States Gasket Company, were tested to determine their suitability for use in gasoline pumps operating at stuffing-box pressures between 5' and 25' of aviation gasoline. One type with graphite-filled-teflon rotary-seal face was found suitable for this service. The type with glass-filled teflon rotary seal face prevented leakage but caused scoring of the stationary seal ring. (Author)

AD-123 839 Div. 14/9, 28/8

Material Lab., New York Naval Shipyard, Brooklyn.
QUALIFICATION TEST OF SYNTHETIC RUBBER O-RINGS, SUBMITTED THE INTERNATIONAL PACKING CORPORATION, BRISTOL, N. H. Final technical rept. 13 Feb 57, 4p. (Lab proj. no. 5903-1) Unclassified report

Available to U. S. Military Organizations only

Minimum processing - no abstract

AD-123 912 Div. 14/8, 14/9, 27/1

Material Lab., New York Shipyard, Brooklyn.
INVESTIGATION OF O-RING CYLINDER LINER AND WATER FERRULE SEALS REMOVED FROM DIESEL ENGINES OF THE USS PC 560, by S. A. Eller and M. Rosenberg. Final technical rept. 9 Oct 56 [19]p. incl. illus. tables, 12 refs. (Lab. proj. no. 5354-10)
Unclassified report

Available to U. S. Military Organizations only

O-rings used for cylinder liners and water ferrule seals that had been subjected to water soaking while installed in diesel engines of the USS PC-560 were removed from the engines and forwarded to the Material Laboratory for investigation. The samples were examined in the condition "as received" by the Laboratory, then the basic elastomeric ingredient in the compound was determined by chemical analysis. Following this tests were made to determine the effects of immersing the seals in distilled water for 10 days at 194±2F on volume
(over)

AD-123 912

swell, tensile strength, elongation, back load, height, and hardness of the material. The results indicated that immersing the seals in water had a beneficial effect on back load and height of the gasket material, but a detrimental effect on the tensile strength, elongation, and hardness. (Author)

AD-125 427 Div. 14/9, 22/1
(Unannounced report)

Rock Island Arsenal Lab., Ill.
AMMUNITION CAN GASKETS, by R. F. Shaw. Technical rept. no. 14 on General Development of Rubber for Ordnance Applications. 13 Apr 49 [16]p. incl. illus. tables (Proj. no. TB4-530A, RIA Lab., rept. no. 49-386)
Unclassified report

The object of this work was to evaluate a commercial gasket material; to develop suitable ammunition can gaskets from several polymers; to determine the physical requirements of a satisfactory gasket material; and to revise the procurement specification to conform to specification 20-116A for rubber and synthetic rubber compounds: automotive and aeronautical applications. The Rudy gasket and 5 RIA formulations were tested for physical properties and for their performance as gaskets in ammunition boxes. Polymers evaluated were Perbunan 18, a butadiene/acrylonitrile (82/18) copolymer, Philprene A, which is a butadiene/styrene copoly-
(over)

AD-125 427

mer (76/24), GR-S 454, a butadiene/styrene (90/10) copolymer, Neoprene GRM-10, and natural rubber. General purpose GR-S copolymer was suitable for the intended purpose. The Rudy gasket did not pass the low temperature flexibility test after exposure for 5 hr at -70°F, but held 10 lb air pressure for 24 hr at that temperature. All the gasket compounds tested met the water resistance and aging requirements; all held 10 lb air pressure for 24 hr at -70°F. Gasket hardness was not critical between the limits 50-62 Shore A durometer. Compression set requirements should probably be 25% maximum, consistent with good commercial practice for compression gaskets.

AD-125 997 Div. 1, 27

Reaction Motors, Inc., Denville, N. J.
A ROTARY SHAFT SEAL ASSEMBLY FOR ROCKET ON ROTOR HELICOPTER ENGINES, by Ben Schulkin. (Final rept.) 31 Mar 56, 19p. incl. illus. tables. (Rept. no. 468-F3)
(Contract NOa (s)-53-1040-c) Unclassified report

AD-127 073 Div. 14, 22

Borden Co., Philadelphia, Pa.
CRIMP AND THREAD SEALANTS, by Wolf Karo, Albin Gapsch, and E. LoPresti. Monthly rept. no. 1. 10 Apr 56, 13p.
(Contract DAI 28-017-501-ORD(P)-2043)
Unclassified report

AD-127 079 Div. 14/1, 14/9, 26/8, 30/2, 30/5

Lincoln Lab., Mass. Inst. of Tech., Lexington
UNSUPPORTED AREA HIGH PRESSURE SEAL, by D. M. Warschauer and William Paul. 5 Oct 56, [1]p. incl. illus. (In cooperation with Harvard U.)
(Contract AF 19(122)458) Unclassified report

AD-127 094 Div. 14, 26
(Unannounced report)

Oak Ridge National Lab., Tenn.
O-RING MOUNTING OF BELLOWS, by M. L. Randolph, M. Slater, and D. L. Parrish. 23 July 56. 1p. incl. illus.
(Contract W7405-eng-26) Unclassified report

Reprint from Review of Scientific Instruments 27:878-878, Oct 56.

AD-128 326 Div. 14/9
(Unannounced report)

Rock Island Arsenal Lab., Ill.
DEVELOPMENT OF AN OIL-RESISTANT SYNTHETIC RUBBER COMPOUND FOR LOW TEMPERATURE APPLICATIONS, by Robert F. Shaw. [Technical] rept. no. 9 on General Development of Rubber of Ordnance Applications. 16 June 48 [16]p. incl. tables (Proj. no. TB4-530A, rept. no. 9; RIA lab. rept. no. 48-632)
Unclassified report

The commercial elastomers Thiokol ST, Perbunan 18, and Neoprene FR were the most satisfactory in low-temperature and oil-resistance properties. Perbunan 18 had the best low-temperature properties when compound with various plasticizers and was used to develop compounds suitable for packings in petroleum fluids. Perbunan 18 base stock was compounded with 12 different plasticizers selected for improvement of low temperature properties and insolubility in petroleum oils. The physical properties of these compounds are tabu-
(over)

AD-128 326

lated and evaluated for suitability. Packings made from compound N6-12 which ranked first in low temperature flexibility, tensile strength, and tear resistance performed best in ordnance mechanisms at -75°F in AN-W-O-366b hydraulic fluid. Tables of physical properties of vulcanizates and the effects of increased plasticizer content are included. (Author) (See also AD-124 836)

AD-128 368 Div. 27/1

Air Technical Intelligence Center, Wright-Patterson Air Force Base, Ohio.
SPARK PLUG WITH GASKET RING (Zundkerze Mit Dichtungsring) by Robert Bosch. [1956] 3p. incl. illus. (Trans. no. F-TS-10028/V, of Swiss Confederate Patent Office, Patent no. 198232, 1 Sep 1938)
Unclassified report

AD-129 444 Div. 22

Rock Island Arsenal Lab., Ill.
THE EFFECT OF "O" RING GROOVE SURFACE FINISH ON PNEUMATIC EQUILIBRATOR PERFORMANCE, by L. R. Raisbeck. 16 Apr 57, 22p. incl. illus. tables (Technical rept. no. 57-956)
Unclassified report

AD-130 210 Div. 4, 14
(Unannounced report)

Products Research Co., Los Angeles, Calif.
[THE DEVELOPMENT OF SEALANTS WHICH WILL WITHSTAND EXPOSURE TO FUEL VAPOR, LIQUID FUEL AND AIR AT TEMPERATURES ABOVE 350°F AND TO DEVELOP STANDARD TEST PROCEDURES FOR EVALUATION OF THE SEALANTS]. Quarterly progress rept. no. 1, 1 Feb-1 May 57 [33]p. incl. illus.
(Contract AF 33(616)3976) Unclassified report

AD-130 661 Div. 14/1, 22/5

Aerojet-General Corp., Azusa, Calif.
IMPROVEMENT OF INERT SEALER IN BOMBS, by E. T. Leidigh, R. D. McLeod, and J. M. Wrightson. Informal progress rept. no. 11, 10 Mar-9 Apr 57. 17 Apr 57, 3p. incl. table (Rept. no. L1911/12-11)
(Contract DA[I] 28-017-501-ORD(P)-2115)
Unclassified report

The development of the polyurethane foam sealer has been completed. High-explosive loading and temperature cycling of the bombs has been completed. No exudation has been observed from either of the two bombs in which the newly developed polyurethane foam was used in place of the standard inert sealer. On the other hand, approximately 1/4 lb of TNT exuded from the bomb filled with TNT without a sealer, while exudation occurred from both bombs filled with TNT using the standard inert sealer and from the bomb in which KIMPAC was used as a sealer. (Author) (See also AD-120 147)

AD-130 772 Div. 8/2

General Hermetic Sealing Corp., Valley Stream, N. Y.
REUSABLE SEALS FOR ELECTRONIC EQUIPMENT, by William Brown and A. Raxdow. Rept. for 1 June 55-31 Oct 56. May 57, 86p. incl. illus. (WADC Technical rept. no. 56-616)
(Contract AF 33(616)2994) Unclassified report

This report covers the investigation, evaluation and establishment of the most effective type of reusable seals for airborne electronic equipment. Fused seals, methods and procedures were developed for hermetically sealed enclosures of electronic assemblies, subassemblies, and equipment, so they can be opened fifteen times for repair and resealed without destroying the contents there-in. The methods developed are satisfactory for field service application. Temperature measurement techniques were worked out in order to determine the exact heat distribution in and on the seal. As a result of comparative measurements, the best
(over)

AD-130 772

possible seal joint, the most appropriate metal for the cans, and the most efficient heat transfer method was determined. A heat reflecting baffle was developed which decreases the temperature inside the sealed enclosure to below 85°C. The comparative tests on heat transfer methods showed very promising results with the electrical band heater. (Author)

AD-130 882 Div. 1/2, 26/8, 30/3

Stupakoff [Ceramic and Mfg. Co., Latrobe, Pa.]
DEVELOPMENT OF CERAMIC RODS AND SEALS FOR USE IN HIGH-TEMPERATURE HYDRAULIC SYSTEMS, by Guy F. Bernardini. Rept. on Aircraft Hydraulic Systems. July 57, 30p. incl. illus. tables, (WADC Technical rept. no. 57-324)
(Contract AF 33(600)30012) Unclassified report

Design of a ceramic rod seal intended for use in a high temperature, high pressure hydraulic system is described. Details of a test circuit incorporating the test cylinder, which includes a ceramic rod and two ceramic seals, are presented. Fabrication of the test cylinder, ceramic components, and testing apparatus is described. The testing methods and results which were employed are presented and discussed. It is concluded that the ceramic rod seal was not fabricated with sufficient accuracy to prevent leakage at a fluid pressure of 3000 psi and a temperature of 400°F. In addition, it was concluded that even if the seal had been fabricated with a
(over)

AD-130 882

much higher degree of accuracy, wear of the ceramic seal would quickly destroy the sealing relationship. During about 12,500 cycles of rod motion excessive wear of portions of both seals was noted. (Author)

AD- 130 960 Div. 10/5

Power Plant Lab., Wright Air Development Center, Wright-Patterson Air Force Base, Ohio.
PLASTIC FITTING, ACID TANK, by John M. McGarry. July 57 [7]p. incl. illus. (WADC Technical note no. TN 57-279) Unclassified report

The purpose of this report is to record the preliminary results of the testing of a Plastic Kel-F fitting for Fuming Nitric Acid tanks. The fitting assembly utilizes a serrated sealing surface and a Kel-F 300 gasket. It is concluded that a plastic acid fitting is feasible. (Author)

AD-131 091 Div. 1, 14

Coast Pro-Seal and Mfg. Co. [Los Angeles, Calif]
HIGH TEMPERATURE RESISTANT SEALANT MATERIALS, by Leonard C. Boller, Gene M. Le Fave and others. Rept. for Jan-July 56, on Rubber, Plastic and Composite Materials. Sep 57, 18p. incl. illus. tables. (WADC Technical rept. no. 56-155, Part 2)
(Contract AF 33(616)2767) Unclassified report

Thorough formulation studies were made with rubber phenolic fill and drain sealants where polyacrylic, butadiene-acrylonitrile polymers constituted the rubber portion. A sealant, sample no. 35, was developed which represents a moderate variation of E-50 (AD-110 633). In the presence of jet fuel, no. 35 performs excellently at 625° to 650°F, but under dry-heat conditions the sealant will not maintain film integrity beyond 600°F. The optimum film thickness was between 0.005 and 0.008 inch. The solids content was adjusted between 18 and 22% in MeCOEt and/or methyl isobutyl ketone. Cabosil, up to 5%, controls rheolog- proper-
(over)

AD- 131 091

ties with no effect on ultimate film performance. Organofluorine polymers in combination with certain rubbers showed little promise, while liquid polybutadiene offered some interesting possibilities. No consistent or above marginal improvement was obtained with cover coats, antioxidants, plasticizers, crosslinking agents, adhesion promoters, curing agents, and leafing agents.

AD- 131 094 Div. 14, 25, 26

Boeing Airplane Co., Seattle, Wash.
DESIGN DATA FOR O-RINGS AND SIMILAR ELASTIC SEALS, by George E. Trepus. Rept. for May 56-Mar 57, on Rubber, Plastics, and Composite Materials. Sep 57, 106p. incl. illus. tables, 33 refs. (WADC Technical rept. no. 56-272, pt. 2)
(Contract AF 33(616)2862) Unclassified report

See also AD-110 598

AD-134 181 Div. 14/1, 14/9

Material Lab., New York Naval Shipyard, Brooklyn.
OIL RESISTANT SYNTHETIC RUBBER GASKET MATERIAL SUBMITTED BY BUCKEYE RUBBER PRODUCTS, INC., LIMA, OHIO. Final technical qualification test rept. 27 Sep 56, 4p. table (Lab. proj. no. 5542-23) Unclassified report

Available to U. S. Military Organizations only

AD-134 197 Div. 14/1, 14/9

Material Lab., New York Naval Shipyard, Brooklyn.
QUALIFICATION TEST OF SYNTHETIC RUBBER O-RINGS SUBMITTED BY NATIONAL SEAL DIVISION, FEDERAL-MOGUL-BOWER BEARINGS, INCORPORATED, NEWARK, NEW JERSEY. 26 Apr 57, 6p. incl. tables (Lab Proj. no. 5902-4)
Unclassified report

Available to U. S. Military Organizations only

AD-134 405 Div. 8/3

Material Lab., New York Naval Shipyard, Brooklyn. QUARTZ CRYSTAL UNIT CORROSION, by A. N. Silverstein and C. Milner. Final technical research and development rept. 26 Apr 57 [16]p. incl. illus. tables (Lab. proj. no. 4960-F-40, 3-150) Unclassified report

Available to U. S. Military Organizations only

This project was undertaken to investigate suitable methods of protecting quartz crystal units from the effects of corrosion including means for improving the seal of these units. Seven hundred crystal units, whose seals have been coated with either an extra thickness of solder or one of ten corrosion resistant lacquers, were manufactured for use in this investigation. The effectiveness of the lacquers and heavier soldered seals is illustrated by the small number of frequency failures, 2 out of 428, resulting from the standard 50 hr salt spray test. It was also seen that 46 (over)

AD-134 405

of 48 units could withstand a 300 hr salt spray exposure without frequency failure. Prevention of excessive corrosion is desirable but not sufficient in itself. A good seal must withstand aging and conditioning tests as well as salt spray exposure. Lacquers C, H, J and K have best met this requirement. A chemical analysis of the lacquers showed that one of two resins was used in each lacquer. No relationship could be established between the resins used and the test results obtained. The area covered and affinity of the coating for the surface were found to be important factors in determining the effectiveness of these protective coatings. The heavier soldered seals were generally not as effective as the lacquered ones. Therefore, increasing the thickness of the solder seal is not considered to be the best method for improving the seal of quartz crystal units. (Author)

AD-134 462 Div. 10/2, 14/5, 14/9

Material Lab., New York Naval Shipyard, Brooklyn. QUALIFICATION TEST OF SYNTHETIC RUBBER DRUM GASKETS SUBMITTED BY HAMILTON RUBBER MANUFACTURING CORPORATION, TRENTON, NEW JERSEY. Final technical rept. 5 Apr 57 [6]p. incl. table (Lab. proj. no. 5004-36) Unclassified report

Available to U. S. Military Organizations only

Results of tests indicate that sample gaskets, Type VI, in both the 3/4- and 2-in. sizes, manufactured from the exhibitor's compound No. XH-63-D are in conformity with all the requirements of Spec MIL-G-432A of 30 Sept. 1952. (Author)

AD-134 463 Div. 10/2, 14/5, 14/9

Material Lab., New York Naval Shipyard, Brooklyn. TYPE APPROVAL EVALUATION OF SYNTHETIC RUBBER DRUM GASKETS MANUFACTURED BY AUBURN RUBBER CORPORATION, AUBURN, INDIANA, SUBMITTED BY IDEKE METAL PRODUCTS CORPORATION, AUBURN, INDIANA. Final technical rept. 30 Jan 57 [6]p. incl. table (Lab. proj. no. 5004-35) Unclassified report

Available to U. S. Military Organizations only

The exhibitor's Type II sample gaskets, in the 3/4 in. and 2 in. sizes, are not in conformity with MIL-G-432A (30 Sep 1952) with respect to the dimensional requirements for the inside diameter. The sample gaskets are in conformity with all the remaining specification requirements when evaluated in conjunction with the improved closures. (Author)

AD-134 493 Div. 27/1

[Material Lab.] New York Naval Shipyard, Brooklyn O-RINGS USED TO SEAL CYLINDER LINERS OF DIESEL ENGINES-INVESTIGATION. 29 Jan 57, 3p. tables (Proj. no. 5354-9). Unclassified report

Available to U. S. Military Organizations only

AD-134 497 Div. 14/8, 31/5

Material Lab., New York Naval Shipyard, Brooklyn. EVALUATION OF PLASTIC STUFFING TUBES FOR BULKHEAD PENETRATION. Final technical rept. 11 Apr 57, 12p. illus. (Lab. proj. no. 5449-2) Unclassified report

Available to U. S. Military Organizations only

AD-134 918 Div. 8/1, 8/2, 14/1

New York U. Coll. of Engineering, N. Y. SEALING TECHNIQUES FOR WAVEGUIDE AND COAXIAL ROTARY JOINTS, by Milton M. Sackman, Michael D. Maccarone, and Fred H. Posner. Quarterly progress rept. no. 9, 1 Sep-31 Nov 56. 49p. incl. illus. table. (Contract DA 36-039-sc-64476) Unclassified report

A basic design criteria is being determined for pressurizing, waveguide and coaxial rotating joints applying a mechanical sealing principle. The equipment for conducting the various tests on the development model of the RG-52 rotary sealed joint was readied. The tests consist of low- and high-speed tests, and pressurized and environmental conditions. The low-speed (1.5 rpm) tests will be conducted with a combination of a small drill press and a lathe; a regular drill press will be used for the high-speed tests. An environmental test chamber was modified to meet the high and low temperature conditions. Insulation was added to enable the (over)

AD-134 918

-65°F to be obtained; the temperature was obtained by cooling the chamber with dry ice. The final design and materials were established. The geometry of the internal parts gives the best choke design, and the lowest torque, temperature, and leakage are obtained; there is a seal at 0 to 60 psi. The carbon compound will be Karbate 22, and the seal piece surface will be type 303 stainless steel. The remainder of the unit will be 61-ST-6 Al, except for bearings, screws, and snap rings. (See also AD-106 777)

AD-135 272 Div. 4/4, 14/1
(Unannounced report)

Naval Research Lab., Washington, D. C. A STUDY OF SEALING COMPOUNDS FOR INTEGRAL FUEL TANKS, by A. T. DiMasi. 18 Sep 45, 1v. incl. tables (NRL rept. no. P-2623) Unclassified report

The effect of a series of organic and inorganic reagents on the fuel resistance, application, stability and adhesiveness of butadiene-acrylonitrile, butadiene-styrene and polysulfide polymers was studied and is reported herein. These materials have been prepared and examined to determine their possible use as sealing compounds for integral fuel tanks. A few results are given as to the practicability of the same materials for use as rubber adhesives. (Author)

AD-136 762 Div. 14/9
(Unannounced report)

Material Lab., New York Naval Shipyard, Brooklyn. EVALUATION OF SILICONE GASKET MATERIAL NO. DSRE-1584 SUBMITTED BY GENERAL ELECTRIC COMPANY. Final rept. 19 Nov 51, 5p. (Lab. proj. no. 4598-225) Unclassified report

Available only by request to BuShips, Wash., D. C.

AD-136 780 Div. 14/8
(Unannounced report)

Material Lab., New York Naval Shipyard, Brooklyn. EXPERIMENTAL COMPOUND S-50, R9-138 SUBMITTED FOR EVALUATION BY ACADIA SYNTHETIC PRODS. DIV. WESTERN FELT WORKS, CHICAGO, ILL. Final rept. 13 June 51 [2]p. (Lab. Proj. no. 4598-209) Unclassified report

Available to U. S. Military Organizations only

AD-136 781 Div. 14/9
(Unannounced report)

Material Lab., New York Naval Shipyard, Brooklyn. LOW TEMPERATURE GASKET COMPOUNDS, EVALUATION OF MATERIAL SUBMITTED BY GENERAL TIRE AND RUBBER CO., WABASH, INDIANA. [Final rept.] 9 Apr 52, 2p. incl. table (Lab. proj. no. 4598-229) Unclassified report

Available to U. S. Military Organizations only

[illegible] 420 Div. 7/1

[illegible] Corp., [illegible], N. Y.
[illegible] LEADING TO THE DEVELOPMENT OF A NICKEL CADMIUM BATTERY HAVING A HERMETICALLY SEALED CONSTRUCTION, by Irwin M. Schulman. Quarterly rept. no. 3, 11 Feb-10 May 57, 41p. incl. illus. table.
(Contract DA 36-039-sc-73009) Unclassified report

Three cell batteries were subjected to automatic cycle service (60 minute charge - 40 minute discharge). Those of 0.45 A.H. capacity (cylindrical cells) were charged at a modified constant potential of 4.5v., and discharged at several rates between .07 and .45 ampere. The 0.1 A.H. batteries (disc cells) were charged at a modified constant potential of 1.35v, and discharged at three rates between .05 and .07 ampere. 0.45 A.H. cells were also cycled under 1 mm Hg vacuum. These cells were charged at a modified constant potential of 1.50v, and discharged at several rates from 0.07 to 0.45 ampere. Data are reported for end of discharge voltages for all batteries being cycled. An
(over)

AD-137 420

evaluation was made of "D" size cylindrical cells at 80°, 0°, and -40°F at two discharge rates (0.4 and 2.0 amperes). Characteristic discharge curves are shown for the various rates at the three temperatures. (Author)

AD-140 116 Div. 6/1, 14/1

Electrical Testing Lab., Portsmouth [Naval Shipyard], N. H.
METHOD OF BLANKING SUBMARINE SONAR STUFFING TUBES, by M. W. Varrell. 15 May 57, [3]p. incl. illus. (Evaluation rept. no. ETL-2541)
Unclassified report

Available to U.S. Military Organizations only

An evaluation was made of the performance of methods of blanking unused sonar stuffing tubes to seal-off entrance of water up to and including pressures of 1000 psi. Three of each of the following items were used in the tests: (1) MS-18691, type 1 plug bodies for sizes A, B, C, D, and E tubes; (2) type 5 stuffing tubes, sizes A, B, C, D, and E with gland nuts, gland rings and 73902-5 packing rings; (3) rubber O rings of proper dimensions to fit the tube; (4) ring packings 73902-5 of proper dimensions to fit the tubes; and (5) steel plates of proper size to fit porthole openings in hydrostatic
(over)

AD-140 116

pressure tank. The type 1 plug bodies satisfactorily sealed the stuffing tubes, employing either the O rings or the 73902-5 packing rings on the seat against leakage up to and including 1000 psi hydrostatic pressure.

AD-140 117 Div. 6/1, 14/1

Electrical Testing Lab., Portsmouth Naval Shipyard, N. H.
SUITABILITY TESTS ON COMMUNITY STUFFING TUBE ASSEMBLIES AND NEOPRENE PACKING IN STEEL STUFFING TUBES, 5 Mar 57 [9]p. incl. illus. table (Suppl. to Evaluation rept. no. ETL-2564, AD-140 118)
Unclassified report

Available to U. S. Military Organizations only

A report is presented of the results of supplementary testing of neoprene packing in steel bulkhead stuffing tubes. The neoprene packing did not prove entirely satisfactory as an effective seal.

AD-140 118 Div. 6/1, 14/1

Electrical Testing Lab., Portsmouth [Naval Shipyard] N. H.
SUITABILITY TESTS ON COMMUNITY STUFFING TUBE ASSEMBLIES AND NEOPRENE PACKING IN STEEL STUFFING TUBES. 8 Apr 57 [6]p. incl. illus. (Evaluation rept. no. ETL-2564(Final))
Unclassified report

Available to U. S. Military Organizations only

An evaluation of the effectiveness of neoprene packing in steel bulkhead stuffing tubes showed that neoprene packing is not entirely satisfactory as an effective seal with rubber covered cables. (See also AD-140 117)

AD-140 125 Div. 7/3, 14/9

Material Testing Lab., Portsmouth Naval Shipyard, N. H.
METHODS OF APPLYING END SEALS TO NEOPRENE-JACKETED CABLE, by Jeremiah F. Sullivan. 1 Oct 56, 1v. incl. illus. tables (Evaluation rept. no. E56-30F)
Unclassified report

Available to U. S. Military Organizations only

Four methods of forming pressure proof end-sealed packings on Neoprene-sheathed cables were tested to determine which was the cheapest and most satisfactory. These methods were: (1) Method of molding end seals prescribed by Drawing 9000-S6202-F-73903. (2) Method using a straight Neoprene compound outlined in a Beneficial Suggestion by Morris & Hollister of the Mare Island Laboratory. (3) Method using a Polyblend-Neoprene blend. (4) Method using a casting resin and a throw-away mold. The test results show that none of these methods is completely satisfactory.
(over)

AD-140 125

The standard method should be revised to give better adhesion between the molded packing and Neoprene cable sheaths. The methods using straight Neoprene and a Polyblend-Neoprene blend are unsatisfactory. The use of a cold setting casting resin holds promise of being satisfactory, while cutting costs by one-half to two-thirds. (Author)

AD-140 180 Div. 11/2, 11/5, 14/9

Aberdeen Proving Ground, Md.
TEST OF FIFTY GALLON, SELF-SEALING, AUXILIARY OR JETTISONABLE FUEL CELLS, by James F. Hansen. Aug 57 [36]p. incl. illus. (Proj. no. TT1-649H, rept. no. 5)
Unclassified report

Available from ASTIA-DSC. Not releasable to foreign nationals or their representatives.

Three cells were installed in racks on various vehicles and were subjected to 38,440 miles operation over road surfaces ranging from paved highway to cross-country. A gunfire test was carried out on three other cells. Gum content of gasoline in the seventh cell was analyzed periodically during storage in the open. The auxiliary or jettisonable self-sealing fuel cells satisfactorily withstood endurance testing when carried in a truck body. When folded for storage or shipping, the collapsible cell occupies approximately one-fifth the space required for the standard fifty gallon metal fuel
(over)

AD-140 180

tank. Gum formed in the fuel cell at an excessive rate under open storage conditions. Generally the self-sealing qualities of these cells were considered satisfactory. (Author)

AD-140 572 Div. 14/1, 22/1, 22/5

Aerojet-General Corp., Azusa, Calif.
DEVELOPMENT OF AN IMPROVED BOMB SEALER FOR ELIMINATION OF INERT-SEALER EXUDATION, by E. T. Leidigh, R. D. McLeod, and A. A. Stenersen. Final rept. 10 May 56-5 June 57. 1v. incl. illus. tables. (Rept. no. 1282)
(Contract DA 23-017-501-ORD-2115, Proj. PA-2-1)
Unclassified report

Preliminary studies were made to determine the most feasible methods for developing suitable materials for use as a bomb sealer. Polyurethane foam was selected as the most promising material for the sealer. The adopted foam formulation was: 120.0g Multron R-18, 17.5g Mondur TD (TDI), 0.9g H_2O, and 0.133g FeAA. The material can be foamed in place without the generation of appreciable heat and is self-curing at 80 F in 72 hr, or can be cured at 110°F in 24 hr. High-explosive loading and temperature-cycling tests were conducted with 13 bombs which were loaded with TNT by using various inert sealers; eleven were subjected
(over)

AD-140 572

to 4-wk temperature-cycling tests between -65° and +160°F (MIL-STD-304). The polyurethane-foam sealer passed the temperature-cycling tests in perfect condition. No exudation occurred from any of the bombs in which either the polyurethane foam or Celite were used as sealers. However, when Celite alone was used as a sealer, sufficient pressure was transmitted to force calking from the seams around the base of the bombs. Exudation occurred from all other bombs that were temperature-cycled. The polyurethane foam met the physical requirements and was compatible with the standard explosives. (See also AD-135 528)

AD-140 613 Div. 22/8

Cornell Aeronautical Lab., Inc., Buffalo, N. Y.
RESEARCH ON HEAT TRANSFER AND OBTURATION FOR SMALL ARMS WEAPONS (Unclassified title) by Gerald A. Sterbutzel. Monthly progress rept. no. 28 for July 57, 15 Aug 57, 31p. incl. illus. tables (Rept. no. CA-948-c-25)
(Contract DAI 30-115-504-ORD(P)-596)
UNCLASSIFIED report

The problem of making the heat-transfer and durability characteristics of ring-seal designs optimum was studied. Tests were conducted on ring seals (SD-1, SD-2, and SD-3) and on barrel seals and Ford skirted seals. The SD-1 seal design showed excellent durability characteristics. It was possible to fire over 5000 rounds without the failure of any of the seals. Over 420 rounds could be fired without cookoff in a ring-seal drum if either IMR or ball propellant were used at an ambient gun temperature of 150°, and at least 420 rounds could be fired without cookoff in a retrofitted drum if the am- (over)

AD-140 613

bient temperature was 145° for IMR propellant and more than 150° for ball propellant. The SD-2 seals tended in the initial test to have extremely good temperature characteristics but the temperature advantage diminished considerably as the tests progressed. Six tests of SD-3 ring seals were completed. No real difference was determined in the erosion of 2 designs, and each of the designs maintained their dimensions quite well after 1457 rounds were fired. Temperature data indicated that the barrel seal holds tremendous promise for improving the cookoff characteristics of the M-39 gun. (See also AD-139 901)

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(Signature & Date)

AD-142 079 Div. 1/2

Douglas Aircraft Co., Inc., Santa Monica, Calif.
EVALUATION OF HIGH TEMPERATURE HYDRAULIC SEALS TO TEMPERATURE OF 550°F PART 1. MECHANICAL EVALUATION, by Charles W. Galloway. Rept. on Aircraft Hydraulic and Pneumatic Systems. Feb 58, 143p. incl. illus. tables (WADC Technical rept. no. 57-76, pt. 1)
(Contract AF 33(616)3026) Unclassified report

The high temperature hydraulic seal research and development program (AD-70 011) was extended to include investigations of hydraulic seals for use with OS-45 and MLO-8200 fluids at temperatures up to 400° and 550°F, respectively. The mechanical tests and the results of the evaluation of experimental elastomeric materials are summarized. Twelve basic test procedures were used: (1) chew resistance screening test; (2) backup screening tests, 400°F; (3) O-ring and backup endurance life, 400°F, OS-45; (4) O-ring temperature cycling endurance life, -65° to 400°F, OS-45; (5) high-low (over)

AD-142 079

temperature performance test, OS-45; (6) O-ring endurance life, 550°F, MLO-8515; (8) static elastomeric seal, 400°F, OS-45; (9) static elastomeric seal endurance life with temperature cycling, -65° to 400°F, OS-45; (10) metallic static seal, OS-45, 400°F; (11) static seal endurance life, 550°F, MLO-8200; and (12) backup endurance life with no mechanical actuation, 400°F, OS-45. Two polymers performed promisingly at 400°F. No seals were acceptable at 550°F, but Viton A had good thermal resistance at 550°F.

AD-142 604 Div. 10/2, 14/9
(Unannounced report)

Aerojet-General Corp., Azusa, Calif.
EVALUATION OF POLY-FBA O-RINGS IN JP-X, by F. H. Cassidy. 26 Apr 55 [4]p. incl. tables (Rept. no. MA-337, Suppl. no. 1) Unclassified report

AD-143 542 Div. 7/1

Sonotone Corp., Elmsford, N. Y.
INVESTIGATION LEADING TO THE DEVELOPMENT OF A NICKEL CADMIUM BATTERY HAVING A HERMETICALLY SEALED CONSTRUCTION, by Irwin M. Schulman and Paul F. Ritterman. Quarterly rept. no. 4, 11 May-10 Aug 57. 50p. incl. illus. tables.
(Contract DA 36-039-sc-73009) Unclassified report

Additional 3-1x1 (0.45 A. H.) batteries were subjected to automatic cycling (60 minute charge - 40 minute discharge) at discharge rates 0.07 to 0.20 ampere. These batteries have received 656 cycles. The 3-1/4x1 (0.1 A. H.) batteries were normalized with constant current charge-discharge cycles and replaced on cycle service. These batteries have received 414 cycles at discharge rates of 0.05 to 0.06 ampere. The vacuum cycling of the 0.45 A. H. cells was extended from 1149 cycles to 2256 cycles during this quarter at discharge rates from 0.07 to 0.15 ampere. The cadmium oxygen reaction, which is supposedly the rate controlling step in the re- (over)

AD-143 542

action mechanism of the sealed cell, was investigated at the low pressures of oxygen. Some reactions were run isothermally while others were allowed to proceed without temperature control. It was found that the reaction rates of the isothermal reactions were approximately 100 times less than the others. (Author) (See also AD-137 420)

AD-143 735 Div. 8/1, 8/2, 14/1, 26/1

New York U. Coll. of Engineering, N. Y.
SEALING TECHNIQUES FOR WAVEGUIDE AND COAXIAL ROTARY JOINTS, by Milton M. Sackman, Michael B. Maccarone, and Fred H. Posser. Quarterly progress rept. no. 8, 1 June - 31 Aug 56. 16p. incl. illus.
(Contract DA 36-039-sc-64476) Unclassified report

AD-143 970 Div. 14/1, 22/8, 22/12

Aberdeen Proving Ground, Md.
TEST OF NEW FACE-TYPE OIL SEALS CODE A PART NO. X16507-X16509, by L. G. Bach. 20 Sep 57, [18]p. incl. illus. (Proj. no. TT1-198, rept. no. 10) Unclassified report

The sealing performance was evaluated of Code A face-type seals, numbers X16507 and X16509 as installed on the M48 90-mm-gun tank. These seals retain the lubricant by a barrier established between 2 face-loaded rings which slide relative to one another during rotation. OES arctic oil was used for the test. Two X16509 seals installed on the compensating idler wheel spindles operated 5037 mi on various proving ground courses. Four X16507 seals installed at the no. 1 and no. 6 road-wheel positions operated 4024 mi. One oil addition was required in the left compensating idler during this period. Eight standard road-wheel seals, part no. 7364672, were modified and tested concurrently as control items. Seven of these seals (over)

AD-143 970

operated 5037 mi with minor oil additions; the eighth failed after 3930 mi. Code A face-type seals are concluded to be superior in sealing efficiency to the standard seal and similar to the performance standard established by the modified standard seals. Face-type seals operate satisfactorily with OES Arctic lubricant under normal ambient conditions. The recommendation is made that (1) the Code A seals be redesigned to attain interchangeability and cartridge construction, and (2) the standard road-wheel seal be modified by bonding or cementing the seal ring-spring adaptor gasket assembly.

AD-144 596 Div. 7/1, 14/1

Burgess Battery Co., Freeport, Ill.
PLASTIC SEALS, by C. J. Broughton. Quarterly engineering progress rept. no. 1, 31 May-31 Aug 57. 11p.
(Contract [DA] 36-039-[sc]-10527) Unclassified report

The purpose of this contract is to establish the production technique necessary to produce a plastic sealing material for dry batteries that will permit the molding and/or extruding of the material into discs which will include brass anode caps. The properties of the material must be such as to preclude leakage at the cathode or anode of the cell after the can has been mechanically spun tight over the plastic disc and hot solder has been applied to the cathode where it crimps to the disc and to the carbon cap on the anode. Equipment will be made for a pilot run of cells for "N", "A", and "B" size sealing discs. (Author)

AD-145 239   Div. 6/1, 14/1

Electrical Testing Lab., Portsmouth Naval Shipyard, N. H.
INVESTIGATION OF COMMUNITY STUFFING TUBE ASSEMBLIES, by E. Shea. 12 Dec 56 [44]p. incl. illus. (Evaluation rept. no. ETL-2564)   Unclassified report

Available to U. S. Military Organizations only

Various types of community stuffing tube assemblies were evaluated. The tests included (1) accelerated aging, (2) high impact shock, (3) vibration, and (4) air pressure at 25 psi at room temperature. The sample was tested for simulated cable slippage under bulk-head deflections. The tubes were examined for pressure effectiveness. Bulkhead deflection and deformation tests were also conducted. A 12-tube assembly with aluminum packing plates, 6 armored cables, and 6 rubber cables with improved rubber grommets was considered satisfactory provided the packing pressure was maintained at 75 psi or over. The connection between the pressure reservoir (over)

AD-145 239

and the packing plates must have at least a 3/8-in. id to permit and maintain adequate flow of packing material at indicated pressures. Modifications were necessary to reduce the over-all weight of the assembly to a degree sufficient to permit conductance of the tests. The inability of an assembly to maintain a minimum of 75 psi might result in leakage around the cables. The use of nylon stuffing tubes with split silastic grommets of approximately 50 durometer hardness was relatively effective in maintaining a seal when rubber cables were used.

AD-145 284   Div. 14/9, 26/6

Materials Testing Lab., Portsmouth Naval Shipyard, N. H.
EXAMINATION OF RUBBER ARTICLES ABOARD THE USS SEA ROBIN (SS407) by Jeremiah F. Sullivan and William R. Sheehan. 1 May 56, 5p. incl. table. (Inspection rept. no. S56-407)   Unclassified report

Available to U. S. Military Organizations only

The over-all condition of the rubber articles aboard the USS SEA ROBIN (SS407) was satisfactory. Much of the equipment was mounted with felt pads which were in poor-to-good shape. Gaskets, in general, were in good condition, the most notable exception being those on the torpedo tube muzzle doors. Although these rubber gaskets were still serviceable, they were worn and damaged and should be replaced. The main deck was covered with plastic (polyvinylchloride) tile in poor condition at points of most traffic and wear. It is recommended that wider use of rubber shock mountings (over)

AD-145 284

of the newer types be considered, especially to replace many of the felt pads now used, and also on the main engines to aid in noise reduction. (Author)

AD-145 307   Div. 14/9

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
DEVELOPMENT OF O-RINGS FOR SEALING OIL AT 300°F, by R. D. Ford. Progress rept. no. 1. 28 May 57, 8p. incl. illus. tables (Rept. no. 7-27)   Unclassified report

Available to U. S. Military Organizations only

Stocks prepared from Paracril D, Hycar 4021, Silicone W-96, and from a blend of Kel-F 3700 and Hycar 4021 were evaluated for suitability for use in seals subjected to air and ASTM Oil No. 2 at 300°F. All of the stocks except the silicone stock had compression sets greater than 100% after 14 days at 350°F in air, and were therefore considered unsuitable for this service. The compression set of the silicone stock under the same conditions was only 47%. This stock also showed good retention of tensile strength, ultimate elongation, and hardness after aging in hot air or in hot oil. Swell after 7 (over)

AD-145 307

days at 300°F in ASTM Oil No. 2 was only 11%. Seals prepared from this stock should be satisfactory for use under static conditions in air and/or oil at 300°F. Further work will be done to see if other silicone rubbers will give stocks having better properties than was obtained with the Silicone W-96 stock. Seals made from the best silicone rubber stock will be subjected to a simulated service test. If they prove to be satisfactory, specification tests and requirements will be proposed on the basis of the properties of the selected stock.(Author) (See also AD-145 308)

AD-145 357   Div. 7/3, 14/1, 14/9

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
MODIFICATION OF UG-1083A/U CONNECTOR-RECEPTACLE TO IMPROVE SEALING OF RG-160/U CABLE, by J. Oser. 9 Oct 56 [14]p. incl. illus. tables (Rept. no. 67-14)   Unclassified report

Available to U.S. Military Organizations only

The Rubber Laboratory has been developing improved seals for RG cables in connector-receptacles by incorporating thick rubber O-rings in the outboard end of the connector-receptacle. The latest work deals with the RG-160/U cable and the UG-1083A/U connector-receptacle. The outboard end of the UG-1083A/U connector-receptacle has been modified by incorporating two thick rubber O-rings, so that satisfactory seals are now obtained with RG-160/U cable. One of the O-rings creates a seal against the outer jacket of the cable; the other O-ring creates a seal against the inner jacket of (over)

AD-145 357

the cable. When tested, these seals withstood 1700 psi water pressure even after aging the assembly under no pressure for 78 hours at 158°F. The modified connector-receptacle contains three fewer parts than the original UG-1083A/U connector-receptacle. (Author)

AD-145 360   Div. 14/9, 31/5

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
DEVELOPMENT OF RUBBER STOCKS FOR USE IN SEALS FOR HYDRAULIC SYSTEMS CONTAINING CELLULUBE 220, by J. M. Holloway. Progress rept. no. 2. 12 Aug 57 [25]p. incl. tables (Rept. no. 90-8)   Unclassified report

Available to U. S. Military Organizations only

The Rubber Laboratory was requested by the Bureau of Ships to develop improved butyl rubber stocks for use in seals for hydraulic systems containing Cellulube 220. The new curing systems for butyl rubber described in Patent Nos. 2, 726, 224, 2, 727, 874, 2, 734, 039, 2, 734, 877, and variations of these systems were studied. Basically, the new curing systems comprise a bifunctional phenolic resin as the curing agent and an acid-yielding material, such as a metal chloride, polychloroprene, chlorosulfonated polyethylene, chlorinated wax, (over)

AD-145 360

etc., as the accelerator of the curing reaction. Stocks accelerated with ferric chloride gave outstandingly low compression set values as well as good tensile properties, but were not suitable because they corroded the mixing equipment and molds. Two improved stocks were developed containing Hypalon 20 as the accelerator, which gave low compression set values and good tensile properties, and which did not stain or corrode the processing equipment. The results of static sealing tests indicate that both stocks would be suitable for gasket service in hydraulic systems under pressures up to 1500 psi and at temperatures of -20°F and above. The development of butyl rubber stocks for this application is being continued. (Author)

AD-145 361   Div. 14/9, 26/8

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
DEVELOPMENT OF STOCK AND SPECIFICATION FOR GASKETS FOR HIGH PRESSURE PIPE AND TUBE UNIONS, by A. E. Barrett. 14 Sep 56 [26]p. incl. illus. tables (Rept. no. 90-6)   Unclassified report

Available to U. S. Military Organizations only

A satisfactory stock was developed for the gaskets for unions in pipes carrying petroleum-base or phosphate-ester hydraulic fluids at pressures up to 10,000 psi, or air at pressures up to 3000 psi. The gasket sizes and shape and the pipe unions are described in BUSHIPS Drawing 5000-S4823-841337, Rev. E. The stock developed was a Hycar 1041 (high acrylonitrile) stock plasticized with Hycar 1312 (liquid acrylonitrile rubber), having a Shore A hardness of 90, a tensile strength of 3990 psi, and an ultimate elongation of (over)

AD-145 361

75%. Gaskets of this stock conditioned in test jigs containing petroleum base fluid (MIL-0-5606) or phosphate ester (Cellulube 220) for 36 days at 220°F sealed against 10,000 psi pressure applied for 30 minutes and against 5000 psi pressure applied for 60 days. Gaskets contained in the test jigs containing no fluid and aged for 36 days at 220°F, sealed air at 3000 psi pressure. Specification tests, procedures, and requirements are recommended based on the results of tests performed on the stock developed. (Author)

AD-145 362 Div. 14/9, 31/5

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
INVESTIGATION OF PROPERTIES OF STOCK USED BY STANDARD PRODUCTS COMPANY FOR LVT P5 TRACK SEAL, DRAWING NO. 1190607 ALT. 1, by J. Oser. 20 May 57, 7p. incl. tables (Rept. no. 154-5) Unclassified report

Available to U. S. Military Organizations only

The Bureau of Ships stated that the rubber seals for LVT-P5 amphibious tank treads, described in drawing no. 1199607 Alt 1, were being made satisfactorily by the Standard Products Company from their compound number H541-1. It was desired that the Rubber Laboratory determine certain physical properties of the vulcanizate used in those seals in order that a specification could be prepared. The Laboratory was informed of the recipe and cure (7 minutes at 330°F) used for the seals, and was given tensile sheets and 1/2-inch thick compression set specimens supposedly (over)

AD-145 362

made according to this recipe. The Laboratory also prepared tensile sheets and compression set plugs according to the recipe. Some sheets and plugs were cured for 7 minutes at 330°F; others were cured for longer times at 330°F and 310°F. The results of tests on these specimens, from the standpoint of comparison between the Company specimens and the Laboratory specimens, were as follows: specific gravity, same; tensile properties, laboratory better; tensile properties after oven aging, approximately the same; tensile properties after oil immersion, approximately the same; oil swelling, same; hot compression set, company better; cold compression set, same. (Author)

AD-145 390 Div. 14/1, 14/9, 33/1

Material Labs., Puget Sound Naval Shipyard, Bremerton, Wash.
TESTING OF TWO COLD-APPLIED CONCRETE JOINT SEALING COMPOUNDS, by W. B. Lew. 22 Mar 57. 4p. incl. table (Rept. no. R-233-57) Unclassified report

Available to U. S. Military Organizations only

Two commercial sealants were evaluated to determine their suitability for use in joints of concrete paving on airfields according to Spec SS-S-170 of 23 May 1956. Sealant A, a cold-applied, jet-fuel-resistant material, consisted of 16 parts by weight of a dry powder catalyst and 44 parts of a viscous liquid base material. Sealant B was a similar material mixed in a ratio of 25 parts by weight of catalyst to 42 parts of base material with the liquid component heated to about 190°F. A penetration test was conducted with sealant A mixed at 70° to 80°F. Tests were also conducted (over)

AD-145 390

with the compound mixed at about 160°F. Tests on sealant B were conducted with the compound mixed at about 190°F. The sealants were tested for penetration on materials, solubility, flow, bond strength, and flame resistance. The results of these tests indicated that the over-all requirement is more nearly met by sealant A than by sealant B.

AD-145 518 Div. 14/1, 14/7, 14/8

Bjorksten Research Labs., Inc., Madison, Wis.
DEVELOPMENT OF SEALER FOR POTABLE WATER TANK INTERIORS, by L. L. Yeager. Final technical rept. Sep 57, 1v. incl. illus. tables. (Contract DA 11-022-ORD-2120, Proj. TT1-789) Unclassified report

Includes AD-145 518(a).

Protective coating systems for corroded metal surfaces are being studied to prevent further corrosion of the underlying improperly prepared surfaces. The systems were to be resistant to prolonged exposure, fresh water, and gasoline. No taste or toxicty was to be imparted to the water. The specimens which were tested consisted of low carbon steel, 24S-T3 aluminum alloy, or FS-1-H24 magnesium alloy. Polyurethane based enamels applied over properly prepared metal surfaces were more suitable for prolonged exposure to potable water and gasoline than coatings of epoxy or acrylic (over)

AD-145 518

resins. Pickled, mild-steel panels, primed with MIL-C-15328A wash primer and top coated with polyurethane enamel withstood 3300 hr of immersion in fresh water without failure. Dow no. 7 treated magnesium panels with or without MIL-C-15328A primer beneath the enamel also successfully passed 3300 hr of water immersion. The enamel adhesion to the Dow no. 7 layer exceeds the Dow no. 7 coating's adhesion to magnesium. Aluminum prepared with MIL-C-10578 acid-type cleaner and Alodine no. 1200 (MIL-C-5541) showed good enamel adhesion; 1560 hr of fresh-water immersion caused no failure in this protection system. The application of protective coatings over rusted, improperly prepared metal surfaces did not give stable systems with resistance to further corrosion. Proper cleaning, and passifying of the metal surfaces prior to enameling is strongly recommended for good service life of the coated interiors. The polyurethanes when properly compounded and cured are highly resistant to moisture and gasoline, but there is some permeability to every coating regardless of type or thickness.

AD-145 832 Div. 14/9, 17/2, 26/8, 27/1, 31/2

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
DEVELOPMENT OF STOCKS AND MOLD FOR SPECIAL CYLINDER LINER SEALS, by A. E. Barrett. Technical rept. 22 Oct [11]p. incl. illus. tables (Rept. no. 165-10) Unclassified report

Available to U. S. Military Organizations only

Serious pitting of the cooling chambers in the aluminum blocks of Packard diesel engines used in minesweepers has been observed. The pitting was apparently due to either corrosion, cavitation, or both. The use of a rubber absorber to prevent cavitation by absorbing the vibrational or shock energy in the cooling water was suggested in Bureau of Ships correspondence. The rubber absorber was to be in the form of a special O-ring having a sleeve which would extend into the cooling chamber. The special O-ring would replace one of the stand- (over)

AD-145 832

ard O-ring cylinder liner seals. At the request of the Bureau, the Rubber Laboratory has developed nitrile rubber stocks of two hardnesses and a mold for the special cylinder liner seal, and has successfully fabricated 20 seals of each stock for trial. (Author)

AD-145 833 Div. 14/9, 26/1, 26/8

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
EFFECT OF HEAT TREATMENT OF CARBON BLACK MASTERBATCH ON PROPERTIES OF BUTYL VULCANIZATE, by R. Chow and R. R. James. Technical rept. 23 Mar 56 [12]p. incl. tables (Rept. no. 12-5) Unclassified report

Available to U. S. Military Organizations only

The Rubber Laboratory was requested by the Bureau of Ships to determine whether a new mixing procedure developed by Esso Laboratories for processing butyl rubber stocks would improve the physical properties of butyl stocks designed for naval applications. The new processing method involved vigorous heat treatment of a butyl rubber-carbon black masterbatch prior to the final mixing of the stock. It was found that the heat treatment caused a significant reduction in ultimate elongation, tear resistance, and Shore hard- (over)

AD-145 833

ness, significant increases in modulus and resilience and minor reductions in hot and cold compression set of vulcanizates produced from the masterbatches. The heat treatment did not improve the hot compression set of the vulcanizates to a point where they would be suitable for use in gaskets or mountings. (Author)

AD-145 846 Div. 14/9, 26/8

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
DEVELOPMENT OF O-RINGS FOR SEALING OIL AT ELEVATED TEMPERATURES, by R. D. Ford. Final technical rept. 18 Sep 57 [16]p. incl. illus. tables (Rept. no. 7-28) Unclassified report

Available to U. S. Military Organizations only

Stocks prepared from Viton A, Silastic 432, Silastic LS-53, Silicone W-96, and SE-30 Silicone Gum were evaluated for suitability for use in seals subjected to air and ASTM Oil No. 2 at 300°F. Only the Silicone W-96 stock had the necessary physical properties for use under these conditions. In simulated service tests, only the seals molded from the Silicone W-96 stock sealed satisfactorily. The Silicone W-96 stock was also evaluated for use as seals subjected to air and ASTM Oil No. 3 at 212°F. Both physical and simulated service tests showed it to be satisfactory. (Author) (See also AD-145 307)

AD-146 228 Div. 14/9, 31/5

Pearl Harbor Naval Shipyard, T. H.
EXAMINATION OF RUBBER GOODS ON THE USS HASSAYAMPA (AO 145) by V. E. Hargrave. 1 July 57 [12]p. incl. tables (Rept. no. 75726) Unclassified report

Available to U. S. Military Organizations only

The problem: To obtain information on the service life of rubber goods on the USS HASSAYAMPA (AO 145). Findings: The ship has been in commission a little over two years. The hatch and door gaskets are beginning to show ozone cracking but slight grooving. Fuel hoses are less than three years old and are in good condition. A few of the larger diameter hoses have been partially crushed during handling. Latex mastic deck covering in wet spaces is giving excellent service. Type II latex decking material is currently being applied to the weather deck. Gray, diamond pattern synthetic rubber deck matting is preferred over the corrugated type for comfort and ease of cleaning. (Author)

AD-146 229 Div. 14/9, 31/4

Pearl Harbor Naval Shipyard, T. H.
EXAMINATION OF RUBBER GOODS ON THE USS STERLET (SS-392) by V. E. Hargrave. 20 July 56 [10]p. incl. table (Rept. no. 71981)
Unclassified report

Available to U. S. Military Organizations only

The problem: To obtain information on the service life and suitability of rubber goods aboard the USS STERLET (SS-392). Findings: The USS STERLET has not experienced appreciable cold water service. The thiokol on the underwater appendages is four years old and shows 10 to 20% peeling. The Type I shaft covering is also four years old and appears to be in good condition. The M-G sets and pump units are well mounted on compression and other standard type mounts. Gage and instrument panels and torpedo room hydraulic units are on felt mounts. The torpedo breech gaskets are badly oil softened and need for an oil resistant gasket stock is indicated. (Author)

AD-146 231 Div. 31/3, 31/5

Pearl Harbor Naval Shipyard, T. H.
EXAMINATION OF RUBBER GOODS ON THE USS SILVERSTEIN (DDE-534) AND THE USS McGINTY (DE-365) by V. E. Hargrave. 27 Nov 56, 8p. (Rept. no. 72990)
Unclassified report

Available to U. S. Military Organizations only

The problem: To obtain information on the service life of rubber goods on the USS SILVERSTEIN and the USS McGINTY. Findings: Rubber articles which showed failure in service were sponge rubber pads on earphones, fabric covers on 15 lb. CO_2 fire extinguishers and high durometer deck expansion joints. Fuel and suction hoses are showing a service life of 12 years or more. Training of ship's personnel in the use of hose repair compound would extend the useful life of these hoses even more. Greater emphasis on proper stowage of RBA masks and proper repair procedures would be of value. (Author)

AD-146 232 Div. 31/3, 31/4

Pearl Harbor Naval Shipyard, T. H.
EXAMINATION OF RUBBER GOODS ON THE USS TIRU (SS-416) by V. E. Hargrave. 5 Sep 56 [10]p. incl. tables (Rept. no. 72175) Unclassified report

Available to U. S. Military Organizations only

The problem: To obtain information on the service life and suitability of rubber goods aboard the USS TIRU (SS-416). Findings: The shaft coatings were Type I rubber and had been in service eight years. Leakage was found at some waterstops and these were a few surface snags. Failure at the waterstops deserves further attention. The neoprene battery compartment linings were also eight years old and still giving good service. Natural rubber torpedo breech gaskets showed oil swelling and should be changed to an oil resistant type. ARL mountings were found to be unevenly loaded and exhibited a fair amount of set after five years of service. Many of the gage panels are still mounted on felt. (Author) (See also AD-146 229)

AD-146 748 Div. 14/1, 14/9

Material Lab., New York Naval Shipyard, Brooklyn.
QUALIFICATION OF SYNTHETIC RUBBER O-RINGS, SUBMITTED BY STILLMAN RUBBER COMPANY, CULVER CITY, CALIFORNIA. Final technical rept. 8 Aug 57 [5]p. incl. table (Lab. proj. no. 5902-5)
Unclassified report

Available to U. S. Military Organizations only

The sample material was composed of a nitrile type synthetic rubber compound. The free sulfur content was found to be 0.2% as against a specified maximum of 0.[illegible]%. The sample O-ring material, compound No. SR 866-70 is not in conformity with the requirements of specification with respect to elongation in the initial condition and after oil immersion, and tensile strength after oven aging. Also, the inside diameter of size 19 O-rings exceeds the specified maximum. (Author) (See also AD-146 749)

AD-146 749 Div. 14/1, 14/9

Material Lab., New York Naval Shipyard, Brooklyn.
QUALIFICATION TEST OF SYNTHETIC RUBBER O-RINGS, SUBMITTED BY E. F. HOUGHTON AND COMPANY, PHILADELPHIA, PENNSYLVANIA. Final technical rept. 16 July 57 [6]p. incl. table (Lab. proj. no. 5902-6) Unclassified report

Available to U. S. Military Organizations only

The sample material was composed of a nitrile type synthetic rubber compound. The free sulfur content was found to be 0.5% as against a specified maximum of 0.3%. The sample O-ring material, compound No. 10V70-215 is not in conformity with the requirements of specifications with respect to free sulfur content. (Author) (See also AD-146 748)

AD-147 092 Div. 4/1, 14/5

Naval Engineering Experiment Station, Annapolis, Md.
JOHN CRANE STYLE 140, METAL-COVERED STERN-TUBE PACKING (SYMBOL 1405) SUBMITTED BY CRANE PACKING COMPANY, MORTON GROVE, ILLINOIS, by R. F. Stone. Qualification rept. 23 Sep 57, incl. tables (EES rept. no. 010367)
Unclassified report

Available to U. S. Military Organizations only

A qualification test of John Crane Style 140, metal-covered, stern-tube packing, product of Crane Packing Company, was conducted under Military Specification MIL-P-16374A. Satisfactory performance was obtained from the packing under simulated-service conditions, but the oleic acid content of the tallow lubricant is higher than allowed by the specification. The packing would be satisfactory for naval use if the tallow lubricant for the fibrous portion were of a quality complying with the specification requirements. (Author)

AD-147 837 Div. 12/1, 12/3, 12/4, 14/1

North American Aviation, Inc., [Downey, Calif.].
FILL AND DRAIN TYPE SEALING OF PRESSURIZED FUEL TANKS AND COMPARTMENTS. 17 Sep 57, 44p. incl. illus. tables (Rept. no. AL-2603)
(Contract AF 33(600)28469) Unclassified report

The state-of-the-art of fill-and-drain sealing which can be utilized to seal minute cracks in missile fuel tanks and compartments is reported. The process consists essentially of filling the tank with a relatively thin sealing compound, applying pressure to force the sealant into all cracks, draining the sealing compound, drying, and allowing the residual film to cure (generally by heating) to a tough elastic seal. In view of the simultaneous high temperature air/fuel conditions imposed on these sealants, the study was limited to Buna N/phenolic resin compounds and silicone-base sealing compounds. Compounds belonging to both groups of sealants were suitable. Established methods for fill-and-drain sealing on a production scale were set up for fuel tanks and
(over)

AD-147 837

compartments of the X-10 and XSM-64. The XSM-64A process specification for fill-and-drain sealing was tentatively set up. Necessary changes and revisions of the fill-and-drain procedure for fuel tanks and compartments of XSM-64A were made on a laboratory scale to be later established on the production scale.

AD-147 856 Div. 7/4, 10/5, 12/4, 27/3

North American Aviation, Inc. [Downey, Calif.].
LOX TANK VENT VALVE ELECTROFILM HEATER. 15 Sep 57, 8p. incl. illus. (Rept. no. AL-2668)
(Contract AF 33(600)28469) Unclassified report

Preliminary qualification tests were conducted on a 250-w electrofilm heater to determine (1) the compatibility of the electrofilm material with liquid and gaseous oxygen and (2) the operating characteristics of the heater when exposed to temperatures down to -320°F, with liquid nitrogen as the medium. The tests include impact sensitivity, temperature control, thermal shock, and arcing and hazard tests. The heater had unlimited possibilities for low-temperature applications, but the usage in liquid and gaseous atmospheres depended on the installation. The heater passed all tests satisfactorily; operating from 140° to -320°F. The O-ring seals were maintained within the operating range when the heater operated in a gaseous oxygen environment. In preparing the heater for exposure to LOX, the surface should be cleaned with trichlorethylene to remove grease.

AD-147 990 Div. 10/2, 14/8, 14/9

North American Aviation, Inc. [Downey Calif.].
FUEL AND HIGH TEMPERATURE RESISTANT ELASTOMERS. 15 Sep 57, 12p. incl. tables (Rept. no. AL-2600)
(Contract AF 33(600)28469) Unclassified report

Two samples of silicon-base polyurethane were cured (450°F, 1 hr), postcured (540°F, 4 hr) and exposed to JP-5 fuel, simultaneously, in the same test pot for 2 hr. One sample was in the liquid phase and the other in the vapor phase. The sample which was exposed to the liquid phase (400° + 10°F) had a 0.5% weight loss; the other (540°F) suffered a 0.1% weight loss. Two sets of 4-in.-diam panels were coated under positive pressure, with modified and pure silicon-base polyurethanes. No leakage was noted with the modified resin and only slight leakage at 750°F. The initial leakage of the uncoated panels was approximately 600 cu in./min. Triethanol amine and DMP-10, -14, and -30 were the most promising catalysts for resins in curing tests of 200°F for 8 hr followed by 12 hr at 250°F, 12 hr at 300°F, and 4 hr at
(over)

AD-147 990

400°F. A study was undertaken of catalysts which would not affect room temperature stability and at the same time reduce cure time and temperature. Panels with catalysts were exposed to liquid vapors at 700°F and 45 psig for 3.1 hr and fuel vapors up to 585°F and 40 psig for 3.1 hr without suffering leakage. Dimethyldiisocyanatosilane, pure silicone isocyanate resin (SIR), and unmodified and modified hybrid SIR were synthesized.

AD-149 681 Div. 14, 17, 26

Ministry of Supply (Gt. Brit.)
WETTING PROPERTIES AND MECHANICAL STRENGTH OF GLASS-TO-METAL SEALS (Benetzungseigenschaften und Mechanische Festigkeit bei Glas-Metall-Werschmelzungen) by W. Weiss. Nov 57 [13]p. incl. illus. tables, 18 refs. (Trans. no. TIL/T 4824 of Glastech. Ber. 29:386-392, 1956; [JSRP Control no. 571617]) Unclassified report

AD-149 799 Div. 22/1

Samuel Feltman Ammunition Labs., Picatinny Arsenal, Dover, N.J.
CONTROL OF EXUDATION IN THE 90mm M71 TNT LOADED SHELL, by Robert W. Heinemann and George J. Horvat. Dec 57, 17p. incl. illus. tables. (Technical rept. no. 2463) Unclassified report

Available to U. S. Military Organizations only

From the time when TNT and TNT-containing explosives were first used in ammunition, the Ordnance Corps was beset with the formidable problem of how to deal with the exudate which stems from them. Sealants and gaskets were experimentally tried as remedies, but they were not completely effective in preventing the flow of exudate into the threaded shell-fuze joint and to the outside of the shell. Absorbents were much more successful. Of the absorbents tested to date, Kimpak, a cellulosic material, and Celite FC, a nearly pure amorphous diatomaceous silica, have been shown to have high absorbing capacities and to be able to effec- (over)

AD-149 799

tively prevent the flow of the exudate into the shell fuze threads of 90 mm M71 shell. Of the two absorbents, Celite FC is easier to apply and appears to have a slightly greater absorbing capacity. It can be used either by itself or in combination with Kimpak. This report shows that the use of absorbents effectively inhibits the flow of exudate in the 90 mm M71 shell. Optimum quantities of absorbent, methods of application, and safety and functioning test requirements for artillery shell and for other types of ammunition still remain to be investigated. (Author) (See also AD-103 502, AD-136 117)

AD-149 961 Div. 22, 26

Rock Island Arsenal Lab., Ill.
TEST OF KEL-F PLASTIC PACKING SHROUDS IN A 75 mm PACK HOWITZER RECOIL MECHANISM, by L. R. Raisbeck. 30 Sep 57, 14p. incl. illus. tables. (RIA Lab. rept. no. 57-2366) Unclassified report

AD-150 474 Div. 27

Naval Ordnance Test Station, China Lake, Calif.
LOCKSEAL JOINT FOR ROCKET COMPONENTS, by Sydney R. Crockett. 11 Oct 57, 53p. incl. illus. tables. (NOTS 1492; NAVORD rept. no. 5275, Supersedes NOTS TM 1966, AD-43 804) Unclassified report

In response to the need for an improved means of uniting the components of a rocket, the Lockseal Joint has been conceived and developed. It locks and seals one component to another by forcing metal from the wall of one into a rubber-lined, circumferential groove in the surface of the other. In work to date it has been applied to 1- to 3-in., soft-steel and hardened-steel motor tubes, and to 1- to 5-in., soft-aluminum and hardened-aluminum tubes--the walls of both steel and aluminum tubes being of varying thicknesses. Numerous tests have demonstrated its superiority in many respects, including strength, axial alignment, and producibility, to joints now commonly used in rocket design. (Author)

AD-150 973 Div. 1

Douglas Aircraft Co., Inc., Santa Monica, Calif.
EVALUATION OF HIGH TEMPERATURE HYDRAULIC SEALS TO TEMPERATURES OF 550°F. PART II. POLYMER COMPOUNDING AND EVALUATING, by R. G. Larkin and F. W. Bickel. Rept. for Aug 55-May 57, on Aircraft Hydraulic Systems. Feb 58, 106p. incl. illus. tables (WADC Technical rept. no. 57-76, pt. 2)
(Contract AF 33(616)3026) Unclassified report

Research toward the development of elastomeric compounds suitable for use in seals for silicate ester hydraulic fluids at 400°F and 550°F is described. Several elastomer gums were compounded and tested for resistance to the subject fluid at the target temperature. Only one polymer, Viton A, exhibited a balance of properties which offer a possible successful solution to the problem. All other polymers evaluated have one or more objectionable characteristics which limit their usefulness for this application. A new app h for (over)

AD-150 973

weighing the merits of compounds, by use of true stress and strain data was developed. Theoretical basic chemical and structural requirements for polymers suitable for the subject application are reviewed and references are cited to substantiate these views. A device for physical testing O-ring specimens in fluid at elevated temperatures was designed and fabricated. A new test tube for fluid aging specimens at high temperature was designed. No specific compound is presented in the report for the proposed application. However, basic compounds having encouraging physical properties, and fair mechanical properties were developed. (Author) (See also AD-142 079)

AD-151 257 Div. 4/3, 14/1, 14/9

Products Research Co., Los Angeles, Calif.
QUARTERLY PROGRESS REPT. NO. 3, AUGUST 1-NOVEMBER 1, 1957, by I. P. Seegman. 15 Nov 57, 1v. incl. illus.
(Contract AF 33(616)3976) Unclassified report

Available only by request to Wright Air Development Center, Wright-Patterson AFB, Ohio, Attn: WCRTM-1

Development of sealants which will withstand exposure to fuel, vapor, liquid fuel at temperatures above 350°F.

Materials with good elastomeric properties were prepared by the action of tetra-isopropyl titanate on various HFPA polymers and copolymers. These materials were severely degraded by exposure to JP-5 fuel at 350°F for 5 days. The material prepared from HFPA, M_n 17,000 at room temperature had a tensile strength in excess of 3000 psi. Elastomers obtained from the resin PAH-1 filled with iron oxide and cured with (over)

AD-151 257

Bz_2O_2 at 250°F showed weight increases of 2 to 3% after exposure to JP-5 fuel at 350°F for 5 days. The effects of numerous additive systems on GE room temperature vulcanizing silicone compound 81712 were investigated. Promising formulations were developed for use as pressurization sealants for high temperature applications. An optimum cure was required for best adhesion of Viton A sealants after water immersion and exposure to oven aging at 450°F. Adhesion to Ti was obtained after exposure to fuel at 450°F by compounding the sealant with $ZnCrO_4$. Filleting sealants which had excellent adhesion to stainless steel after exposure to Type II fuel at 450°F for 40 hr were developed from Viton A. A formulation for a groove sealant was developed which was able to withstand 7 days of total immersion in Type II fuel and JP-4 fuel at operating pressures of 25 psi, followed by 48 hr at 450°F for 7 days at 350°F air aging. (See also AD-144 877)

AD-151 181 Div. 14/1, 14/9, 25/8, 26/8

Boeing Airplane Co., [Seattle, Wash.]
DESIGN DATA FOR O-RINGS AND SIMILAR ELASTIC SEALS, by Frank W. Tipton, George E. Trepus and others. Rept. for 1 June-31 Dec 57 on Rubber, Plastic, and Composite Materials. Apr 58, 88p. incl. illus. tables (WADC Technical rept. no. 56-272, pt. 3)
(Contract AF 33(616)2867) Unclassified report

This is a continuation of a study to gain knowledge of design data for O-rings and similar elastomeric seals. The literature survey on O-rings and seal design has been continued and enlarged to include the effects of irradiation and extreme low temperature on seal design. Low temperature tests, relaxation and volume change tests, screening tests, and functional tests have been conducted. The functional tests include pulsed annulus tests with various groove configurations and reciprocating shaft tests with and without different types of backups at room temperature and at elevated temperatures. (over)

AD-151 181

No overall relationship between physical property values and seal efficiency has been found. It has been noted, however, that certain physical properties appear to be important to seal life for specific seal conditions. (Author) (See also AD-131 094)

AD-151 729 Div. 26, 20

Atomic Energy Research Establishment (Gt. Brit.).
INSULATED LEAD-THROUGH SEALS FOR USE AT HIGH TEMPERATURES, by F. Wade. May 57, 2p. illus. (AERE rept. no. R/R 2271; HX. 3021) Unclassified report

Materials and processes are described for making seals brazed to metals with a Ti-cored Cu-Ag eutectic alloy. The seals were made from Vitreosil capillary tube and 30% Ni/Fe; from Steatite, Sintox, and synthetic sapphire. The metals can be either 42% Ni/Fe or 48% Ni/Fe. Good results were also obtained with Nilo K (29% Ni/17% Co/54% Fe). Quartz-metal seals should not be operated at temperatures above 150°C because of a sharp increase in the coefficient of expansion of 30% Ni/Fe.

AD-151 747 Div. 12/1, 12/3, 14/9, 26/8

North American Aviation, Inc. [Downey, Calif.].
FABRICATION AND TESTING OF LIQUID COOLED INFLATABLE WARHEAD DOOR SEAL. 13 Sep 57, 30p. incl. illus. tables (Rept. no. AL-2635)
(Contract AF 33(600)28469) Unclassified report

Liquid-cooled inflatable seals were developed to meet the service requirements for a pressurized access door under 16 psig of compartment pressure and with a skin temperature of 700°F. Two types of silicone seals, designed to resist extrusion through a 1/8-in. door gap, satisfactorily sealed the Navaho warhead door. Application of Dow-Corning silicone grease (DC-300, 100,000 cps) aided in the initial sealing and sealing. Of the 2 seal configurations tested, the silicone-rubber-capped version was superior to the silicone seal whose crown was reinforced by glass-fiber tape. The procedures for fabricating and testing these seals are discussed in detail.

AD-153 033 Div. 22

Rock Island Arsenal Lab., Ill.
TEFLON PACKINGS WITHOUT CUP RINGS FOR HYDROPNEUMATIC RECOIL MECHANISMS, by W. E. Peterson. 25 Nov 57, 37p. incl. illus. tables (Proj. no. TR1-1031; Technical rept. no. 57-2788) Unclassified report

Experimental Teflon plastic packings, of two designs without supporting cup rings, were tested in 75 MM Pack Howitzer and 105 MM Howitzer Recoil Mechanisms at sub-zero, normal and elevated temperatures. Type R packings, with a rectangular cross section, distorted the anti-friction metal backup. Type V packings, with angular sides eliminated this condition. The experimental packings did not provide as good a seal as standard packings with standard packing spring loads, but produced less wear on the rod and cylinder surfaces. Floating piston frictions were considerably lower than with standard packings, however, recoil (over)

AD- 153 033

mechanism action was not noticeably affected. The Teflon packings were away quite rapidly during gymnastication. This wear reduced the packing spring loads considerably in several test mechanisms. Felt and synthetic sponge rubber saturated with oil were used in place of grease in the floating piston void of the 75 MM Pack Howitzer Recoil Mechanisms. The felt became dry and charred during the tests. The synthetic sponge rubber was not noticeably affected and provided good lubrication for the duration of the test. A new type of index pin packing, consisting of a Teflon plastic sleeve surrounded by synthetic rubber, was used in place of the conventional all synthetic rubber packing, which had given trouble in prior tests. The new type packing functioned satisfactorily under all temperature conditions. (Author) (See also AD-72980)

AD-153 347 Div. 8/4

Sylvania Electric Products, Inc., Woburn, Mass.
INDUSTRIAL PREPAREDNESS STUDY: TRANSISTOR MANUFACTURING, by C. M. Chang. Quarterly rept. 1 Apr-30 June 57, on Device 2. 17p. incl. illus. (Contract DA 36-039-sc-72719) Unclassified report

Progress is described which was made in the hermetic seal problem by package modifications and improvements in brazing. With the new package, the diametral dimension was increased to 0.600 in., there is a 0.020-in. thick shoulder around the package, and the over-all height was reduced to 0.300 inches. The advantages tend to decrease the thermal stress at the glass lead-throughs and thus reduce cracks. Together with the improvements in brazing, the leak problem caused to be serious. Beta stability was improved by discontinuing the use of SR98 in the encapsulation process. Units were vacuum baked for 16 hr at 105°C after etching and before heliarc welding, and were baked at 100°C for 100 additional hours after heliarc welding. Initial data showed good current gain stability. Work on process (over)

AD-153 347

improvement continued, particularly with respect to alloying and electrolytic etching. Greater temperature control was achieved. The 150 state-of-art samples were delivered. (Author)

AD-153 356 Div. 14

Air Technical Intelligence Center, Wright-Patterson Air Force Base, Ohio.
RUBBER IN HYDRAULICS, by L. Kopec. [1957] 2p. (Trans. no. F-TS-8979/I of Strojirenstvi 5:301-305, 4 Dec 55) Unclassified report

A study on the mixing of rubber with thermoplastic and thermosetting materials is presented. The physical properties of rubber and their determination in accordance with Czech State Standards are described. A classification of various types of rubber is given. Various methods of employing rubber in the role of packing (sealing) materials for hydraulic mechanisms are listed. Various rubber packing sleeves; rubber resistant to the effects of oil, etc. are discussed. Chapter 1 deals with the State Standards for the basic characteristics of rubber: its tensile strength, hardness, elasticity, resistance to acids and bases, thermal properties, frost resistance, resistance to liquid fuel, solvents and oils. In chapter 2, the author discusses the possibilities (over)

AD-153 356

of the application of rubber in hydraulics; including static packing, and dynamic packing. The latter is divided into packing for rotational movement, linear packings and combinations. He discusses packings (gaskets) made of soft rubber with the exception of static packings made of high pressure plates, and piston gaskets made of asbestos fiber with metal inserts. (Author)

AD-153 546 Div. 8/4, 14/2

General Electric Co., Schenectady, N. Y.
ADVANCES IN CERAMICS RELATED TO ELECTRONIC TUBE DEVELOPMENT, by Louis Navias. Apr 53, 89p. incl. illus. tables, 31 refs. (Rept. no. RL-833) Unclassified report

Some of the older vacuum tubes operating at 1-Mc frequency or less used glass envelopes and porous ceramic refractories for internal supports. Some of the more modern "electronic tubes" operating at 10,000 Mc (3-cm wave length) require vacuum-tight ceramic envelopes having: (1) physical stability at operating temperatures (preferably a highly crystalline composition); (2) good mechanical strength; (3) surfaces which will take metallizing; (4) metal or alloy components of comparable thermal expansions; (5) low dielectric losses, even after being heated in a reducing atmosphere such as hydrogen, or in a vacuum. After some historical background, the report discusses changes in properties of a few crystals and glasses and gives many detailed composition studies of four types of bodies, including (over)

AD-153 546

such properties as linear thermal expansion and dielectric properties at 3-cm wave length, viz.:

Alumina	K 8.4	PF 0.0006
Forsterite	K 6.3	PF 0.0006
Zircon-talc	K 7.5	PF 0.0026
Steatite	K 6.6	PF 0.0005

A brief account is given of the development of sealing techniques and some details are given for three modern methods of making ceramic-metal seals by using (1) Mn-Mo, (2) MoO_3, (3) TiH_2 as the active metallizing agents. (Author)

AD-153 553 Div. 7/1

Sonotone Corp., Elmsford, N. Y.
INVESTIGATION LEADING TO THE DEVELOPMENT OF A NICKEL CADMIUM BATTERY HAVING A HERMETICALLY SEALED CONSTRUCTION, by Irwin M. Schulman. Final rept. 10 Aug 56-30 Nov 57. 74p. incl. illus. tables.
(Contract DA 36-039-sc-73009) Unclassified report

The reaction mechanism of the sealed-type sintered-plate Ni-Cd cell was investigated. Cycle-service data indicated that, for an extended cycle life under the 60 min charge - 40 min discharge duty cycle, discharge rates under 0.150 amp should be used when charging at a constant potential of 1.50 v/cell. This applies to an 80% voltage regulation. A greater number of cycles may be obtained if the voltage regulation is reduced to 77%. The 1X1 cells cycled at 120°F and at 1 to 2 mm Hg pressure received only 100 cycles. The 1/4 x 1 cell fell below 80% regulation because of charging efficiency. (over)

AD-153 553

The low temperature characteristics of D size cells were investigated. These will give about 85% of room-temperature capacity at 0°F at a discharge rate of 0.4 amp and 55% of room temperature capacity at -40°F at the same rate. At a discharge rate of 2.0 amp about 83% of room-temperature capacity can be expected at 0°F, and 43% at -40°F. The complete reaction which occurred at the charged Cd negative in the presence of O and KOH solution was considered as
$2Cd + O_2 + 2H_2O \rightarrow 2Cd(OH)_2$ and as $2Cd + O_2 \rightarrow 2CdO$.
A comparison of rate of change with rate of reaction indicated that no simple relation exists between the two. An investigation into temperature coefficient yielded a value of 5.5×10^{-7} mol of O consumed/min/°K. The reaction rate of the negative Cd plate decreased with increased concentration of the electrolyte solution in which it had been charged. Low concentration of hydroxide which resulted in an increased concentration of H_2O appeared to cause an increase in the rate at which $Cd(OH)_2$ is produced.

AD-153 642 Div. 14, 22, 30

Rock Island Arsenal Lab., Ill.
INVESTIGATION OF GLASS FOR RECOIL MECHANISM CYLINDER COATINGS, by L. R. Raisbeck. 12 Nov 57, 25p. incl. illus. tables (Proj. no. TR1-1031; Technical rept. no. 57-2695) Unclassified report

Friction and wear properties were determined for glass, chromium plate and uncoated alloy steel in sliding contact with leather, Teflon, aluminum alloy and silver. Oils, Specification 2-132, and Specification MIL-O-5606 were used for lubrication. Tests were conducted at normal room temperature using a load of 500 psi and a velocity varying from 0 to 4 inches per second. There was no significant difference in friction due to cylinder material. There was a highly significant difference in friction due to packing material. Teflon had the lowest and leather the highest coefficient of friction. Dynamic friction decreased as the number of rubbing cycles increased from 100 to 34,000. Teflon plastic produced the least wear on the cylinder materials. Leather wore (over)

AD-153 642

deep grooves in the glass. Silver was more wear resistant than Teflon and aluminum. Leather could not be quantitatively evaluated. Glass and chromium plating were more wear resistant than uncoated steel except when used with leather. Slightly lower frictions and slightly less wear of the cylinder and packing materials were obtained when specimens were lubricated with oil, Specification 2-132, than when lubricated with oil, Specification MIL-O-5606. (Author)

AD-154 494 Div. 22, 26

Springfield Armory, Mass.
DEVELOPMENT OF DRUM SEALS FOR GUN, AUTOMATIC, 30mm, T182, by M. W. Pastore. Technical rept. on Gun, Automatic, 30mm, T182. 20 June 58, 26p. incl. illus. (Rept. no. SA-TR7-2015)
Unclassified report

Effort was made to eliminate stoppages caused by failure of drum seals in the 30-mm T182 automatic gun to extend the barrel life. Firing tests demonstrated that an adequate barrel breech face life and seal life could be obtained by using experimental drum seal no. 7 (a modification of the original cylindrical seal with the outer diameter reduced and 2 sealing rings added) in combination with a fluted cartridge chamber and a drum cradle reinforced between side arms and gas cylinder housing.

AD-154 814 Div. 4/1, 30/6

Atomic Energy Research Establishment (Gt. Brit.).
BUBBLE LEAK DETECTION, by J. G. S. Biram. Mar 57 [14]p. incl. illus. tables, 20 refs. (AERE rept. no. CE/R 2066; HX. 2797)
Unclassified report

Some preliminary work was carried out (August 1954) to find the sensitivity of leak detection by gas bubbles in some organic fluids. The method was several hundred times more sensitive than with water for a given gas pressure and at about 15 p. s. i. (gauge) was comparable with ordinary vacuum testing, and it was used by unskilled personnel. Hydrogen-ether was the most sensitive combination, but hydrogen or helium and methanol were usually employed for safety reasons; bottled compressed air can be used, with some loss of sensitivity. (Author)

AD-154 972 Div. 14/1, 31/3

Directorate of Materials and Explosives Research and Development (Gt. Brit.).
THE SEALING OF THREADED JOINTS: PART I: SEALING COMPOSITIONS AND JOINT DESIGN, by R. W. Bryant and W. A. Dukes. Oct 57 [25]p. incl. illus. tables (Rept. no. DMXRD/SEAL/57/3; [JSRP Control no. 580271])
Unclassified report

The most important signal factor in the sealing of threaded joints is the quality of the sealing composition. Cements which dry merely by evaporation of solvent are greatly inferior to solvent-free cements and to lutings. R.D. 1151B is the least, and R.D. 1286 the most, satisfactory cement tested. The next most important factor is the combination of joint design with mode of application of the sealing composition. Application to both threads is preferable, if it is permissible to force composition into the assembly. Sealing is also improved by screwing on to either (a) an external shoulder if composition is applied to the male threads, or (b) an internal shoulder if composition is applied to the female
(over)

AD-154 972

threads. A joint not screwed on to a shoulder is relatively difficult to seal. The metal and dimensions of the joint, and even incorrect application of the sealing composition, are individually of minor importance. (Author)

AD-155 541 Div. 1/2, 27/1

Propulsion Lab., Wright Air Development Center, Wright-Patterson Air Force Base, Ohio.
INVESTIGATION OF FIVE HYDRAULIC PUMPS REMOVED FROM KC-135 AIRPLANES WHICH EXPERIENCED HIGH TEMPERATURE INCIDENTS, by P. J. Strick. [9 Feb 58, 30p. incl. illus. (WADC Technical note no. 58-130)
Unclassified report

The case return flow of these pumps was above normal acceptable limits. The pumps were not in serviceable condition because of high internal leakage rates, O ring deterioration, and scored valving surfaces. The unsatisfactory condition of these pumps was caused by operation in KC-135 airplanes with fluid at temperatures above 300°F. It is recommended that (1) the airframe manufacturer provide adequate cooling provision to dissipate heat input of hydraulic pump which raises the fluid temperature in the reservoir, pump fluid supply line, and pump case return line, and (2) the pump manu-
(over)

AD-155 541

facturer take action to improve pump wear characteristics by redesign or improved quality control procedures. (Author)

AD-155 606 Div. 14

Products Research Co., [Los Angeles] Calif.
DEVELOPMENT OF HIGH TEMPERATURE SEALANTS, by Irvin P. Seegman, William Chevrvas and others. Rept. for 1 Feb 57-1 Feb 58, on Rubber, Plastics, and Composite Materials. June 58, 160p. incl. illus. tables (WADC Technical rept. no. 58-89) (Contract AF 33(616)3976)
Unclassified report

Fill and drain sealants from Viton A: The system consisted of a dilute solution of Viton A with phenolic resins as a primer; the Viton A sealant was compounded with silicone-coated SiO_2 and MgO_2 at 28% solids in methyl isobutyl ketone and an amine curing solution. The system had excellent resistance to fuels at 450°F, and good adhesion to stainless steel and Ti which was retained after oven-aging and immersion in salt H_2O. Filleting sealants for fuel tanks: An 80% solids sealant with excellent fuel and dry-heat resistance at 450°F was developed from Viton A cured with hexamethylenediamine. A 100% solids formulation
(over)

AD-155 606

based on PAH-1 had excellent resistance to JP-5 fuels at 350°F when cured with Dz_2O_2, but its initial tensile properties were poor. After this exposure, good adhesion was retained to stainless steel primed with either DC-A-4014 or phenolic-triallyl cyanurate. Formulations based on Viton A, LS-53 or 1F4 plasticized with either PAH-1 or reverted LS-53U, had good tensile properties but poor adhesion to stainless steel. Groove sealants from fluorinated polymers: A sealant formulated from LS-53U and 1F4 had the required temperature properties of the proposed 250° specification and was capable of reinjection after prolonged immersion. Development of pressurization sealants: Room-temperature vulcanizing compounds were developed which had outstanding high-temperature properties after a stepwise post cure. Other investigations included: evaluation of fluorinated polyesters, evaluation of Silastic S-2077, recuring of silicones with fuel resistant materials, and inorganic polymers. (See also AD-144 877, AD-151 257)

AD-155 689 Div. 14, 17

Horizons, Inc., Cleveland, Ohio.
A RESEARCH PROGRAM ON THE INVESTIGATION OF SEAL MATERIALS FOR HIGH TEMPERATURE APPLICATION, by Raymond H. Baskey. Rept. for 1 Jan 57-31 Mar 58 on Ceramic and Cermet Materials. June 58, 64p. incl. illus. tables (WADC Technical rept. no. 58-181) (Contract AF 33(616)3891)
Unclassified report

A systematic study was made of the wear behavior of (1) pure refractory hard metals (high-melting hard substances with a metallic character such as CrB_2, CrN, or $CrSi_2$), (2) binary alloys of pure refractory hard metals, bonded with Ni, and (3) ternary alloys of pure refractory hard metals, bonded with Ni and infiltrated with Ag. These materials were screened to determine wear mechanisms at speeds up to 30,000 fpm at loads of 14 psi without liquid lubricant. Such brittle pure refractory hard metals as VB_2, VSi_2, WSi_2, and MoB have fairly low wear rates against both
(over)

AD-155 689

tool steel and Inconel, but are too brittle for most applications. Alloys of Ni and refractory hard metals were not satisfactory. Ternary alloys containing WB or CrN showed excellent low wear rates. Tool steel is limited to temperatures up to 1050°F, and Inconel may be too soft for top temperature performance. A ZrC-coated, unimpregnated graphite seal had good wear properties at 1050°F and sliding speeds of 14,000 fpm against stainless steel type 303. Kentanium K162B when run dry against Kentanium K162B at sliding speeds of 14,000 fpm at 1100°F gave the finest wear performance. (See also AD-87 235)

AD-156 311 Div. 20, 14

Atomic Energy Research Establishment (Gt. Brit.).
AN APPARATUS FOR SEALING ALPHA-ACTIVE SAMPLES IN EVACUATED GLASS AND SILICA CAPSULES, by J. P. Evans and M. B. Waldron. May 57, 4p. illus. (AERE rept. no. M/M 160; HX. 2974)
Unclassified report

An apparatus is described for sealing α-active samples in glass and silica capsules. The capsule is rotated at speeds from $\frac{1}{2}$ to 1 r. p. m. for silica and 30 to 60 r. p. m. for glass; a simple carbon arc makes an annular weld between the full diameter of the capsule and a closely fitting re-entrant closure. The capsule is then evacuated through a tube which forms the other end of the capsule, suitably constricted for sealing by means of another arc. This procedure allows sealing without over-heating the specimen. If desired a thermocouple can be located subsequently in the re-entrant cap for accurate temperature measurement. (Author)

AD-156 416 Div. 14, 26

Detroit Arsenal, Center Line, Mich.
SEALING OF SLIP COVER CONTAINER ASSEMBLIES, by Alexander A. Zadrozny. Final informal rept. 11 Nov 57 [11]p. incl. illus. table (Rept. no. 4017) Unclassified report

A total of twenty-one (21) experimental sealers were tested with reference to subject application under extreme pressure, temperature, and humidity conditions. Of these sealers, numbers 8 and 20 were most versatile for good all-around performance. From a general standpoint, sealers could be classified basically into two categories: those capable of optimum sealing performance, such as numbers 7, 14, 19 and 21; and those easiest to handle (apply and remove) such as numbers 1 and 16. In the former case, sealers 7, 14, 19 and 21 were capable of withholding internal pressures up to 7 psi and could effect a proper closure through temperature extremes of -65 F to 212 F. Sealers 1, 8, 16 and 20 fell somewhat below these (over)

AD-156 416

optimum performance limits. However, considerable flexibility was exhibited in handling which indicated that subject compounds could be readily applied and removed in service. (Author)

AD-156 901 Div. 8

New York U. Coll. of Engineering, N. Y.
SEALING TECHNIQUES FOR WAVEGUIDE AND COAXIAL ROTARY JOINTS, by Milton M. Sackman, Michael B. Maccarone, and Fred H. Posser, Sr, Rept. no. 10 (Final) 1 Aug 54-31 Mar 57. 169p. incl. illus. tables.
(Contract DA 36-039-sc-64476) Unclassified report

Studies were completed on the determination of sealing techniques for waveguide and coaxial rotary joints. A complete discussion of all pertinent design data and related information is included with emphasis on a design approach utilizing techniques obtained in this study. Performance and design features of the developed rotary seal installed in an RG-52 rotary joint are discussed with respect to related parameters and the methods which were applied in reaching a final design meeting the specified requirements. Information is included regarding the work performed in applying the established sealing techniques to 2 existing coaxial rotary (over)

AD-156 901

joints. The appendixes contain the following information: data from the experimental work on the magnetic fluid and the magnetic mechanical seals, history with recorded data on the experimental work performed in the development of the final design, developed design criteria, and final testing data of the development models. (See also AD-134 918)

AD-157 356 Div. 7/1, 14/1, 14/8

Burgess Battery Co., Freeport, Ill.
PLASTIC SEALS, by C. J. Broughton. Quarterly progress rept. no. 2, 1 Sep-30 Nov 57. 10p.
(Contract [DA] 36-039-sc-10527) Unclassified report

See also AD-144 596, AD-157 357

AD-157 357 Div. 7/1, 14/1, 14/8

Burgess Battery Co., Freeport, Ill.
PLASTIC SEALS, by C. J. Broughton. Quarterly progress rept. no. 3, 1 Dec 57-28 Feb 58. 1 Mar 58, 11p.
(Contract [DA] 36-039-sc-10527) Unclassified report

Various thermosetting and thermoplastic materials were investigated. Polyester resins appear suitable for seals for dry cells. The only thermoplastic material that appears suitable for seals is irradiated polyethylene. (See also AD-144 596, AD-157 356)

AD-157 699 Div. 22, 26

Sciaky Bros., Inc., Chicago, Ill.
DESIGN AND FABRICATION OF A PILOT WELDER FOR BONDING AND SEALING THE CLOSURE OF LARGE SIZE TOXIC CHEMICAL MUNITIONS, by Henry James. Progress rept. no. 1, 1-31 Jan 58, 19 Feb 58, 2p.
(Contract DA 18-108-cml-6437) Unclassified report

The objective of this contract is to design and fabricate a Pilot Welder for bonding and sealing the closure of large size toxic chemical munitions. The Pilot Welder will accommodate Chemical and Ordnance type munitions as follows: 105-mm. Shell, M360 and T302; 155-mm. Shell, M121 and M122; 175-mm. Shell, T223; 8-in. Shell, T174; and 4.5-in. Rockets, T164. The Pilot Welder will accommodate such other munitions as are within the size range defined by the preceding types. (Author) (See also AD-157 700, AD-204 463)

AD-157 761 Div. 27/1

Armour Research Foundation, Chicago, Ill.
METALLIC SEAL AND GLAND FOR STATIC SEALING APPLICATIONS IN AIRCRAFT POWER PLANT FLUID SYSTEMS, by S. F. Faunce. Final rept. 22 Mar 55-1 June 56. 51p. incl. illus. tables (WADC Technical rept. no. 56-378)
(Contract AF 33(600)29636) Unclassified report

Results are given for the testing and development of differently shaped metallic seals which are to be used at high temperatures with the Marmam V-band clamp as the flange fastener. The initial phase of this work, the design study, established in a qualitative manner the surface finish and pressure that must be maintained at the sealing surface as a function of the sealing material and the fluid to be sealed. It was found that a truly gas-tight seal with metal seals appears to be extremely difficult to obtain. However, satisfactory gas sealing of the joint was attained for static pressures with the seal developed; (over)

AD-157 761

but the seals failed during the pressure cycling phase of the test. These failures were due to the fatigue stresses in the seal which occurred due to the relative movement of the flanges. The flange movement during cycling was due to movement of the V-retainer of the clamp. Tests indicate that this movement is very much dependent upon the "relative fit" between the V-retainer and the flanges. Recommendations given for further study of metallic seals include a stress analysis of the seals, an investigation of surface phenomena, the determination of maximum amounts of gas leakage to be expected, and further investigation into impregnated fiber metal seals. (Author)

AD-157 911 Div. 14/1, 26/7

Huff, M., and Co., Newark, N. J.
INVESTIGATION OF SEALING QUALITIES OF GREASEPROOF BARRIER MATERIALS, by Leonard C. Drabant. Monthly progress rept. no. 2, Jan 58, 4p.
(Contract DA 28-017-501-ORD-2699, Proj. PA-4-20) Unclassified report

AD-157 912 Div. 14/1, 26/7

Huff, M. and Co., Newark, N. J.
INVESTIGATION OF SEALING QUALITIES OF GREASEPROOF BARRIER MATERIALS, by Leonard C. Drabant. Monthly progress rept. no. 3, Jan 58 [16]p. 1958 incl. illus.
(Contract DA 28-017-501-ORD-2699, Proj. PA-4-20) Unclassified report

A brief summary of evaluations made of the following items is presented: (1) backings; (2) films including cellulosic compounds, nylon, vinyl compounds, polycarbonates, polychlorotrifluoroethylene, polymethylmethacrylate, polyvinyl alcohol, polypropylene, rubber hydrochloride, low- and high-density polyethylenes, styrene (oriented), acrylonitrile styrene, polyvinylbutyral, polyesters, cellophane, and foil; and (3) coatings and adhesives. (See also AD-157 911, AD-157 530)

AD-158 530 Div. 14/1, 26/7

Huff, M. and Co., Newark, N. J.
INVESTIGATION OF SEALING QUALITIES OF GREASEPROOF BARRIER MATERIALS, by Leonard C. Drabant. Monthly progress rept. no. 1, 1958, 4p.
(Contract DA 28-017-501-ORD-2699, Proj. PA-4-20) Unclassified report

See also AD-157 912

AD- 158 661 Div. 14, 22

Borden Co., Philadelphia, Pa.
INVESTIGATION OF THREADED JOINT SEALANTS AND CRIMPED JOINT SEALERS, by Wolf Karo. Monthly rept. no. 14 (Final) 30 Dec 55-31 Oct 57. 10p. incl. illus. tables.
(Contract DAI 28-017-501-ORD(P)2043, Proj. nos. PA-9-11A and PA-9-11-D) Unclassified report

Attempts were made to supply formulations of sealants for crimped and threaded joints on fuzes. Early experiments indicated that the pot life of polysulfide sealants could be extended by modification of the accelerator mix, although this modification made the preparation of a uniform composition difficult. The breakout torques obtained was about 35 in.-lb, calculated on the basis of a 1-1/8-in. bolt diameter. Studies were made of MPL-monomer thread sealants. Primary efforts involved developing additives to MPL-monomer and new monomers related to MPL that would reduce the breakout torque, and testing various (over)

AD-158 661

compositions both after aeration and after cure by a highly reactive initiator system. Three methods of polymerization of MPL monomer were studied: (1) self-sealing polymerization, (2) a process in which MPL-monomer compositions are both activated and inhibited by an air stream, and (3) by incorporating various polymeric materials in MPL-monomer.

AD-159 156 Div. 14/1, 14/6, 14/8, 22/5

Aerojet-General Corp., Azusa, Calif.
DEVELOPMENT OF AN IMPROVED BOMB SEALER FOR ELIMINATION OF INERT-SEALER EXUDATION, by R. D. McLeod and J. D. Sohl. Addendum 1 to Final rept. for 13 Jan-31 Mar 58. [16]p. incl. illus. (Addendum 1 to rept. no. 1282)
(Contract DA 28-017-501-ORD-2115) Unclassified report

A new end-sealing compound of polyurethane foam was developed and subjected to rigid temperature-cycling tests. The sealing material was loaded in bombs filled with TNT and compared with other materials. Of the materials tested, only Celite (diatomaceous earth) and the polyurethane foam prevented exudation. Two bombs sealed with polyurethane foam and 2 bombs sealed with Celite were subjected to drop tests from a 40-ft tower onto 6-in.-thick steel resting on 24 in. of concrete. Each of the 4 bombs was dropped from 3 positions: horizontal, base down, and nose down. All drops were satisfactory, without detonations, and only normal damage (over)

AD-159 156

to the bomb casings was experienced. (See also AD-140 572)

AD-160 124 Div. 8/2

Acton Labs., Inc., Mass.
DEVELOPMENT OF HIGH TEMPERATURE FILM TYPE ROTARY POTENTIOMETERS, by Robert Alger. Scientific rept. no. 3, 1 Dec 57-28 Feb 58. 1v. incl. illus. table.
(Contract AF 33(616)3977) Unclassified report

Efforts were concerned with the potentiometer design and materials selection. The design was finalized, and 5 potentiometers were built. The design greatly reduced the problems of ganging potentiometer units; all units are identical except for the second shaft (for rotating the hollow shaft) which comes in various lengths depending on the number of units in the gang. Three potentiometers were placed on high-temperature stability test. Results, after about 300 hr, were satisfactory. Five films (70% of Rh and 30% Ge, by weight) were evaporated to investigate a high contact noise characteristic. Results indicated that inadequate control existed during deposition. With the elimination of this charac- (over)

AD- 160 124

teristic, the film passed all contract requirements. A Silastic RTV-501 material proved satisfactory for use in the capsule contact; the Silastic formed an airtight seal while withstanding temperatures up to 260°C without deterioration. (See also AD-150 247)

AD- 160 402 Div. 31, 14

Material Lab., New York Naval Shipyard, Brooklyn.
DECK CALKING DEVELOPMENT PROGRAM (NS 033-200) SUBTASK 2. Final rept. [18 Nov 57] [1]p. (Lab. proj. no. 4969-45) Unclassified report

Available from ASTIA. An expanded version is available to U. S. Military Organizations only as AD-160 452.

160452
AD- ~~160 402~~ Div. 31, 14

Material Lab., New York Naval Shipyard, Brooklyn.
DECK CALKING DEVELOPMENT PROGRAM (NS 033-200) SUBTASK 2. Final rept. [18 Nov 57] [1]p. (Lab. proj. no. 4969-45) Unclassified report

Available from ASTIA. An expanded version is available to U. S. Military Organizations only as AD-160 452.

161 633
AD- ~~161 694~~ Div. 14, 27

Material Lab., New York Naval Shipyard, Brooklyn.
EVALUATION OF O-RINGS USED TO SEAL CYLINDER LINERS OF DIESEL ENGINES SUBMITTED BY PRECISION RUBBER PRODUCTS CORPORATION, DAYTON 7, OHIO, by S. A. Eller. Final rept. 18 Feb 58, 9p. illus. tables (Lab. proj. no. 5354-13) Unclassified report

Available from ASTIA. An expanded version is available to U. S. Military Organizations only as AD-161 638.

AD-161 694 Div. 14, 27

Material Lab., New York Naval Shipyard, Brooklyn.
EVALUATION OF O-RINGS USED TO SEAL CYLINDER LINERS OF DIESEL ENGINES SUBMITTED BY PRECISION RUBBER PRODUCTS CORPORATION, DAYTON 7, OHIO, by S. A. Eller. Final rept. 18 Feb 58, 9p. illus. tables (Lab. proj. no. 5354-13) Unclassified report

Available from ASTIA. An expanded version is available to U. S. Military Organizations only as AD-161 638.

AD-161 745 Div. 30, 14, 27

Material Lab., New York Naval Shipyard, Brooklyn.
COMPRESSION STRESS RELAXATION APPARATUS, DESCRIPTION OF APPARATUS AND PROCEDURE FOR INCLUSION IN A SPECIFICATION FOR DIESEL ENGINE O-RING LINER SEALS. Final rept. 8 May 57, 3p. illus. (Lab. proj. no. 5354-12) Unclassified report

Available to U. S. Military Organizations only

AD-161 932 Div. 26, 14

Directorate of Materials and Explosives Research and Development (Gt. Brit.).
THE STRENGTH OF THREADED JOINTS: PART I: RESISTANCE TO MECHANICAL SHOCK, by R. W. Bryant and W. A. Dukes. May 58 [11]p. incl. illus. tables (Rept. no. DMXRD SEAL '58 1; [JSRP Control no. 580621]) Unclassified report

The strength (that is, the failing torque) of a cemented threaded joint is the sum of the tightening torque and the contribution of the cement. The resistance of steel threaded joints (assembly weighing 2 lb; thread 1 inch in diameter) to a single axial impact when dropped 4 ft. 6 in. on to concrete has been investigated. For joints screwed on to the external shoulder: (i) a tightening torque of 30 ft lb is necessary to resist one impact if there is no sealing composition in the joint, (ii) a tightening torque of 10 ft lb is necessary is the joint is luted and (iii) less than 10 ft lb (perhaps nil) is neces- (over)

AD-161 932

sary if the joint is cemented. Similar, but perhaps higher, tightening torques are required if the joint is screwed on to the internal shoulder. (Author) (See also AD-123 604, AD-146 589)

AD-161 951 Div. 7, 14

Burgess Battery Co., Freeport, Ill.
PLASTIC SEALS, by C. J. Broughton. Quarterly progress rept. no. 4, 1 Mar-31 May 58. 18 June 58, 13p.
(Contract DA 36-039-57-10527) Unclassified report

Investigations of the material "gunk" and the thermoplastic material "Irratiated Polyethlene" as suitable seals for dry cells has been carried. It was decided that the effort would be concentrated on the thermosetting material "gunk". A mold was made for 'N' size seals and seals were molded using the polyester "gunk". 'N' size cells were than fabricated using these seals. The majority of the seals failed, due to cracks which developed in the molded "gunk" material, after aging for a few days. Good cells were selected and stored at 130°F. The spinning tools were used in making the 'N' cells and proved very effective with no difficulty encountered. (Author) (See also AD-157 357)

AD-161 971 Div. 20, 16, 26
(Unannounced report)
Massachusetts Inst. of Tech. [Div. of Sponsored Research] Cambridge.
PROCUREMENT OF SERVICES PERTINENT TO THE DETERMINATION OF BACTERIAL RESISTANCE OF PACKAGES, by B. E. Proctor. Annual rept. 26 Sep 56-30 Sep 57, on Investigation of Bacterial Resistance of Packages. 11p. tables.
(Contract DA 19-129-qm-758) Unclassified report

See also AD-200 862

AD-162 329 Div. 22

Frankford Arsenal, Philadelphia, Pa.
MODIFICATION OF DESIGN OF GASKETS USED IN BOXES, AMMUNITION, CALIBER 30, M19A1, T38E1 AND T38E2, by William J. Rose. Jan 58, 21p. incl. illus. tables (Proj. no. TS1-2; Rept. no. 103) Unclassified report

Not releasable to Foreign Nationals

Method: Gaskets employing three sizes of grooves were compared with solid gaskets for load vs. compression characteristics, compression distribution and environmental storage characteristics when inserted in caliber .30, T38E1 ammunition boxes. Conclusions: A grooved gasket, when used in place of a solid gasket, improves load versus compression characteristics and compression distribution without affecting the environmental storage characteristics of caliber .30, T38E1 ammunition boxes. Recommendations: (1) A gasket employing a 1/8 inch wide by 1/16 inch groove on the underside (over)

AD-162 329

be specified for use in caliber .30, T38E1, T38E2 and M19A1 ammunition boxes. (2) The groove shape shall be either semicircular or rectangular with corner radii. (3) A grooved gasket be utilized in all future ammunition box designs employing a gasket in the cover. (Author)

AD-162 472 Div. 31, 14

Materials Labs., Puget Sound Naval Shipyard, Bremerton, Wash.
OBSERVATIONS ON THE PROPELLER HUB OIL LEAKAGES ON THE USS WINDHAM COUNTY (LST-1170) NAD THE USS WESTCHESTER COUNTY (LST-1167), by W. B. Lew. Mar 58, 21p. illus. (Technical rept. no. R-367) Unclassified report

Notice: Only Military Offices may request from ASTIA.

It is concluded that: (1) Although "O" ring dimensions are reported in nominal or AN-1227 sizes, it is preferred that actual size plus tolerances and stock numbers be used on plans. (2) Checking "O" ring dimensions, width or cross-section diameter, and circumference (O.D.) is necessary to insure proper fit into the groove; because a seal, to be effective against internal pressure, requires that the O.D. of the "O" ring be equal to the O.D. of the groove. (3) "O" ring, piece 71, and blade sealing ring, piece 521, are (over)

AD-162 472

vulnerable to oil leakage from the hub. (4) Before assembly, final check of the clearances involving the retainer plate, thrust ring, and crank ring may be made by measuring the dimensions S_1, S_2, $S_1 + S_2$, and S_3 as described in paragraph on "Assembling Procedures for Components that Affect Hub Sealability". (5) Teflon-coating of the retainer plate, piece 47, does not appear to be effective in reducing wear of the latter in the installation on the USS WESTCHESTER COUNTY. (Author)

AD-162 475 Div. 31, 14

Material Labs., Puget Sound Naval Shipyard, Bremerton, Wash.
EVALUATION OF BLADE SEALS FOR CONTROLLABLE PITCH PROPELLERS, by W. B. Lew. Nov 57, 8p. illus. tables (Progress rept. no. R-369) Unclassified report

Notice: Only Military Offices may request from ASTIA.

Conclusions: The laboratory tests indicate that a seal cavity clearance of 0.375 inch would probably cause rapid wear of the blade seal with the design "A" because that blade seal stalled the rotation of the test apparatus indicating that excessive frictional resistance or drag would develop between the rubber lip and the retainer plate of the hub assembly. Enlarging the seal cavity clearance in a service installation, as recommended by the supplier "A" to between 0.390 and 0.410 inch, is not desirable unless that blade seal is finally adopted as the standard blade seal. Meanwhile, any (over)

AD-162 475

enlargement of the seal cavity would result in excessive clearance for the blade seals of the designs "B" and "C". (Author)

AD-162 479 Div. 31, 14

Material Labs., Puget Sound Naval Shipyard, Bremerton, Wash.
EVALUATION OF NO. 57-0007-8 DECK SEALER SUBMITTED BY PRESSTITE ENGINEERING COMPANY, by J. R. Freeborn. Mar 58, 3p. illus. (Rept. no. R-382) Unclassified report

Available to U. S. Military Organizations only

The calking material has possible application for calking decks of minesweepers if there is a practical way of mixing and applying such a quick-set type of compound. (Author) (See also AD-145 501)

AD-162 481 Div. 31, 14

Material Labs., Puget Sound Naval Shipyard, Bremerton, Wash.
LATEX UNDERLAY FOR FLIGHT DECKS ON USS LEXINGTON (CVA-16) by W. H. Miles and D. S. Peterson. Apr 58, 8p. illus. (Development rept. no. E-379) Unclassified report

Available to U. S. Military Organizations only

A latex underlayment was installed as a sealer and filler under flight deck planking in areas undergoing alterations. Installation methods, coverage weight and costs were considered. The use of underlayment appears to be a definite improvement in eliminating leaks and water pockets. In areas not undergoing repairs, a gunning compound was used to seal leaks. (Author) (See also AD-65 188)

AD-162 726 Div. 30

Naval Research Lab., Washington, D. C.
A NEW THERMAL CONDUCTIVITY LEAK DETECTOR AND ITS APPLICATIONS, by Clarke C. Minter. 29 May 58, 6p. incl. illus. table (Rept. no. 5150)
Unclassified report

The report points out the reasons for the failure of the Pirani tube as a sensitive leak detector and describes a simple thermal conductivity apparatus of special design having several advantages over the mass spectrometer so widely used for this purpose. Procedures are described for using the new apparatus to locate leaks in vacuum or pressure systems using hydrogen or helium as probe gas. Experiments to determine the sensitivity of the apparatus toward small changes in concentration of Freon in air are described, and it has been concluded that leaks in refrigeration equipment can be located even when appreciable concentrations of Freon are present in the ambient air. (Author)

AD-162 882 Div. 26, 14

[Air Technical Intelligence Center]
Wright-Patterson Air Force Base, Ohio.
SEALING OF HYDRAULIC UNITS WITH METALLIC SLOT RINGS (Uplotneniye Gidravlicheskikh Agregatov Razreznymi Metallicheskimi Kol'tsami) by T. M. Bashta. [1958] 9p. incl. tables (Trans. no. F-TS-9397/III of Vestnik Mashinostroyeniya, No. 9, 1957, pp. 20-24)
Unclassified report

Metallic slot piston rings, which have found wide application in various machines, prove to be the simplest and longest-lasting of all sealings used in hydraulic units. This metallic (most often made of cast iron or bronze) ring with a rectangular cross section and a joint of arbitrary configuration is fitted into a groove cut into the surface of the piston. The advantage of such a sealing is the cheapness and adaptability to work in the heaviest of hydraulic systems under temperature conditions, and also the long period of service, which
(over)

AD-162 882

runs to many years for work in these systems. The principle drawback is the need for precise machining of the opening of the cylinder. The difficulty is that piston rings are not free to compensate, as are soft elastic packings, for local irregularities in the surface or variations from the geometrical form of the cylinder towards oval, cone, or barrel-shaped forms. Another shortcoming is the impossibility of obtaining complete hermeticity, principally because the sealing ring itself has play at the joint which needs packing. (Author)

AD-163 020 Div. 14, 12,

Jet Propulsion Lab., Calif. Inst. of Tech., Pasadena.
A REVIEW OF SEAL MATERIALS FOR GUIDED-MISSILE APPLICATIONS, by Robert J. Berg. 7 Mar 58, 16p. incl. tables (Progress rept. no. 20-340)
(Contract DA 04-495-ORD-18, ORDCIT Proj.)
Unclassified report

A survey was made of the available seal materials to select those most usable for present and immediate future needs in guided-missile programs. The development was undertaken of a series of laboratory or bench-type tests which might be used to evaluate new seal materials. The plastic materials studied included (1) polyethylene, (2) Teflon, (3) Kel-F, (4) nylon, and (5) polyvinyls. Rubber-like materials studied included (1) GR-S, (2) natural rubber, (3) Buna-N, (4) silicon rubber, (5) neoprene, (6) butyl rubber, and (7) Hypalon. The change in the physical properties of seal materials as a result of exposure to various substance was measured. Tests are given which are intended to serve
(over)

AD-163 020

as a guide to the selection of O-ring seal materials. The tests include: (1) tensile strength and elongation of O-rings, (2) compression set under constant deflection, (3) determination of O-ring cross section and volume, and (4) property change in liquids.

AD-200 028 Div. 22/1, 22/8, 25/8

Cornell Aeronautical Lab., Inc., Buffalo, N. Y.
RESEARCH ON HEAT TRANSFER AND OBTURATION FOR SMALL ARMS WEAPONS, by Gerald A. Sterbutzel. Monthly progress rept. no. 34, for June 58. 15 July 58, 52p. incl. illus. tables (Rept. no. GA-948-C-34)
(Contract DAI 30-115-504-ORD(P)-596)
Unclassified report

Series 10, 12, and SAR-1 ring seals were tested. Series 10 seals were in satisfactory condition after almost 10,000 rounds of firing. Tests were completed on the first set of series 12 rings seals. The performance of the seals was satisfactory from a temperature standpoint. Tests were conducted on the SAR-1 ring seals. The SAR-1 drum permits the use of an insert in the present production drums. The insert is installed so that the rear edge of the insert is behind the leading edge of the case. The insert in this situation is pinned in place and between the insert and the drum is bonded
(over)

AD-200 028

an aluminum filled epoxy resin. This drum was instrumented for temperature readings, and 16 tests were completed. The temperature rise at the front of the instrumented round in the fuse detonator for this configuration equalled in the first tests the cookoff susceptibility for a regular nonretrofitted drum.

AD-200 082L Div. 14, 7

Aeronautical Material Lab., Naval Air Material Center, Philadelphia, Pa.
PROPOSED REVISION OF MILITARY SPECIFICATION MIL-S-8516B, SEALING COMPOUND, SYNTHETIC RUBBER, ACCELERATED (FOR ELECTRIC CONNECTORS AND ELECTRIC SYSTEMS) by C. Nadler and J. F. Turner. Partial rept. 31 Mar 58, [23]p. incl. illus. (Proj. TED no. NAM AE-42127; Rept. no. NAMC AML AE-1055) Unclassified report

Submit request via Bureau of Aeronautics, Navy Dept., Wash. 25, D. C.

An investigation was conducted for the purpose of revising Specification MIL-S-8516B to cover compounds having a variety of cure rates and to rectify any existing deficiencies or discrepancies in the specification. Sample compounds were obtained from presently qualified manufacturers. Based on the test results obtained with these materials, requirements for fast, dium
(over)

AD-200 082L

and slow curing type compounds were developed. Functional characteristics of these compounds are equivalent to those of present Specification MIL-S-8516B materials and in addition, the compounds possess improved working properties. Two suppliers for fast type compounds and one for the medium and slow type are available. (Author)

AD-200 230 Div. 7 14

Material Lab., New York Naval Shipyard, Brooklyn.
DEVELOPMENT OF CABLES, TYPE TCTX-3 UTILIZING VARIOUS SEALERS SUBMITTED BY ANACONDA WIRE AND CABLE COMPANY, HASTINGS-ON-HUDSON, N. Y. by S. S. Pigman. Final rept. 10 Apr 58, 6p. tables (Lab. proj. no. 5845-6)
Unclassified report

Tests were conducted on type-TCTX-3 cables to determine (1) the effects of the use of silicone-based and duxseal-type sealants on cable performance and (2) the effects of armor paint on silicone rubber cable sheathing. One sample contained a duxseal-type sealer, (Code A); and the other sample was filled with a silicone based sealer, (Code B). Painted specimens were coated with aluminum paint, (Code C). No significant difference in performance under the gas flame was indicated between samples sealed with Code A or B sealants, or between painted and unpainted cables. Exposure in the air convection oven for various periods up to 96 hr at 500° F
(over)

AD-200 230

caused little change in the insulation resistance of both painted and unpainted specimens filled with a Code B sealant. Both the painted and unpainted specimens filled with a Code A type sealant showed an increase in insulation resistance. The duxseal-type (Code A) material was unsatisfactory for TCTX cables. Prolonged aging did not indicate any further reduction in the physical properties of the sheath material after the first 24 hrs in either painted or unpainted samples. Painted samples containing the Code A filler showed the greatest deterioration after aging. Armor paint applied directly upon the outer surface of the sheath produced very little change in the physical properties after 1 mo at room temperature.

AD-200 381 Div. 1

Kett Corp., Cincinnati, Ohio.
SELF SEALING TANK PROGRAM, by W. F. Savage and M. G. Ginzberg. Summary rept. 29 Apr 55, 1v. incl. illus. table (Rept. no. KX-37)
(Contract AF 33(600)27046) Unclassified report

This report summarizes work done to develop self-sealing fuel cells of other than the conventional chemical self-sealing construction. An initial theoretical analysis evaluated sixteen methods and variations thereof for obtaining mechanical seals. Eight of these designs were subjected to preliminary gunfire tests and/or static tests. Two methods, flexible flapper valves and a spring actuated curtain, showed most promise for future development and these devices were subjected to additional static and gunfire tests. The curtain demonstrated ability to seal gunfire wounds on flat surfaces to a slow seep. In the final gunfire tests the flexible flapper valves failed to achieve satisfactory seals. (Author)

AD-200 382 Div. 1, 14

Kett Corp., Cincinnati, Ohio.
SELF-SEALING TANK PROGRAM, by W. F. Savage and M. G. Ginzberg. Summary rept. on Phase 1. 5 Oct 54, 1v. incl. illus. tables (Rept. no. KX-31)
(Contract AF 33(600)27046) Unclassified report

Designs for construction of self-sealing fuel cells of other than conventional self-sealing construction were studied. The program followed included an initial theoretical analysis (of sixteen possible designs), preliminary gunfire tests of small panels, static tests, and further gunfiring of test panels. Two constructions were evolved which show promise of satisfactory sealing characteristics. Design and construction of fuel cubes embodying these two methods is recommended. (Author)

AD-200 385 Div. 1

Firestone Tire and Rubber Co. of Calif. Los Angeles.
HIGH TEMPERATURE EFFECTS ON FUEL CELL MATERIALS, by John L. Dum. Final rept. for Nov 55-July 56 [on Phase 1]. Aug 56, 75p. incl. illus. tables. (WADC Technical rept. no. 56-428)
(Contract AF 33(616)31288) Unclassified report

A study was made of the effects of high temperature on the fuel container properties of self-sealing, bladder (E-107A), and nonrip tanks (NR-1173) and fittings. Experiments were performed at 200°, 250°, 300°, and 500°F to determine tensile strength, puncture resistance, sealant swelling rates, flex cracking resistance, diffusion of fuel, and fitting torque. The tensile strength dropped gradually with increasing aging temperatures up to 350°F. The most serious result was a stiffening and eventual embrittlement of the coating materials which eventually become so severe that any flexing produced a stock rupture which would permit
(over)

AD-200 385

fuel leakage. At 450°F, the tensile strength dropped faster with aging than stock embrittlement. Swell rates were mildly affected by sealant heat exposure. Diffusion rates showed little change except at elevated temperatures where they increased rapidly. Aging tests with JP-5 fuel showed that service life of the materials was longer than that indicated by hot air aging tests. (See also AD-200 386)

AD-200 850 Div. 3, 22

Army Chemical Corps Engineering Command, Army Chemical Center, Md.
INSTALLATION AND OPERATION OF LINE FOR THE REHABILITATION OF DEFECTIVE SHELL, CHEMICAL, GB, 155mm, M121, by Abraham Rednor, Thomas F. Bradley, and Joseph S. Carrion. June 58, 45p. incl. illus. tables (Rept. no. ENCR-23)
Unclassified report

This report describes the installation and operation of equipment, conveyors and associated inspection facilities for reworking defective M121 chemical shell bodies by machining fresh radii at the press-fit shoulder and honing to remove superficial imperfections in the shell press-fit surface. Magnetic particle inspection was included to screen out shell with sub-surface forging cracks. The program is covered from initial organization and preliminary engineering phases through design, installation and operation of the line and process improvements incorporated to increase recovery rate of
(over)

AD- 200 850

reworked shell. Photos, drawings and operating instructions are included in the appendixes for further clarification of the extent of the project. (Author)

AD-200 862 Div. 29, 16, 28

Massachusetts Inst. of Tech. [Div. of Sponsored Research] Cambridge.
PROCUREMENT OF SERVICES PERTINENT TO THE DETERMINATION OF BACTERIAL RESISTANCE OF PACKAGES, by B. E. Proctor. Progress rept. no. 7, 1 Oct-30 Nov 57. 6p. tables.
(Contract DA 19-129-qm-758) Unclassified report

An investigation was made of the frequency of penetration of flat sheets of film (plain, creased and 1/8" heat seals) by Monascus purpureus. Statistical analysis of the data showed no significant difference in the frequency of penetration by M. purpureus of unirradiated and irradiated films that were either heat sealed, plain or creased. A significant difference (5%, 1%, or 0.1% level) was found when results for M. purpureus were compared with similar results for Serratia marcescens, Rhodotorula rubra and Escherichia coli. It was found that there was less penetration of laminated films sealed with a packaging
(over)

AD-200 862

industries' heat sealer than of laminated films sealed with a Vertrod sealer. Tests were carried out with Clostridium sporogenes to determine the frequency of penetration of flat sheets. Preliminary work was carried out with Fusarium culmorum. (Author)
(See also AD-161 971)

AD-200 898 Div. 14, 31

Directorate of Materials and Explosives Research and Development (Gt. Brit.).
THE SEALING OF THREADED JOINTS. PART 2. THERMAL CYCLING AND MECHANICAL SHOCK, by R. W. Bryant and W. A. Dukes. July 58 [20]p. incl. illus. tables (DMXRD rept. no. SEAL 58/2; [JSRP Control no. 581054]) Unclassified report

The effect of mechanical stresses caused either by cyclic changes of temperature, or by impact, on the sealing of various threaded joints has been determined. Thermal cycling does not affect the sealing of joints unless the components are made of dissimilar metals. The most important single factor is the joint design (external, internal, or no shoulder), coupled with mode of application of sealing composition. The next most important factor is the quality of the sealing composition. R.D. 1286 is the best cement, and R.D. 1284 (Luting Mk. 8) the best luting, examined. There is no significant difference between the performance of joints
(over)

AD-200 898

made of different bimetallic combinations. A single axial impact caused by dropping steel threaded assemblies "end-on" through 4 ft. 8 in. on to concrete has little effect on the sealing. (Author) (See also AD-199 095)

AD-201 024 Div. 26, 17, 31
(Unannounced report)

Naval Engineering Experiment Station, Annapolis, Md.
SYNTRON CARTRIDGE UNIT - FAILED SEAL RING OF HASTELLOY B, by Martin W. Rausch. 2 Oct 52, 3p. incl. illus. (EES rept. no. 4G(2)101717)
Unclassified report

Failure of a Hastelloy B face ring contained in a Syntron cartridge unit for 5-5/8" diameter shaft was cause for comparative corrosion tests to be made of discs of Hastelloys B and C. Sample discs in contact with graphite washers were exposed in Severn River water and in sea water. No appreciable damage was sustained by Hastelloy C as a result of these tests. Discs of Hastelloy B, however, showed susceptibility to contact corrosion and also suffered corrosion attack and roughening beyond the contact area. The surface roughening of the test samples was similar to that exhibited by the contact face of the seal ring. (Author)

AD-201 047 Div. 14
(Unannounced report)

Westinghouse Electric Corp., Essington, Pa.
FAWICK ROTORSEAL TEST, by E. F. Millar. 28 Nov 49, 2p. incl. illus.
(Contract NObs 34234, Proj. Wolverine)
Unclassified report

The Fawick Rotorseal designed for a maximum speed of 2500 r.p.m. and air pressure of 150 p.s.i.g. was tested at 10,000 r.p.m. and 200 p.s.i.g. The packing grease heated in few min. and extruded from the bearing. After 10-min. operation with pressure reduced to 50 p.s.i.g. the grease was smoking. A thermocouple inserted in the housing to indicate abnormal temperatures was used as an index of deterioration of the moving parts. Another test under similar conditions but using a continuous lubrication resulted in overheating after 39 3/4 hrs. The Rotorseal is not recommended for applications in the range of the test conditions without auxiliary cooling and continuous lubrication.

AD-201 085 Div. 7, 8
(Unannounced report)

Aeronautical Electronic and Electrical Lab., Naval Air Development Station, Johnsville, Pa.
TEST OF LIMIT SWITCH SEALS ON TYPE X2875 METAL BELLOWS SEALED SWITCHES, by H. B. Sullivan. 26 Aug 48, 5p. illus. (TED no. ADS EL-602; AEEL rept. no. 4-49) Unclassified report

Aircraft switches X2875 and BZE7RNT2 were tested simultaneously in accordance with specifications AN-N-9 and AN-S-50. The following tests were carried out: ozone resistance, oil aging, dust and sand, altitude and temperature, icing, and condensation. The switches were actuated to simulate service conditions except during the ozone resistance and oil-aging tests. Type X2875 switch is superior to the standard BZE7RNT2. The metal bellows of the X2875 should be sealed to prevent ice and foreign particles from accumulating in the bellows pleats. The rubber gasket of the cover plate
(over)

AD-201 085

provides sufficient sealing; no additional sealing is required at any conduits. (Author)

AD-201 201(serial) Div. 14, 26

Huff, M. and Co., Newark, N. J.
INVESTIGATION OF SEALING QUALITIES OF GREASEPROOF BARRIER MATERIALS, by Leonard C. Drabant. Monthly progress rept. nos. 4-6 [1958]. 3 nos.
(Contract DA 28-017-501-ORD-2699, Proj. PA-4-20) Unclassified reports

Unabstracted reports with this title include:
Monthly progress rept. no. 4, [1958], AD-201 201
Monthly progress rept. no. 5, [1958], AD-201 202
Monthly progress rept. no. 6, [1958], AD-201 203
(See also AD-157 912, AD-201 204, AD-202 445)

AD-201 202 Div. 14, 26
(Unannounced report)

Huff, M. and Co., Newark, N. J.
INVESTIGATION OF SEALING QUALITIES OF GREASEPROOF BARRIER MATERIALS, by Leonard C. Drabant. Monthly progress rept. no. 5. [1958] 3p.
(Contract DA 28-017-501-ORD-2699, Proj. PA-4-20) Unclassified report

Collected on AD-201 201

(See also AD-201 201)

AD-201 203 Div. 14, 26
(Unannounced report)

Huff, M. and Co., Newark, N. J.
INVESTIGATION OF SEALING QUALITIES OF GREASEPROOF BARRIER MATERIALS, by Leonard C. Drabant. Monthly progress rept. no. 6 [1958] 7p. incl. tables.
(Contract DA 28-017-501-ORD-2699, Proj. PA-4-20) Unclassified report

Collected on AD-201 201

AD-201 204 Div. 14, 26

Huff, M. and Co., Newark, N. J.
INVESTIGATION OF SEALING QUALITIES OF GREASEPROOF BARRIER MATERIALS, by Leonard C. Drabant. Monthly progress rept. no. 7. [1958] 11p. incl. tables.
(Contract DA 28-017-501-ORD-2699, Proj. PA-4-20) Unclassified report

Samples were made of barrier combinations of 0.00035-in.-gage Mylar and cotton with thread counts 44 x 40, 28 x 26, and 56 x 44. These were coated with the N-1 coating previously used on the M-6 barrier. Five additional production samples of the M-6 type barrier were made to check for reproduceability. Two hundred and fifty samples of barrier combinations M-6, M-21, M-22, and M-23 were tested by the Picatinny Arsenal adhesion test method to determine the effects of different quantities of coating. Sixty adhesion tests were run on M-6 reproduction. Testing of Mylar and acetate combinations for establishing the correlation of WVTR/lb of wax
(over)

AD-201 204

was completed. Quick leak tests (138) were conducted on method IA-1 and IA-2 packages both dipped and undipped and from a variety of materials. Further evaluations of various microcrystalline waxes and wax blends were conducted. A new wax blend, which indicated promising results, was used in the production of a sample barrier material for tests of effectiveness as compared to that of the N-1 coating used on the M-6. (See also AD-201 203)

AD-201 273 Div. 26

Feltman Research and Engineering Labs., Picatinny Arsenal, Dover, N. J.
PROPOSED NEW SEAM STRENGTH TESTS FOR MIL-B-117A INTERIOR PACKAGING BAGS, by Richard H. DeVore. Aug 58, 17p. incl. illus. tables. (Proj. no. TB4-006G; Technical rept. no. 2534) Unclassified report

Available from ASTIA. An expanded version is available to U. S. Military Organizations only as AD-202 446.

Seams made from various types of heat-sealable barrier materials were tested at room temperature and at 100°F and 160°F to determine whether or not a 5-minute room temperature test could be substituted for the usual 1-hour elevated temperature test. In the 5-minute test, only 2 weights were used instead of the several weights required by the conventional tests. In this test the weights were suspended from the test specimens so that
(over)

AD-201 273

the load would be applied directly on fin-type seams fabricated from the test materials. At least 5 specimens of each material, cut from the material as received, were used in each phase of the tests. All of the weights used (3.5 lb, 2 lb, 24 oz, 12 oz, 10 oz, 7.5 oz, and 3.75 oz) were suspended from the specimens by means of clamps. The 5-minute room temperature test was found satisfactory when a 3.5-lb weight was used for Class b, c, and e bag material specimens (Specification MIL-B-117A) and a 2-lb. weight for Class d (MIL-B-117A) bag material specimens. (Author)

AD-201 321 Div. 8, 26

Bulova Research and Development Labs., Inc., Woodside, N. Y.
AUTOMATIC MACHINE FOR SEALING OF QUARTZ CRYSTAL UNITS, by R. Giglio Tos. Quarterly progress rept. no. 4, 25 Mar-24 June 58. [11]p. incl. illus.
(Contract DA 36-039-sc-72775) Unclassified report

Tooling difficulties on the HC-6/U holder were resolved. A Raytheon #1103 DC welder and Model K welding head were ordered. Evacuation chamber and controls are being assembled. Test samples of HC-6/U holders welded using the DC welder were evaluated. Welding of units using AC welder was not fully satisfactory. Photomicrographs of a weld and an expression for years of "seal life" are shown. (Author) (See also AD-157 394, AD-157 395, AD-158 639)

AD-201 739 Div. 13

Army Engineer Research and Development Labs., Fort Belvoir, Va.
SLEEVE-TYPE MECHANICAL SEALS FOR DIRECT-DRIVEN, MULTI-STAGE, PETROLEUM PIPELINE PUMPS, by Bryant L. Johnson. 2 June 58, 40p. incl. illus. (Rept. no. 1527-TR) Unclassified report

The investigation, selection and testing of a sleeve-type mechanical seal of commercial design for Corps of Engineers, direct-driven, petroleum pipeline pumps are covered by this report. Mechanical seal manufactures were invited to submit proposals on a seal design to meet requirements established by the Petroleum Equipment Branch. Crane Co., Chicago, proposed designed which was modified to evolve the seal that was fabricated and tested. The report concludes in part that: (a) the seal is suitable for Corps of Engineers, direct-driven, petroleum pipeline pumps; (b) this type seal is suitable for use in other similar pumps for petro- (over)

AD-201 739

leum distribution service; (c) a substantial reduction and simplification in spare seal parts stocks will be possible by the adoption of this seal. (Author)

AD-201 835 Div. 14

Pitman-Dunn Labs. Group, Frankford Arsenal, Philadelphia, Pa.
SEALING OF ANODIZED ALUMINUM, by F. Pearlstein. May 58, 11p. incl. illus. tables (Proj. no TB4; Memo rept. no. MR-671) Unclassified report

A number of sealing treatments utilizing boiling aqueous solutions and/or dispersions were applied to anodized 24S-T3 aluminum panels. The treated panels were coupled with copper discs and exposed to salt spray. Dichromate-sealed panels exhibited higher resistance to galvanic corrosion than any of the other singly sealed panels. Several double sealing systems, incorporating dichromate for either the first or second seal, increased the corrosion resistance over panels which were dichromate-sealed only. Some improvement was achieved by presealing in water or post-sealing in waterglass, but more significant improvement was obtained with panels presealed in 0.5 per cent Antarox A-400 or 0.5 per cent nickel acetate. Some of the sealing (over)

AD-201 835

treatments considerably increased the lubricity of the surface of anodized panels. Five-minute immersion in hot Teflon, polyethylene, wax, or diglycol stearate dispersions decreased the coefficient of friction of the surface by 75 to 88 per cent. (Author)

AD-202 438 Div. 22, 25

Cornell Aeronautical Lab., Inc., Buffalo, N. Y.
RESEARCH ON HEAT TRANSFER AND OBTURATION FOR SMALL ARMS WEAPONS, by Gerald A. Sterbutzel. Monthly progress rept. no. 32 for Apr 58. 15 May 58, 59p. incl. illus. tables (Rept. no. GA-948-C-32)
(Contract DAI 30-115-504-ORD(P)-596)
Unclassified report

Tests are discussed which were conducted on series 10 ring seals, SA-2 ring seals, and series 11 ring seals. Each has an outside diameter of 1.289 in. and uses the standard washer-type seal retainer. (See also AD-158 762)

AD-202 439 Div. 22, 25

Cornell Aeronautical Lab., Inc., Buffalo, N. Y.
RESEARCH ON HEAT TRANSFER AND OBTURATION FOR SMALL ARMS WEAPONS, by Gerald A. Sterbutzel. Monthly progress rept. no. 33 for May 58. 15 June 58, 31p. incl. illus. tables (Rept. no. GA-948-C-33)
(Contract DAI-30-115-504-ORD(P)-596)
Unclassified report

The study of ring seals was continued for the optimization of heat transfer and durability characteristics. Testing was confined to seals with an outside diameter compatible with drums being used for the SA type ring seals. Each had an outside diameter of 1.289 in. and could use the standard washer-type retainer. The first set of Springfield Armory ring seals (designated SA-1) were tested to 10,477 rounds without failure. The total number of rounds fired through these seals is now 12,585. The temperature performance of SA-2 ring seals was superior to SA-1 ring seals (See also AD-202 438)

AD-202 445 Div. 14, 26

HUFF, M., and Co., Newark, N. J.
INVESTIGATION OF SEALING QUALITIES OF GREASEPROOF BARRIER MATERIALS. Monthly progress rept. no. 8. [1958] 1v. incl. illus. tables. (Contract DA 28-017-501-ORD-2699, Proj. PA-4-20)
Unclassified report

The performance of new barrier combinations with new coatings was compared with the N-1 coating previously used on the M-6 barrier. The Picatinny Arsenal adhesion test method and a modified version of that method were used to determine the effect of variations in barrier combinations. Tests were conducted in the machine and cross directions on all materials. All samples were conditioned for a minimum of 4 hr in a constant-temperature room prior to testing. New coatings, designated N-12, N-13, and N-14, which indicated promising results, were used in the production of sample barrier materials. These samples were made of the same materials used in th 6, (over)

AD-202 445

except for the new coatings. Quick-leak tests were conducted on IA-2 packages made with QPL and newly developed materials. Cyclic tests were performed on method IA-2 packages. The M-35-28 combination failed to meet the 20-lb tensile-test requirements; tensile-test values averaged 18 lb in the unaged and aged conditions. Four sets of 5 samples each of G-85 yielded 5 failures in a total of 20 tests and resulted in averages of 4.2, 8.4, 9.0, and 10.8 in., respectively, for the destruction test. Inspection of the method IA-2 package interiors revealed failure in the cyclic exposure tests as evidenced by the presence of moisture. (See also AD-201 204, AD-203 833)

AD-203 011L Div. 26, 14

Industrial Lab., Mare Island Naval Shipyard, Vallejo, Calif.
STANDARDIZATION OF REQUIREMENTS FOR BUTYL RUBBER TO BE USED WITH NONFLAMMABLE HYDRAULIC FLUID, by Arthur W. Scott. Specification-Development test rept. 30 Apr 58 [12]p. incl. tables. (Rept. no. 3118-58) Unclassified report

Notice: Only Military Offices may request from ASTIA.

To establish limits for specification requirements for hydraulic seal materials to be used with nonflammable Cellulube 220 hydraulic fluid, a butyl stock (submitted by one of the manufacturers participating in this standardization requirement project) was evaluated at temperatures of 77°F, 158°F and 212°F, and for differernt lengths of time. Results indicate that this stock is very poor in compression set at elevated temperatures both before and following immersion in the hydraulic fluid (over)

AD-203 011L

although the other properties were considered to be normal and comparable to other butyl stocks treated. (Author)

AD-203 143L Div. 14

Material Lab., New York Naval Shipyard, Brooklyn.
QUALIFICATION TEST OF MARINE BRAND NO. 8 GASKET CEMENT, SUBMITTED BY L. W. FERDINAND AND COMPANY, INCORPORATED, NEWTON LOWER FALLS, MASSACHUSETTS. Final rept. 23 Apr 58, 4p. table (Lab. Proj. no. 5448-24) Unclassified report

Available to U. S. Military Organizations only

AD-203 158L Div. 14, 11, 27

Material Lab., New York Naval Shipyard, Brooklyn.
EVALUATION OF O-RINGS USED TO SEAL CYLINDER LINERS OF DIESEL ENGINES, SUBMITTED BY GOSHEN RUBBER CO., INC., GOSHEN, INDIANA, by S. A. Eller. Final rept. 23 Apr 58, 9p. illus. (Lab. Proj. no. 5354-14) Unclassified report

Available to U. S. Military Organizations only

AD-203 833 Div. 14, 26

Huff, M. and Co., Newark, N. J.
INVESTIGATION OF SEALING QUALITIES OF GREASE-PROOF BARRIER MATERIALS. Monthly rept. no. 9. [1958] [23]p. incl. tables.
(Contract DA 28-017-501-ORD-2699, Proj. PA-4-20) Unclassified report

Samples were made of the new barrier combinations M-05-26-2380-1, -2, and -3. Water vapor transmission tests were made on the flat, lock-seam, and end-closure areas of M-6 reproductions and other barrier combinations. Quick-leak tests were conducted on method IA-2 packages made with newly developed barrier materials. Additional cyclic exposure tests were conducted on method IA-2 packages. Coating M-05-26-2331 exhibited a transmission rate of moisture twice that previously obtained when compared with other improved coatings; this result was attributed to additives. Although changes in coatings did not show (over)

AD-203 833

any significant effect on water vapor transmission, improvements were apparent in flexibility, formability, and adhesion. Results obtained from water vapor transmission tests on the M-6 reproductions were consistent with previous results. (See also AD-202 445)

AD-204 016L Div. 14, 26

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
PRELIMINARY EVALUATION OF SEVERAL ELASTOMERS FOR USE IN HYDRAULIC SYSTEMS, by J. M. Holloway. Technical rept. 20 Apr 55 [14]p. incl. tables (Rept. no. 90-4) Unclassified report

Notice: Only Military Offices may request from ASTIA.

Screening tests were performed on thirteen elastomers to determine the most promising materials for use as gaskets, diaphragms and packings in noninflammable hydraulic fluids, such as Houghtosafe, Arochlor 1248, Pydraul F-9, as well as in inflammable MIL-O-5606 petroleum base fluid. The qualifying tests used were: a. Tensile strength, ultimate elongation, and volume change after immersion in each fluid for 94 hours at 158°F. b. Compression set, in air, after 46 hours at 194°F. The test data indicate that only Kel-F elasto- (over)

AD-204 016L

mer might be satisfactory for use in hydraulic systems containing any one of the four fluids. All of the butadiene-acrylonitrile copolymers, with the possible exception of Hycar 1014, might be satisfactory in hydraulic systems using Houghtosafe or MIL-O-5606 oil. GR-S-1023, if used in Houghtosafe only, was judged the most satisfactory of all the elastomers for use in hydraulic systems. Thiokol ST, Thiokol FA, M-5400, Philprene VP-15; and Philprene VP-25 were considered unsatisfactory because of high hot compression set. (Author)

AD-204 064L Div. 14

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
DEVELOPMENT OF GASKET STOCKS COMPLYING WITH REQUIREMENTS OF SPECIFICATION MIL-G-1858GB, CLASSES 1 AND 2, by R. R. James. Technical rept. 10 Oct 57 [25]p. incl. tables (Rept. no. 87-3) Unclassified report

Notice: Only Military Offices may request from ASTIA

The Rubber Laboratory has developed three stocks for use in the manufacture of O-ring gaskets complying with the requirements of Military Specification MIL-G-1858GB for gaskets for electrical junction boxes. One stock, 275-433, is for use in gaskets required to conform with Class 1 requirements. Two stocks, 200-195 and 200-196, are for use in gaskets required to conform to Class 2 requirements. Hot compression set tests were performed on these stocks in addition to the specification tests. (Author)

AD-204 065L Div. 14, 31

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
DEVELOPMENT OF RUBBER STOCKS FOR USE IN SEALS FOR HYDRAULIC SYSTEMS CONTAINING CELLULUBE 220, by A. E. Barrett. Progress rept. no. 3. 29 Nov 57 [15]p. incl. tables, 12 refs. (Rept. no. 90-9) Unclassified report

Notice: Only Military Offices may request from ASTIA.

The Rubber Laboratory was requested by the Bureau of Ships to develop rubber stocks for the manufacture of seals to be used in hydraulic systems containing Cellulube 220. It was shown in earlier work that butyl rubber was affected less by Cellulube 220 than any other elastomer. By means of the compounding information given in Patent No. 2,734,039, it was possible to obtain butyl rubber vulcanizates with low hot compression set. This patent describes the vulcanization of (over)

AD-204 065L

butyl rubber with a dimethylol phenolic resin, a chlorosulfonated polyethylene and a zinc compound. The Laboratory has continued the investigation of this curing system. The variables studied were the type of zinc oxide and the type of butyl rubber. It was found that the type of zinc oxide had no effect on the properties of the vulcanizate, except that untreated French-process zinc oxide did not disperse well. Butyl 215 yielded vulcanizates with better physical properties than Butyl 035 or Butyl 325. The Laboratory investigated the effect of variable channel-black loading and variable oil-content on the physical properties of Butyl 215 cured by the phenolic resin system. Stocks having nominal durometer hardnesses of 40, 60 and 80 were developed. (Author) (See also AD-145 760)

AD-204 937 Div. 13
(Unannounced report)

Naval Engineering Experiment Station, Annapolis, Md.
TEST OF HIGH PRESSURE, HIGH TEMPERATURE STEAM FLANGE TYPE JOINT SEALED WITH A FLEXITALLIC GASKET (TYPE 347SS, 900 LB, SPECIAL CG) by N. L. Navarre and R. W. Stevens. 5 Feb 51, 9p. incl. illus. (Rept. no. EES 6A66781) Unclassified report

A flange-type joint sealed with a Flexitallic gasket was subjected to thermal shock under simulated steam conditions of 1025°F and 900 p.s.i. One gasket (with no bending movement applied) withstood 100 c. of thermal shock. Tests were conducted on 3 other gaskets with a bending moment of 40,000 in.-lbs. applied to the joint externally. One failed after 51 c.; the other 2 withstood 100 c. of thermal shock. The results indicated that the Flexitallic gasket seal between flat-face flanges withstands 50 c. and possibly 100 c. of thermal shock between 1025° and 530°F under 900 p.s.i.

AD-205 020 Div. 31, 14
(Unannounced report)

Material Lab., New York Naval Shipyard, Brooklyn.
APPROVAL EVALUATION OF CABLE END SEALS UTILIZING AN ASPHALTIC HYDROCARBON OF THE MINERAL RUBBER TYPE EXHIBITED BY GENERAL CABLE CORP., PERTH AMBOY, N. J. Final rept. 21 July 52 [6]p. incl. illus. (Lab proj. no. 5287-13) Unclassified report

The asphaltic hydrocarbon of the "mineral rubber" type sealing compound used on these end seals adheres tightly to the sheath of the cable forming a compact and non-porous layer around the end of the cable. The satisfactory results obtained from the evaluation of the subject end seals indicate that the type of material and construction used by the manufacturer provides an end seal with serviceability, which compares favorably with the end seal specified in Mil-C-915 of 15 Nov 1949 with amendment 5 of 15 July 1951. (Author)

AD-205 179L Div. 14, 10

Naval Engineering Experiment Station, Annapolis, Md.
IMPROVED DURA SEAL BRAND OF MECHANICAL SEAL FOR GASOLINE PUMP SERVICE, SUBMITTED BY DURAMETALLIC CORPORATION, KALAMAZOO, MICHIGAN, by R. F. Stone. Qualification rept. 14 Oct 57, 6p. incl. illus. tables (EES rept. no. 010329) Unclassified report

Available to U. S. Military Organizations only

Improved Dura Seal brand of mechanical seal, manufactured by Durametallic Corporation, was tested to determine its suitability for use in gasoline pumps operating at stuffing box pressures between 5' and 25' head of aviation gasoline. The seal was found to be suitable for this service. (Author)

AD-205 478L Div. 14

Rubber Lab., Mare Island Naval Shipyard Vallejo, Calif.
DEVELOPMENT OF RUBBER STOCKS FOR USE IN SEALS FOR HYDRAULIC SYSTEMS CONTAINING CELLULUBE 220, by J. M. Holloway. Progress rept. no. 4, 6 June 58, 12p. tables (Rept. no. 90-11) Unclassified report

Available to U. S. Military Organizations only

AD-205 587L Div. 22, 14, 27

Naval Ordnance Test Station, China Lake, Calif.
LABORATORY PERFORMANCE TESTS OF STANDARD AND EXPERIMENTAL O-RINGS AS ROCKET MOTOR SEALS, by H. C. Hunnel and R. W. Conover. Summary rept. July 54-Jan 58. 11 Aug 58, 37p. incl. illus. tables (NOTS 2062; NAVORD rept. no. 6380) Unclassified report

Submit request via Naval Ordnance Test Station, China Lake, Calif., Attn: Code 7506

In a continuing search for a satisfactory pressure-seal gasket for rocket motors, O-rings made of various formulations and shapes were tested and evaluated. Performance tests developed as reported in NAVORD Report 3401, NOTS 988 are affirmed for use as a basis for a specification for rocket motor O-ring seals. Tests reported cover a wide variety of materials and modified formulations of standard materials. Various shapes of O-ring seals are evaluated. (Author) (See also AD-82 306)

AD-205 810 Div. 29, 14, 26
(19 Nov 58)

Midwest Research Inst., Kansas City, Mo.
DEVELOPMENT OF NEW METHODS OF SEALING FLEXIBLE CONTAINERS, by B. W. Beadle. Progress rept. no. 6, 17 Feb-16 Apr 58. 23 June 58, 12p. incl. tables.
(Contract DA 19-129-qm-923) Unclassified report

Analysis of the data from the expanded statistical peel seal experiment brought out the following points: (1) The surface representing the effect of sealing conditions on the strength of the resultant seals is apparently highly complex, with one or more critical regions existing. (2) Differentiation of the second degree equation approximating this surface indicated that the optimum condition for sealing the FX-3886 (vinyl-foil-mylar) was a "minimum-maximum", thus supporting (1) above. (3) The use of the resilient silicone rubber pad decreases the standard deviation by about one-third. (4) In no case was there any evidence of sealer position effect. The (over)

AD-205 810

experiment does give weight to the existence of areas with high peel strengths although it failed to exactly define the surface. The existence of several seals having above average strengths at the optimum sealing condition as determined by the experiment suggests that there may be major differences in the appearance and function of these seals. Microscopic examination of mounted seal specimens confirmed this. Those seals having the highest strength were considerably different in physical appearance. Some extrusion of the thermoplastic layer was noted along with irregular fusion of this layer. (Author) (See also AD-205 693, AD-205 951)

AD-205 951 Div. 29, 14, 26
(19 Nov 58)

Midwest Research Inst., Kansas City, Mo.
DEVELOPMENT OF NEW METHODS OF SEALING FLEXIBLE CONTAINERS, by B. W. Beadle. Progress rept. no. 7, 17 Apr-16 June 58. 21 July 58, 12p. incl. tables.
(Contract DA 19-129-qm-923). Unclassified report

Further evaluation of the rounded jaw sealer has shown this method to be very effective. Seals produced on the FX-3886, vinyl-foil-Mylar, laminate have given efficiencies as high as 88 per cent. The average seal strength obtained at an apparent optimum set of sealing conditions was 11.4 lb/in. This is equivalent to 74 per cent efficiency. Examination of the data suggests that further work toward increasing the flow of the vinyl layer during sealing might produce 100 per cent efficient seals. Work on an additional vinyl-foil-Mylar laminate has produced seals of 18.0 lb/in strength. This material, 0.001 in. vinyl - 0.001 in. foil - 0.001 in. Mylar, (over)

AD-205 951

did not give as high efficiencies as the FX-3886, 69 per cent being the maximum. However, this material has a tensile strength of 26.2 lb/in, about twice that of the FX-3886 laminate. Attempts to use the rounded bar sealer on Pliofilm-containing laminates met with failure; little or no improvement in seal strength was obtained. In general, the rounded jaw sealer appears very promising and is simple in operation and would be readily adaptable to existing packaging equipment. (Author) (See also AD-205 810).

AD-206 106 Div. 7, 14

Electrical Testing Lab., Portsmouth Naval Shipyard, N. H.
RUBBER GROMMETS FOR SEALING ELECTRICAL CABLE, PROPOSED BY LONG BEACH NAVAL SHIPYARD. Evaluation rept. 9 Apr 58 [9]p. incl. illus. tables (Rept. no. ETL-2800) Unclassified report

AD-206 224 Div. 29

Ekco-Alcoa Containers, Inc., Wheeling, Ill.
DEVELOPMENT OF AN EXPENDABLE HERMETICALLY SEALED CONTAINER FOR PRE-COOKED MEALS, by R. J. O'Brien. Rept. no. 9, (Final) 30 Mar 56-30 Sep 57. 4p.
(Contract DA 19-129-qm-594) Unclassified report

The design and development of an hermetically sealed container for pre-cooked dehydrated meals and the further development of a package to accommodate the container were presented. Aluminum foil was found to meet the temperature and barrier specifications of the projected packages. In making up the package, it was found that three individual units would be better than one single compartmented container. The aluminum foil containers were rectangular and heat sealed. The sealant used was a .004 in. polyethylene coating.

AD-206 374 Div. 29, 14, 26

Midwest Research Inst., Kansas City, Mo.
DEVELOPMENT OF NEW METHODS OF SEALING FLEXIBLE CONTAINERS, by B. W. Beadle. Progress rept. no. 4, 17 Oct-16 Dec 57. 12p. incl. tables.
(Contract DA 19-129-qm-923) Unclassified report

The urethane adhesive has produced the strongest and most efficient adhesive peel seals obtained to date. Peel strengths of 10.3 lb/in were obtained on vinyl-foil laminates. This material and adhesive produced a peel seal package of 32 per cent efficiency. Strong peel seals were also produced on acetate and polyethylene laminates. Three new mylar-foil laminates have been received and are being used in a statistical study of the effect of variable sealing conditions on the strength of peel and lap seals. The first part of these experiments has been completed. The variance in results on peel-sealed mylar-foil-vinyl laminates has been disconcerting. A numerical analysis of these data is being performed. It appears that expansion of the ex- (over)

AD-206 374

perimental design will be required to obtain statistically significant data. (Author) (See also AD-200 224, AD-205 951).

AD-206 446 Div. 8, 14

Raytheon Mfg. Co., Newton, Mass.
CERAMIC MINIATURE SEVERE ENVIRONMENT ELECTRON TUBE, by Edmund Mockus. Quarterly progress rept. no. 5, 1 Mar-31 May 58. [9]p. incl. illus. table.
(Contract DA 36-039-sc-73192) Unclassified report

High heater currents were traced on finished tubes to a large gas volume. This was caused by parts outgassing when the main seal was being made. A new pre-processing schedule was arrived at which solved the large volume gas problem. Large scale tests are now being run to establish if any further problems exist. Metal solders for "glueing" the grid side rod to the ceramic spacer have been abandoned in favor of ceramic or non-conducting cements. (Author) (See also AD-200 841)

AD-206 622 Div. 30
(2 Dec 58)

Naval Ordnance Test Station, China Lake, Calif.
AN EVALUATION OF LEAK-DETECTION METHODS, by Robert H. Martin and Wm. C. Shaw. 15 Oct 58, 34p. incl. illus. (NOTS 1993; NAVORD rept. no. 5868) Unclassified report

The following industrial techniques for finding and measuring small leaks are described: bubble methods, volume-sharing method, and halogen, radioactive, and helium tracers. The use of the helium leak detector is discussed in detail as it was found to be the most suitable for working with small, sealed items of ordnance. (Author)

AD-206 707 Div. 14, 10
(10 Oct 58)

Material Lab., New York Naval Shipyard, Brooklyn. RUBBER HOSE, DIAPHRAGMS AND GASKETS RESISTANT TO SPECIAL FUELS, REPORT OF DEVELOPMENT AND STANDARDIZATION OF SCREENING TESTS FOR EVALUATION OF. Final rept. 10 July 58, [7]p. incl. illus. (Lab. proj. no. 5908-2) Unclassified report

Standardized screening tests are presented which were developed for the evaluation of elastomeric materials for use in the fabrication of hoses, gaskets, and diaphragms for shipboard handling of rocket propellant chemicals.

AD-206 773L Div. 14
(24 Nov 58)

Coast Guard, Washington, D. C.
BUOY POCKETS, by C. F. Scharfenstein, Jr.
[3 Nov 58] 1v. incl. illus. tables (Rept. no. 177) Unclassified report

Notice: Only Government Agencies may request from ASTIA. Others request approval of U. S. Coast Guard, Washington, D. C.

Three types of gasketing material (a noncuring polysulphide caulking compound, a rubber gasket with cloth insert, and 6 sizes of a neoprene-cork composition gasket) were tested in 3 types of buoy pocket closures. The torque readings made from a direct reading needle were 97% accurate. The narrower gaskets were the easiest with which to make the watertight joint closure. The spraying by a high-pressure fire hose test, the 21-day awash test, and the 7-day submergence test proved nothing more than the 1-atm submergence test. (over)

AD-206 773L

If the closure passed the 1-atm test, it passed the other longer tests at less pressure. The noncuring, polysulphide, caulking compound was a satisfactory sealer. It was easily applied, required no precision fitting, was adaptable to nonuniformly flat mating surfaces, and had sufficient body to withstand 1-atm of air pressure without seal violation. The rubber-cloth insert gasket sealed the closure only 2 out of 6 times under 85 to 140 ft-lb of torque in the 1-atm air pressure tests. No water was found in the pocket after the fire hose, 21-day awash, and 7-day submergence tests. The neoprene-cork composition gaskets all passed the 1-atm air pressure/submergence test. The neoprene-cork composition and polysulfide gaskets performed better than the rubber.

AD-207 016L Div. 14, 11, 27

Material Lab., New York Naval Shipyard, Brooklyn. EVALUATION OF O-RINGS USED TO SEAL CYLINDER LINERS OF DIESEL ENGINES SUBMITTED BY ACADIA SYNTHETIC PRODUCTS, DIVISION OF WESTERN FELT WORKS, CHICAGO, ILLINOIS, by S. A. Eller. Final rept. 22 Aug 58 [18]p. incl. illus. (Lab. proj. no. 5354-17) Unclassified report

Notice: Only Military Offices may request from ASTIA. Others request approval of Chief, Bureau of Ships, Navy Dept., Wash. 25, D. C.

AD-207 035L Div. 14, 27

Material Lab., New York Naval Shipyard, Brooklyn. EVALUATION OF O-RINGS USED TO SEAL LINERS OF DIESEL ENGINES SUBMITTED BY GARLOCK PACKING COMPANY, PHILADELPHIA, PENNSYLVANIA. Final rept. 26 Sep 58 [16]p. incl. illus. tables. (Lab. proj. no. 5354-19) Unclassified report

Notice: Only Military Offices may request from ASTIA. Others request approval of Chief, Bureau of Ships, Navy Dept., Wash. 25, D. C.

AD-207 190L Div. 14, 27

Material Lab., New York Naval Shipyard, Brooklyn. EVALUATION OF O-RINGS USED TO SEAL CYLINDER LINERS OF DIESEL ENGINES SUBMITTED BY NATIONAL SEAL DIVISION, FEDERAL-MOGUL-BOWER BEARINGS, INCORPORATED, NEWARK, NEW JERSEY. Final rept. 29 Sep 58 [19]p. incl. illus. tables (Lab. proj. no. 5354-20) Unclassified report

Available to U. S. Military Organizations only.

AD-207 191L Div. 14, 27

Material Lab., New York Naval Shipyard, Brooklyn. EVALUATION OF O-RINGS USED TO SEAL CYLINDER LINERS OF DIESEL ENGINES SUBMITTED BY RUBBER LABORATORY, MARE ISLAND NAVAL SHIPYARD, by S. A. Eller and A. A. Stein. Final rept. 28 Aug 58 [17]p. incl. illus. tables (Lab. proj. no. 5354-18) Unclassified report

Available to U. S. Military Organizations only

AD-207 505 Div. 22, 25
(18 Dec 58)

Cornell Aeronautical Lab., Inc., Buffalo, N. Y.
RESEARCH ON HEAT TRANSFER AND OBTURATION FOR SMALL ARMS WEAPONS, by Gerald A. Sterbutzel. Monthly progress rept. no. 38 for Oct 58. 15 Nov 58, 53p. incl. illus. tables (Rept. no. GA-948-C-38)
(Contract DAI 30-115-504-ORD(P)-596)
Unclassified report

Two systems have been developed which in themselves are complete solutions of the cook-off problem in fighter aircraft currently being considered. One of these systems utilizes air flow over the drum to make possible the firing of at least 420 rounds at an ambient temperature of 150° without cook-off. The other - and the simpler solution - requires only a modification of the drum and seal design to make possible the firing of 420 rounds without cook-off under the severest flight conditions in any of the latest fighters. The study of this (over)

AD-207 505

type seal - known as ring seals - is continuing for optimization of heat transfer and durability characteristics. Ring seals are very small compared with the standard production seals. They have been tested with a number of variations in dimensions. All present testing is confined to seals with an outside diameter of 1.280" which can use the standard washer type seal retainer. The work discussed this month concerns tests of P-2 ring seals at temperatures between -48 and -67°F, and tests of the SAR-2 and SAR-3 retrofitted drums. (Author) (See also AD-205 837)

AD-207 538L Div. 14, 27

Material Lab., New York Naval Shipyard, Brooklyn. EVALUATION OF O-RINGS USED TO SEAL CYLINDER LINERS OF DIESEL ENGINES SUBMITTED BY RAYBESTOS-MANHATTAN, INC., MANHATTAN RUBBER DIVISION, PASSAIC, NEW JERSEY, by S. A. Eller. Final rept. 30 July 58 [16]p. incl. illus. tables (Lab. proj. no. 5354-10) Unclassified report

Available to U. S. Military Organizations only.

AD-207 979L Div. 26, 14, 22

Rock Island Arsenal Lab., Ill.
INVESTIGATION OF G-T RING PACKINGS FOR 155 MM GUN EQUILIBRATORS, by L. R. Raisbeck. 1 Dec 58, 16p. incl. illus. tables (RIA Lab. rept. no. 58-3290) Unclassified report

Available to U. S. Military Organizations only

Two sets of Palmetto G-T Ring packings were tested in 155 MM Gun Equilibrators. One set of packings was equipped with phosphorous bronze backup rings and the other set with Teflon backup rings. The two sets of packings also differed slightly in the dimensions of the elastomer sections. The equilibrators were subjected to friction and gymnastication tests at sub-zero, normal and elevated temperatures to determine if the packings were superior to conventional O-rings. The equilibrator with packings utilizing phosphorous bronze backup rings completed 40, 150 cycles of gymnastication without failure but the polished plunger [illegible]ace [illegible]er)

AD-207 979L

was severely worn. The equilibrator with packings utilizing Teflon backup rings failed after 7,275 cycles due to loss of nitrogen pressure. Both of the equilibrators had higher frictions than standard equilibrators with O-ring packings. It was concluded that G-T rings when used in the 155 MM Gun Equilibrator do not seal as well as O-rings, have higher frictions than O-rings, reduce or eliminate spiral twisting, and produce excessive wear on the mating surface when phosphorous bronze backup rings are used. (Author)

AD-208 687 Div. 29

Little, Arthur D., Inc., Cambridge, Mass.
VAPOR-PROOF CLOSURE. Final rept. 5 Nov 58, 16p. incl. illus. (Rept. no. C-60613)
(Contract DA 19-129-qm-839) Unclassified report

Efforts were made to develop an improved closure system for special clothing. The closure system is to be erosion proof against guided-missile propellants; gas-, liquid-, and air-tight; durable; fail-safe; and easily operated. Gluing, pneumatic, purse-string, snapspring, zipper, bayonet, and continuous-locking closures were considered. The newly developed continuous-locking closure was considered the most promising system. This system embodies 2 molded rubber portions that act as seals. Right-and left-wound springs vulcanized in the rubber, when brought together, mesh and form a hole through their common centers for the entire distance of the closure. Into this hole, a flexible wire is inserted by means of a winding device located at one end of the closure. Although additional improve-
(over)

AD-208 687

ment of the mechanism is considered necessary, the continuous-locking concept shows promise because of the following features: (1) it can be manufactured in any length; (2) it is practically indestructible; (3) it is flexible in all directions and jam-free; (4) it has a unique emergency release mechanism; (5) when closed, it is air-and liquid-tight up to 15 psi; and (6) it could be designed to withstand much higher pressures.

AD-208 838 Div. 14

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
PRELIMINARY INVESTIGATION OF GENTHANE-S, by J. W. Hollister and R. E. Morris. Technical rept. 9 Jan 59 [16]p. incl. illus. tables (Rept. no. 79-3) Unclassified report

The Rubber Laboratory has investigated a urethane rubber, Genthane-S, to determine in a preliminary way its suitability for the manufacture of mountings, hoses and gaskets. In the crude form, this rubber, unlike other urethane prepolymers, is a solid and in general is processed like other crude synthetic rubbers. Notable features of Genthane-S vulcanizates are the following: (1) Excellent tensile properties and good tear resistance. (2) Excellent resistance to hot air at temperatures up to 250°F. (3) Excellent resistance to petroleum oils and aromatic gasoline. (4) Excellent resistance to weather. (5) Good resistanc compression set at 212°F and to drift under load a om tem-
(over)

AD-208 838

perature. (6) Good resistance to water, salt water and dilute acids at temperatures up to 130°F. (7) Good resilience. (8) Poor resistance to hot water, caustic solutions and phosphate plasticizers. (9) Poor resistance to compression set at temperatures below 30°F. Genthane-S does not appear to be particularly suitable for use in mountings, but does appear to be suitable for use in hoses and gaskets not subjected to hot water, caustic solutions, ester plasticizers, or, in the case of gaskets, to temperature below 30°F. (Author)

AD-209 169L Div. 30

Atomic Energy Research Establishment (Gt. Brit.).
THE 1441A OXYGEN LEAK DETECTOR, by R. S. Barton. Sep 57, 5p. illus. (AERE rept. no. GP/M 189; H. 24502) Unclassified report

Submit request via Technical Information Service, Oak Ridge, Tenn.

An auxiliary power pack, designed for use in conjunction with the 1075 ionization gauge control unit, has been developed for the purpose of rapidly converting an ionization gauge, used in a vacuum system for pressure measurements, into a diode for leak detection using O_2 as the probe gas. Entrance of the gas into the vacuum envelope leads to a decay of electron emission from the hot cathode of the diode and this is observed on a meter suitably positioned in the gauge circuit. A description is given of experiments made to obtain the information necessary for the design of the a—ve mentioned pack followed by a brief description c type finally built for stores issue. (Author)

AD-209 179 Div. 22, 11, 14

[Aberdeen Proving Ground, Md.]
ENVIRONMENTAL (DESERT) TEST OF TWO OE-10 LUBRICANT SEALING SYSTEMS FOR M48A2 TANK ROADWHEEL BEARINGS, by Duane R. Bade. 14 Aug 58 [14]p. incl. illus. table (Proj. no. TB5-1401/Y316, memo. rept. no. 1) Unclassified report

Removal of the "O" ring from the standard seal allows oil leakage at two different areas. To halt leakage between the seal and road arm it is necessary to have a machined gasket bearing surface on the external dust shield and to cement gasket TD 11763 to this surface. No satisfactory means of stopping leakage between the dust shield and the spring adaptor assembly has been determined. Application of Code A elastic cement failed to completely halt the leakage. It has been noted that contact of gasket TD 11763 with OE 10 oil for approximately 24 hours causes an increase in size of approximately 4%. The conditions encountered by road
(over)

AD-209 179

wheel seals necessitate a high standard of installation, as improper handlin of the seal before installation may influence seal performance. In an attempt to minimize leakage, all corrosion and dirt particles should be brushed from the bearing spacer. New "O" rings should also be used since contact with lubricant swells and deforms the rings and also causes partial loss of the materiel's elasticity. All new "O" rings should be closely inspected for defects prior to installation. See Inclosure 6. The Code A seal does not have interchangeability on the road-wheel arms or provision for bearing inspection without removal of the seal retaining screws. (Author)

AD-210 323 Div. 26, 8

Stanford Electronics Labs., Stanford U., Calif.
ELECTROFORMED CERAMIC-TO-METAL SEAL FOR VACUUM TUBES, by M. D. Hare, H. F. Keller, and H. A. Meneses. 17 Nov 58, 11p. incl. illus. (Technical rept. no. 453-3)
(Contract DA 36-039-sc-73178) Unclassified report

An electroformed ceramic-to-metal seal was developed to bond a large number of components such as an external--circuit TWT. The seals are electroformed with Cu by means of a plating bath. The process is as follows: (1) the ceramic components are metalized by the standard Mo-Mn process; (2) the metalized areas are plated with Ni and sintered; (3) Cu is plated on the Ni and fired to the sintering or flow temperature; (4) Au plating may then be put on and diffused to the Cu at 400°C; (5) the parts are jigged together with provision being made for electrical contact to the internal components and the metalized portions of the ceramics; (6) Lucite solution may be used as a sealer at the inter-
er)

AD-210 323

faces of the components, evaporated, and pumped out of the assembly during bakeout; (7) any electrically conductive area which is not to be plated is masked off with acid-resistant tape or lacquer; (8) the assembly is put through a cleaning cycle; and (9) the assembly is immersed in an acid-Cu bath at a current density of about 40 amp/sq ft for 30 sec, after which the current density is reduced to 10 amp/sq ft. for a 24-hr period. A deposit of 0.008 in. of electroformed Cu in 24 hr is sufficient to assure a vacuum-tight seal. This method of bonding tube components forms the bond at room temperature with materials which can be subsequently fired to high temperatures. (See also AD-135 048)

AD-210 822L Div. 14, 4

Atomic Energy Research Establishment (Gt. Brit.).
THE PERMEABILITY OF A NATURAL RUBBER AND SOME RUBBER-LIKE MATERIALS TO H_2, He, N_2, O_2 AND A, by R. S. Barton. Nov 58, 4p. illus. (Rept. no. AERE Z/M 210; HL 58/3140) Unclassified report

Submit request via Technical Information Service, Oak Ridge, Tennessee

See also AD-116 938

AD- 211 410 Div. 14, 4

Industrial Lab., Mare Island Naval Shipyard, Vallejo, Calif.
BUTYL RUBBER COMPOUND NO. X-1150-30 SUBMITTED BY PARKER SEAL COMPANY, by Arthur W. Scott. 25 Feb 59, 3p. incl. tables (Suitability rept. no. 0179-59) Unclassified report

AD-212 518L Div. 14, 31

Material Lab., New York Naval Shipyard, Brooklyn. CALKING AND WEATHERING COMPOUND, TAPES FOR BIMETALLIC FAYING SURFACES, SUITABILITY OF. Progress rept. no. 2, 23 Mar 59, 5p. incl. tables (Lab. proj. no. 5540-4) Unclassified report

Notice: Only Military Offices may request from ASTIA. Others request approval of Chief, Bureau of Ships, Navy Dept., Wash. 25, D. C.

See also AD-208 732.

AD-212 634L Div. 14

Industrial Lab., Mare Island Naval Shipyard, Vallejo, Calif.
STANDARDIZATION OF REQUIREMENTS FOR BUTYL RUBBER TO BE USED WITH NONFLAMMABLE HYDRAULIC FLUID (ONE STOCK) by Arthur W. Scott. Specification-Development test rept. 19 Nov 58 [8]p. incl. tables (Rept. no. 6466-58) Unclassified report

Notice: Only Military Offices may request from ASTIA. Others request approval of Chief, Bureau of Ships, Navy Dept., Wash. 25, D. C.

To establish limits for specification requirements for hydraulic seal materials for use with non-flammable pure phosphate ester type hydraulic fluid, a butyl stock designated as B-5 was evaluated at temperatures of 77, 158 and 212 degrees F, and for different time periods. Test results indicate that this stock is satisfactory for the intended purpose, particularly in its having low (over)

AD-212 634L

compression set at 212 degrees F. (Author) (See also AD-203 011)

AD-212 661 Div. 26, 14

Huff, M., and Co., Newark, N. J.
INVESTIGATION OF SEALING QUALITIES OF GREASPROOF BARRIER MATERIALS, by Leonard C. Drabrant. Monthly progress rept. no. 14, 1-28 Feb 59 [25]p. incl. illus. tables
(Contract DA 28-017-501-ORD-2699, Proj. PA4-20)

Mechanical, semimechanical, and manual techniques were applied for closing packages made from M-6 and M-05-26-2380-3 barrier materials. The time required to fabricate the packages with each of these techniques was recorded, and each package was then subjected to the vacuum quick-leak test, MIL-P-116C, Amend. 1 (Navy). Tests showed that the M-6 barrier material can be fabricated into a Method IA-2-type package by the use of the new method of closure with any one of the sealing techniques. (See also AD-208 926)

AD-212 866L Div. 14, 31

Material Lab., Puget Sound Naval Shipyard, Bremerton, Wash.
EVALUATION OF A NEOPRENE COMPOUND FOR CALKING OF FLIGHT DECKS (NS-033-200) by J. R. Freeborn. Jan 59, 2p. (Technical rept. no. R-419) Unclassified report

Available to U. S. Military Organizations only

AD-213 564L Div. 31, 14

Materials Labs., Puget Sound Naval Shipyard, Bremerton, Wash.
OBSERVATIONS ON MODIFICATIONS FOR CORRECTING HUB OIL LEAKAGE ON CONTROLLABLE PITCH PROPELLERS ON THE USS TERRELL COUNTY (LST 1157) AND USS WHITFIELD COUNTY (LST 1169) (NS-033-200) by W. B. Lew. Nov 58 [12]p. incl. illus. tables (Technical rept. no. R-413) Unclassified report

Available to U. S. Military Organizations only.

AD-213 568L Div. 14, 31

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
DEVELOPMENT OF A SYSTEM FOR SPLICING O-RINGS ABOUT THE PROPELLER SHAFTS OF LST CLASS 1156 VESSELS (NS-033-200) by J. W. Hollister. 10 Oct 58 [12]p. incl. illus. tables (Rept. no. 90-10) Unclassified report

Notice: All requests require approval of Chief, Bureau of Ships, Navy Dept., Wash. 25, D. C.

AD-213 956 Div. 27, 14, 31

Naval Engineering Experiment Station, Annapolis, Md.
ENDURANCE TEST OF A PACKARD MODEL 1D-1700 DIESEL ENGINE FITTED WITH EES SPECIAL CYLINDER LINER SEALS, by William H. Barnes, III. Summary rept. 6 Mar 59, 69p. incl. illus. tables (EES rept. no. 510002AJ) Unclassified report

An expanded version is available to USMO only as AD-213 959L

The aluminum crankcases of Packard diesel engines in Fleet use have been subject to varying degrees of cavitation-erosion damage near the cylinder liner seal ring area. In Fleet and Station tests, special seal bands developed by the Engineering Experiment Station for use in the top seal ring groove of the cylinder liners have reduced erosion in the critical water seal area for engine operating periods up to 1800 hours. This re- (over)

AD-213 956

port summarizes the results of a 2851-hour test to evaluate improved seal band materials produced by the Mare Island Naval Shipyard. Some of the new bands were reinforced with nylon inserts. The performance of the seal bands was excellent, and no significant cavitation-erosion occurred in the critical crankcase seal areas. High crush and low crush connecting rod bearings and Flexitallic cylinder head gaskets were evaluated concurrently in the 2851-hour test. The high crush bearings performed satisfactorily; those with low crush were unsatisfactory, due to loosening and movement in the rod. The Flexitallic cylinder head gaskets proved superior to the standard flat cylinder head gaskets previously used in this engine. Flexitallic head gasket life, under the severe Station test conditions, averaged approximately 1000 hours. (Author)

AD-214 003L Div. 22

Rock Island Arsenal Lab., Ill.
TEST OF KEL-F PLASTIC PACKING SHROUDS IN A 105 MM HOWITZER RECOIL MECHANISM, by L. R. Rasbeck. 3 Apr 59, 14p. incl. illus. tables (RIA Lab. rept. no. 59-899) Unclassified report

Notice: Only Military Offices may request from ASTIA. Others request approval of Diamond Ordnance Fuze Labs., Wash. 25, D. C. Attn: ORDTL 06.33

Not releasable to Foreign Nationals

A 105 MM Howitzer Recoil Mechanism was assembled with heat-treated Kel-F plastic packing shrouds, synthetic rubber fillers and aluminum alloy cup rings. Optimum floating piston Belleville spring loads were established by comparing the radial packing pressure obtained with a Kel-F packing shroud to that obtained with a standard Teflon shroud. The mechanism was gymnas- (over)

AD-214 003L

ticated a total of 25,000 cycles at ambient temperatures of -70F, 75F and 220F and was considered to be still in serviceable condition. After completion of the gymnastication tests, the recoil oil was slightly emulsified, and the rod and cylinder surfaces were in good condition. However, the Kel-F plastic packing shrouds were all badly cracked. It was concluded that the Kel-F plastic material was not suitable for recoil mechanism packing shrouds due to possible cracking in service. (Author) (See also AD-6252, AD-149 981)

AD-214 082 Div. 26, 8, 14

Corning Glass Works, N. Y.
PILOT PRODUCTION OF GLASS CRYSTAL HOLDERS (HC-(XM-2)/U; HC-(XM-3)/U; HC(XM-4)/U, by R. K. Whitney. Quarterly progress rept. no. 3, 22 Nov 58-27 Feb 59, 1v. incl. illus. tables.
(Contract DA-36-039-sc-81255) Unclassified report

Preproduction sample glass crystal holders were manufactured, and equipment was applied to develop a satisfactory sealing technique. Covers (or bulbs) as base assemblies were fabricated for HC-(XM-2)/U, HC-(XM-3)/U, and HC-(XM-4)/U crystal holders. Sealing experiments with varying cycle times, frequencies, and power settings indicated that satisfactory seals can be produced. Alloys of known melting points, 230°, 250°, 285°C, which were attached to mounting springs as well as soldered to the centers of actual quartz crystals, melted and flowed under optimum sealing conditions. Various samples were sealed with equipment having ratings of a 20-kva input and a 16-kw maximum output. (See also AD-211 019)

AD-214 101L Div. 20, 30

United Kingdom Atomic Energy Authority (Gt. Brit.).
BURST CARTRIDGE DETECTION. A STUDY OF MIXING IN A CALDER CHANNEL, by T. Burgoyne. 1959, 7p. illus. (Technical note no. IGR-TN/W-1016)
Unclassified report

Submit request via Technical Information Service, Oak Ridge, Tenn.

If the gaseous fission products associated with can failure are not fully mixed with the coolant, a low concentration at the sampling hole could make the damage less sensitive to detection. To investigate this effect, nitrous oxide was injected into the air flow in a model Calder channel and then detected downstream with an infra-red gas analyser. Mixing along the fuel elements was found to be slow, and the distribution by no means complete even after half the length of a channel, but this is largely compensated by good mixing between the end of the top fuel element and the sampling position.
(over)

AD-214 101L

In the worst case, when failure occurs near the outlet end of a channel, the sensitivity may be reduced by a factor of 2. This is unlikely to have a serious effect on efficiency of detection. (Author)

AD- 214 262L Div. 6, 14, 31

Electrical Testing Lab., Portsmouth Naval Shipyard, N. H.
SUITABILITY TESTS ON NYLON STUFFING TUBES FOR BULKHEAD APPLICATION -NSS-687-017-FRANKLIN PLASTIC INCORPORATED-EXHIBITOR, by E. Shea. Evaluation rept. 19 Sep 58, 5p. illus. (Rept. no. ETL-2955)
Unclassified report

Notice: Only Military Offices may request from ASTIA. Others request approval of Chief, Bureau of Ships, Navy Dept., Wash. 25, D. C.

Available on loan only

Samples complied with the vibration test requirements of Military standard 167 of 20 Dec 1954. Samples complied with the high impact shock requirements of Spec MIL-S-901B of 19 Dec 1955. Samples were satisfactory with respect to torsion requirements. Samples were not consistent with respect to impact test requirements. Samples contained moisture in amounts ranging from
(over)

AD-214 262L

2.3% to 4.8% by weight. Samples were satisfactory with respect to specific gravity and melting point requirements of Spec MIL-M-19098 of 11 Nov 1955. Samples exhibited relative viscosity values below that considered acceptable by the Bureau of Ships for the types Z-35A and Z-35B black nylon stuffing tubes. (Author)

AD-214 269 Div. 7

Electrical Testing Lab., Portsmouth Naval Shipyard, N. Y.
TERMINALS AND END SEALS FOR MINERAL INSULATED (MI) CABLES-NSS672-200. Evaluation rept. 31 Dec 56, 6p. illus. tables (Rept. no. ETL-2452)
Unclassified report

Representative samples of ceramic insulated end seals purchased from the Advance Vacuum Products Company, Stamford, Connecticut, together with suitable lengths of mineral filled (ML) cable, were submitted to evaluation tests. Results of tests revealed end seal cable assemblies to be generally unsatisfactory with respect to water leakage tests at 200 pounds per square inch and erratic with respect to performance during the dielectric strength tests on units which were silver soldered together. Cause of leakage was attributed to excessive heat required to obtain a satisfactory silver-soldered joint. The sample assemblies which were lead-tin soft soldered together were satisfactory when
(over)

AD-214 269

exposed to the 200 pounds per square inch water leakage tests. (Author)

AD-214 848L Div. 22

Rock Island Arsenal Lab., Ill.
LUBRICATORS FOR RECOIL MECHANISM FLOATING PISTON PACKINGS, by F. Blessin. 23 Apr 59, 17p. incl. illus. tables (Rept. no. 59-1115)
Unclassified report

Notice: Only Military Offices may request from ASTIA. Others request approval of Diamond Ordnance Fuze Labs., Wash. 25, D. C. Attn: ORDTL 06.33

Not releasable to Foreign Nationals

Fifteen materials were selected for consideration as lubricators for recoil mechanism packings. Samples of each material were subjected to screening tests to determine sub-zero and elevated temperature effects and oil wicking action. A polyester fiber felt was selected for test as a packing lubricator. Two tests were conducted with a 105 MM Howitzer Recoil Mechanism, one with a lubricator of the selected felt on each side of the
(over)

AD-214 848L

nitrogen end packing on the floating piston, and the other without lubricators. Total gas leakage past the nitrogen packing on the floating piston was approximately the same in both tests during 25,000 cycles of gymnastication at sub-zero, normal and elevated temperatures. However, visual examination indicated that the nitrogen end floating piston packing in the latter test was worn more than in the former test in which lubricators were used. The lubricators were not damaged by the test. Additional recoil mechanism tests, with and without lubricators on the floating piston, are currently in progress. Information obtained in these tests will be considered before conclusions are drawn or recommendations are made. (Author) (See also AD-6252)

AD-215 094 Div. 14, 31

Material Labs., Puget Sound Naval Shipyard, Bremerton, Wash.
EVALUATION OF RUBBER SEAL OF A SPLIT DESIGN FOR OVERSHAFT HUB OIL INJECTOR ON THE USS TIOGA COUNTY (LST 1158) (NS-033-200) by Wah B. Lew. Apr 59, 4p. (Rept. no. R-421)
Unclassified report

AD-216 630 Div. 1, 9
(9 Oct 59)

Bendix Products Div., Bendix Aviation Corp., South Bend, Ind.
STUDY AND PRELIMINARY DESIGN OF A HIGH TEMPERATURE AIRCRAFT WHEEL, BRAKE, AND TIRE ASSEMBLY, by Gilbert T. Stout. Rept. for 5 May 58-4 May 59, on An 800°F Wheel, Brake and Tire System. 1v. incl. illus. tables, 67 refs. (WADC Technical rept. no. 59-310)
(Contract AF 33(616)5503)
Unclassified report

Design of a wheel, brake, and tire capable of operating from an ambient temperature of 800°F appears feasible. However, there are many areas which necessitate intensified study. Most prominent are mechanical tire approaches, bearings and lubricants, and hydraulic or pneumatic dynamic seals. The mechanical tire and tread material is the most revolutionary and challenging region to be conquered. Various mechanical tire approaches were considered, ~~~~ igns ir)

AD-216 630

utilizing spring configurations were more acutely investigated. Initial studies indicate that a mechanical tire possessing a spring configuration will involve a considerable weight penalty in comparison to the conventional wheel and casing presently being used for low temperature applications. (Author) (See also AD-142 117, AD-150 973)

AD-216 838 Div. 8

McCoy Electronics Co., Mt. Holly Springs, Pa.
GLASS INCLOSED MINIATURE CRYSTAL UNITS, by Howard C. Long. Quarterly rept. no. 2, 1 Nov-31 Jan 59. 1v. incl. illus. tables
(Contract DA 36-039-sc-74942)
Unclassified report

Vacuum bake-out with cold traps in bake-out oven and sealing machine appears to improve yield of sealed units. The stainless steel loop springs have been considered unsatisfactory and their use has been dropped at least temporarily. The Philips clips are unsatisfactory for use with crystal units of frequency less than 30 m.c. Temperature cycling results in general are encouraging the overall width of the S-curves being appreciably less than ± 0.003% nominal frequency for all units tested except for three units for which the odd shaped S-curve indicated an otherwise faulty unit. The major problem continues to be that of finishing the crystal after sealing to a frequency falling within the range-0.0005 to + 0.001% nominal frequency. (Author) (See also AD-210 080)

AD-218 038 Div. 17, 14, 31

Coast Guard, Washington, D. C.
CORROSION TESTS OF METALLIC JOINTS, by C. F. Scharfenstein, Jr. 18 May 59, 30p. incl. illus. tables (Rept. no. 183) Unclassified report

An evaluation was conducted of different means and materials to effectively insulate bimetallic joints of the design being used in boat construction at the Coast Guard YARD using YARD production procedures. Aluminum-to-aluminum and aluminum-to-steel joints were made up using steel, cadmium-plated steel and aluminum Huck Lockbolts as fasteners. Insulating materials under consideration included: Chrome Lock Tape, Type N, MIL-G-20241, Symbol 2291; 3M Insulating Type, EC1202 1" x 1/16" manufactured by Minnesota Mining and Manufacturing Company. All liquid materials were afforded two hours curing time at 70°F before fasteners were tightened. Samples were caulked with PRC 1201-T compound around edges after assembly. Samples were subjected to a thirty-day (over)

AD-218 038

immersion test in a 10% aqueous sodium-chloride solution and to 200 hours of accelerated weathering. It was concluded that the use of Chrome-Lock Tape insulating material between faying surfaces with all faying edges caulked with the polysulfide rubber caulking compound was the most satisfactory method under the conditions of application and testing. Aluminum fasteners were best for aluminum-to-aluminum joints; Stainless steel fasteners were best for aluminum-to-steel joints. (Author)

AD-218 730 Div. 8, 26

Ontario Research Foundation, (Canada).
INVESTIGATION INTO THE SEALING OF QUARTZ CRYSTAL HOLDERS TYPE HC-6/U, by B. W. Schumacher. Final rept. on Phase 2 for 1 Apr 56-31 May 58 on ECDC Proj. C 19, 72p. incl. illus. tables. Unclassified report

Solder sealing was investigated in an attempt to improve this technique as a method of hermetrically sealing quartz crystal units in metal holders. Differences in the quality of some commercial units were examined. None of the factors such as stress in the cans or differences in thermal expansion coefficients were shown to be a source of failure. However, there were some indications of poor bonding in older units. With respect to the bases for the HC-6/U holder, a reliable product can be made if the manufacturing process is properly controlled and risky shortcuts are avoided. Good workmanship and observing the simple rules in the soldering (over)

AD-218 730

process appear to be the decisive factors in obtaining good units. The time between pre-tinning the cans and assembling the units should be short to avoid oxidation. A handling method was studied for an ultrasonic soldering process; fluxes are avoided, and the penetration of the solder appeared to be good. A simple, fast, and sensitive leak test method is described for glass enclosed units. (See also AD-219 139)

AD-218 903 Div. 22

Lake City Arsenal, Independence, Mo.
STUDY OF GAS LEAKS, M52A3 SERIES PRIMER, by D. L. Stonger. Final rept. June 59, 36p. incl. illus. tables, 15 refs. (IED rept. no. 59-3) Unclassified report

Conclusions: Primer gas leaks encountered during Ordnance acceptance testing of contractors primers is not the result of poor product design, but is attributed to use of substandard components and/or inadequate quality procedures (since approximately 98 per cent of the primer gas leaks occurred in primers produced by one contractor). The type of propellant (i.e., IMR 4903, IMR 7065, or WC 870) cannot be correlated with the incidence of primer gas leaks. Component parts of the firing contact pin assembly - M39 gun, and the bolt assembly - M61 gun, are replaced most often as the result of expected wear or fatigue. Conclusive information on primer gas leaks at extreme temperature (over)

AD-218 903

ranges (i.e., - 65F, 165F) is not available on ammunition fired through the M39 or M61 gun. Primer gas leaks cannot be correlated with the existence or nonexistence of the crimp which is used to secure the primer in the primer pocket of the case. The probability that primer gas leaks will occur is greatest when ammunition is fired through the M39 gun. Primer gas leaks are not currently considered a problem when firing electric primed ammunition through the M39 or M61 gun when the ammunition is conditioned at - 65F ambient or 165 F. (Author)

AD-219 109 Div. 27
(27 July 59)

Koppers Co., Inc. [Baltimore, Md.]
EVALUATION OF DESIGNS AND MATERIALS FOR HIGH SPEED-HIGH TEMPERATURE SHAFT SEALS FOR TURBOJET ENGINE APPLICATIONS, by John J. Brenza, John H. Fuchsluger and others. Rept. for 1 May 57-31 Dec 58 on Gas Turbine Technology. 88p. incl. illus. tables (WADC Technical rept. no. 56-267, pt. 2)
(Contract AF 33(616)5015 [Continuation of Contract AF 33(616)3004]) Unclassified report

This development program was undertaken to develop rubbing seals that would operate for 1000 hours, at 1000°F, and at a rubbing speed of 30,000 feet per minute. Accordingly, new materials were investigated; these materials were incorporated in seals; and their performance as seals was evaluated. In all, one hundred and sixty-three materials were screened on the basis of room temperature and 1000°F, dry, rub- (over)

AD-219 109

bing, wear tests and 1000-hour, 1000°F oxidation tests. Material combinations found to be promising were next evaluated as seals under increased operating conditions, and eventually under conditions duplicating those existing in a jet engine. (Author) (See also AD-110 636)

AD-219 139 Div. 8, 26

Ontario Research Foundation (Canada).
INVESTIGATION INTO THE SEALING OF QUARTZ CRYSTAL HOLDERS TYPE HC-6/U, by B. W. Schumacher. Quarterly progress rept. no. 4 on Phase 2, 1 June 57-15 Mar 58. 24 Mar 58 [9]p. incl. illus. table. Unclassified report

The long term studies of existing units were finished by tabulating the results and by preparing some graphs showing the changes in leak size over the period of observation. Solder tests with model joints and special units were continued. An improvement by using other materials than customary seems to be unlikely, since good workmanship seems to be the most important factor for obtaining a high quality seal. At present, ultra sonic soldering techniques are being investigated. A number of commercial units were cut and diamond polished. Pictures of the solder seam were taken with a magnification of 50. They show that improvements in (over)

AD-219 139

workmanship may still be possible. In some cases a poor bond between glass and metal base was revealed. A series of tests was prepared with the objective of determining the effect of leaks on the crystal and reducing these effects by placing a drying agent inside the enclosure of the crystal. Serious experimental difficulties were encountered and the tests were discontinued. On the new type of glass enclosed crystals we studied the possibility of leak tests by means of an induced glow discharge. A reliable leak test seems possible. (Author) (See also AD-141 968, AD-218 730)

AD-220 048

during the sealing operation was assembled. A 250-w. tunable (15 to 50 kc.) signal source, a magnetrostrictive transducer, and focusing horn were attached to the controlled condition heat sealer. The transducer and focusing horn were substituted for the lower sealing jaw. This equipment subjected the seal area to high frequency vibration during the sealing operation. Definite improvement in the strengths of contaminated seals was obtained with certain laminate-contaminant combinations, while others failed to improve in strength. The greatest effect noted was an improvement of low order seal strengths. Equivalent improvement in higher order seal strength was not generally observed. (Author) (See also AD-205 951)

AD-220 048

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AD-220 293 Div. 14, 7
(27 July 59)

Burgess Battery Co., Freeport, Ill.
PLASTIC SEALS, by C. J. Broughton. Quarterly progress rept. no. 8, 1 Mar-1 May 59. 1 June 59, 11p.
(Contract DA 36-039-sc-10527) Unclassified report

Battery seals were made of regular polyethylene, polypropylene, and ethylcellulose. Regular polyethylene was used in determining the effectiveness of crimping of the tools. Half of all polyethylene samples, which had passed the soldering vacuum test, leaked under the vacuum test after being soldered to the side of a Zn can. Polypropylene was 70% effective after crimping. A fairly effective seal of ethylcellulose was obtained, but only by precrimp heating. Heat treatment of the 3 types of substances (which had passed the presoldering vacuum test) prior to crimping resulted in 95% of the seals passing the vacuum test after soldering. (See also AD-203 976, AD-208 026)

AD-220 698 Div. 8
(18 Aug 59)

McCoy Electronics Co., Mount Holly Springs, Pa.
GLASS INCLOSED MINIATURE CRYSTAL UNITS, by Howard C. Long. Quarterly rept. no. 3, 1 Feb-30 Apr 59, on Development of Glass Inclosed Miniature Quartz Crystal Units. 1v. incl. illus. tables.
(Contract DA 36-039-sc-74942) Unclassified report

Test results obtained during sealing using slotted supports are discussed. Overall yield figures including initial fabrication and sealing range from 65% to 85%, rough estimate. This calculation counts as satisfactory only those units finishing after seal to within -0.0005% to +0.0001% nominal frequency. The percentage yield when considering temperature cycling only ranges from 60% to 100%, the average figure being 79%. Vacuum bake-out of the crystal units was eliminated. All glass parts are cured at 175°C for 3 hr in a ventilated oven. The cold trap has become an integral part of the sealing operation. Tightened specifications are being asked for (over)

AD-220 698

on the glass parts, with probably the selection of a better grade glass tubing for manufacture of the bulbs. (Author) (See also AD-218 838)

AD-221 401 Div. 22, 10, 29
(10 Dec 59)
(Unannounced report)

[Naval Weapons Lab., Dahlgren, Va.]
GUNFIRE TEST OF GOODYEAR SELF SEALING FUEL CELLS WITH 20mm PRACTICE AND .60 CALIBER BALL AMMUNITION, by B. K. Lloyd. Final rept. 28 July 48, 1v. incl. illus. (Proj. TED no. NPG AE 53041); NPG rept. no. 73) Unclassified report

Descriptors: Fuel tanks*; Seals; Vulnerability.

Open-ended Terms: 20mm, Cal 60.

AD-221 404 Div. 10, 22
(10 Dec 59)
(Unannounced report)

[Naval Weapons Lab., Dahlgren, Va.]
GUNFIRE TEST OF SELF SEALING FUEL HOSE WITH VARIOUS NEW TYPE FUELS, by B. K. Lloyd. Final rept. 28 July 48, 1v. incl. illus. (Proj. TED no. NPG AE 626001; NPG rept. no. 70) Unclassified report

Descriptors: Fuel hose*; Vulnerability; Tests; Seals*.

AD-221 461 Div. 14
(4 June 60)

Material Lab., New York Naval Shipyard, Brooklyn.
SHELF AGING OF SYNTHETIC AND NATURAL RUBBER MATERIALS; INVESTIGATION OF. Progress rept. no. 2. 3 Dec 59. [17]p. incl. illus. (Lab. proj. no. 5974, pt. 3 and 4) Unclassified report

Descriptors: Rubber*; Synthetic rubber*; Gaskets; Rubber gaskets*; Vibration isolators*; Packaging*; Mechanical properties; Temperature; Aging.

Packaged and unpacked specimens of natural rubber hatch door gasket and neoprene GR-T shock mount material, Mare Island stocks 13-383 and N-31-1, respectively, were aged in a Greer oven at 140° ± 2°F. At aging intervals of 3 weeks during an elapsed period of 18 weeks, packaged and unpackaged specimens of both stocks were removed from the oven and tested for tensile strength, elongation, tensile product, modulus at 300%, swelling ratio, compression set, and stress relaxation. The tests indicated that the packaging had no beneficial effect on hatch door and shock mount materials subjected to accelerated aging. (See also AD-229 186L)

AD-221 803 Div. 7 U
(10 Feb 61)
(Unannounced report)

Naval Civil Engineering Lab., Port Hueneme, Calif.
INSTRUMENTATION OF SAN NICOLAS DIESEL ENGINE POWER PLANT ENGINES WITH BOILING CONDENSING COOLING, by Earl J. Beck. 7 May 58, rev. 16 June 58, 12p. incl. illus. tables (Proj. NY 012 015-3; Technical note no. N-337, suppl. to Technical rept. no. 008, AD-163 134)
Unclassified report

DESCRIPTORS: *Auxiliary power plants, Tests, *Diesel engines, Starting, Temperature, *Electric power production, Fuel seals, Alternating current, Temperature control, Cooling, Heat transfer.

The U. S. Naval Civil Engineering Laboratory, Port Hueneme, California, cooperated in the instrumentation of an engine at the plant of the supplier for the San Nicolas power plant expansion engines, and cooperated in conducting exploratory testing of the engine at elevated

steam pressures. The initial instrumentation was designed primarily to detect possible sealing ring problem areas, but allowed partial analysis of the heat transfer and temperature distribution in the area of junction of cylinder and head, including the upper portion of the piston ring travel. The analysis, with isothermal plots under four conditions of loading and cooling is given, including one run with forced circulation of low temperature coolant. A proposal for simple but more precise instrumentation of one of the two 8-cylinder engines to be installed on San Nicolas Island is made. From the tests made, it was concluded that probably critical metal temperatures were well within established safety limits. (Author)

AD-221 803

AD-222 071 Div. 27, 1, 4 U
(Unannounced report)

Pesco Products Div., Borg-Warner Corp., Bedford, Ohio.
RESULTS OF A STUDY OF THE BEST PUMPING SYSTEM FOR A LIQUID PARA HYDROGEN AIRCRAFT FUEL SYSTEM, by G. H. Caine and S. Prelewski. Dec 58, 1v. incl. illus. table, 21 refs. (Pesco engineering rept. no. 3042; Document no. 276)
(Contract AF 33(616)3642)
(WADC TR 58-679) Unclassified report

DESCRIPTORS: Pumps, *Fuel pumps, Centrifugal pumps, Design, Aviation fuels, Liquefied gases, *Hydrogen, *Cryogenics, Fuel tanks, Heat transfer, Seals, Thermal insulation, Fluid flow, Friction, Bearings, *Fuel systems, *Aircraft equipment, Materials, Rotary seals.

Open-ended Terms: SF-1.

Problems in pumping liquid para H (SF-1), pumping from low liquid levels in a tank, some useful constructions, and configurations of tank-

mounted SF-1 boost pumps and their inter-relations with tank configurations are illustrated and discussed. Boost pump requirements for pumping SF-1 from a tank to give net position suction head (NPSH) for a main pump at the end of a 100-ft. (typical) insulated line are defined. A general solution of this problem is derived. Using real data on liquid H, the general solution is also presented in a graphical nomographic form from which particular solutions are obtained by entering particular values of variables (line diameter, length, heat leak rate, flowrate, NPSH needed at far end of the transfer line, etc). Means of boost drive, thermal isolation and static rotative sealing are illustrated and discussed. (Author)

AD-222 071

AD-222 593 Div. 27 U
(TISTA/SEB)
(Unannounced report)

Rocketdyne, Canoga Park, Calif.
MODIFICATION INSTRUCTION NUMBER NINE. DELETION OF XLR101-NA-1 VERNIER ENGINE PROPELLANT VALVE POSITION INDICATOR ON MA-1 ENGINE.
21 Oct 57, 14p. incl. illus. (Rept. no. R469-9)
(Contract AF 04(647)98)
Unclassified report

DESCRIPTORS: (Liquid rocket propellants, Rocket motors, Fuel systems, *Valves, Seals, Fuel seals, Gas leaks, Reliability, Design.)

Identifiers: Vernier engines, LR 101 engines.

The change consists of removing the present vernier propellant valve's Kel-F lip seal, microswitches, shaft, and end cap to eliminate the problem of leakage; replacing the existing fuel poppet spring with a different one to increase closing reliability; and making wiring changes to the GSE to make it compatible with the changed engine. (Author)

AD-222 684 Div. 14, 25 U
(TISTM/GEC)
(Unannounced report)

Naval Research Lab., Washington, D. C.
A STUDY OF GLASS SEALERS, by Peter King and Allen L. Alexander. 9 Sep 42, 11p. illus. (NRL rept. no. P-1934)
Unclassified report

DESCRIPTORS: (*Glass, *Coatings, *Optical coatings, Adhesion, Seals, *Films, Acrylic resins, Textiles, cellulose.) (Tests, Test methods, Impact shock, Explosion damage, Climatic factors.)

To protect ordinary glass from scattering as a result of bomb blast and gun blast, a restraining material was needed for emergency application to the surface of the glass. This report described a study made on 19 such materials by several methods of test. Recommendation was made that 4 materials be further evaluated by testing at the Naval Proving Grounds. (Author)

AD-222 849 Div. 3, 30 U
(TISTB/AW)
(Unannounced report)

Biological Labs., Frederick, Md.
ENGINEERING CONSTRUCTION, OPERATION, MAINTENANCE AND DEVELOPMENT IN RESTRICTED AREA. SECTION XIII. GASKETS,
by James F. Matousek. 15 Nov 45. 36p. incl. illus. tables (Special rept. no. 42, Section no. 13)
Unclassified report

No automatic release to Foreign Nationals.

DESCRIPTORS: (*Biological laboratories, Construction, Maintenance, Operation, Structures, Buildings, Military engineering.) (Gaskets, Seals, Flanges.)

AD-222 852 Div. 3, 30 U
(TISTB/AW)
(Unannounced report)

Biological Labs., Frederick, Md.
ENGINEERING CONSTRUCTION, OPERATION, MAINTENANCE AND DEVELOPMENT IN RESTRICTED AREAS. SECTION XVI. SHAFT SEALS,
by James F. Matousek. 15 Nov 45. 19p. incl. illus. tables (Special rept. no. 42, Section no. 16)
Unclassified report

No automatic release to Foreign Nationals.

DESCRIPTORS: (*Biological laboratories, Construction, Maintenance, Operation, Structures, Buildings, Military engineering.) (Hydraulic seals, Centrifugal pumps.)

AD-225 213 Div. 4, 26
(4 Sep 59)

Reaction Motors Div., Thiokol Chemical Corp., Denville, N. J.
EVALUATION OF BEARINGS AND SEALS IN 90% HYDROGEN PEROXIDE, by H. O. Petersen, S. R. Matos, and J. Hyll. Rept. for 1 June 58-28 Feb 59. 1v. incl. illus. tables (Rept. no. RMD 103F, Vol. 2)
(Contract NOas 56-506-c) Unclassified report

The results show that it is feasible to run a bearing submerged in room temperature 90% hydrogen peroxide, and submerged in 90% hydrogen peroxide when decomposition of hydrogen peroxide is taking place in a large capacity catalyst bed located adjacent to the bearing housing. A sleeve bearing consisting of a compatible material such as teflon with glass as an additive will give better performance than a stainless steel roller or needle bearing which tend to contaminate the peroxide. Face seals and segmented shaft seals which have the stationary sealing elements made of teflon with
(over)

AD-225 213

glass as an additive will perform well in peroxide at room temperature, but not as well under high temperature conditions produced by peroxide decomposition in the catalyst bed. The teflon elements disintegrate quickly under the high temperature environment, particularly on the segmented shaft seals. An S-Monel France type shaft seal will perform well in combination with a Ni-resist sleeve under the high temperature conditions. However, the leakage rate is high because this is a "controlled gap" type seal. Test results indicate that the best sealing arrangement for the high temperature conditions is one which uses a positive seal (either a face or shaft type with a teflon-glass stationary sealing element, which is shielded from the high temperature gases by a series of all metal "controlled gap" seals. (Author) (See also AD-217 364)

AD-225 495 Div. 14

Products Research Co., Los Angeles, Calif.
[DEVELOPMENT OF SEALANTS WHICH WILL WITHSTAND EXPOSURE TO FUEL, VAPOR, LIQUID FUEL AT HIGH TEMPERATURES] by I. P. Seegman. Quarterly progress rept. no. 5, 1 Apr-1 July 58, 15 July 58, 56p. incl. tables.
(Contract AF 33(616)3976) Unclassified report

Properties of the best fill and drain system developed in the first year were determined after exposure to JP-5 fuel and to air at 500°F and 550°F. Efforts to increase solids content of Viton A formulation G-50 were unsuccessful. Solvent based Viton A formulations were investigated as faying surface sealants. Good adhesion to stainless steel was retained after exposure to fuel at 450°F with a 6 day open time. For titanium a different formulation is required and results were unsatisfactory with a 6 day open time. Filleting sealants have been prepared and evaluated before and after exposure
(over)

AD-225 495

to JP-5 for up to 30 hours at 550°F. Adhesion to stainless steel and titanium has been retained following 120 hours exposure in JP-5 at 450°F plus 18 hours in JP-5 at 550°F. A literature survey was made of the effects of nuclear radiation on plastic and elastomeric materials. Preliminary work with various organic antirads indicates that they either inhibit the cure of Viton A or they are fuel extractible. (Author) (See also AD-151 257, AD-307 052)

AD-225 501 Div. 26, 3, 22
(8 Sep 59)

Sciaky Bros., Inc., Los Angeles, Calif.
WELDING DEVELOPMENT PROGRAM FOR MEDIUM CARBON HEAVY WALL MUNITIONS CLOSURES, by J. W. Christensen. Summary rept. 10 July 59, 31p. incl. illus.
(Contract DA 18-405-cml-211) Unclassified report

The problem was to develop leakproof resistance weld sealing of the shell and burster casing of 155 mm munitions for the containment of the toxic agents. Resistance welds attained within the subject munitions parts were of high strength and leakproof. Basic weld tooling and standard test equipment used during development is adaptable to production application. Resistance weld joining of the munitions does not require the use of any material other than the as is munition components. The feasibility of the leakproof resistance welding of 155 mm medium carbon heavy wall munitions has been demon-
(over)

AD-225 501

strated. Utilization of the developed techniques and basic tooling designs requires only the pre-weld machining preparation of shells and burster casings to commercial tolerances. (Author) (See also AD-204 680)

AD-226 196 Div. 8, 26
(2 Oct 59)

Knights, James, Co., Sandwich, Ill.
GLASS HOLDER CRYSTAL UNITS, by Robert E. Christian. Quarterly rept. no. 2, 1 June-31 Aug 59. 7p.
(Contract DA 36-039-sc-81274) Unclassified report

The development and production of one machine suitable for sealing quartz crystal units in miniature glass holders is being attempted. The induction heater frequency was increased to 4 mc, producing a decrease in work-coil losses and adequate heat for sealing. A satisfactory heating fixture was also designed. The single position sealing unit was assembled and preliminary testing was completed.

AD-226 294 Div. 8
(5 Oct 59)

Eitel-McCullough, Inc., San Bruno, Calif.
CERAMIC TUBE REFINEMENT, by John Richter and Jack W. White. Scientific rept. no. 4, 1 Apr-30 June 59. [15]p. incl. illus.
(Contract AF 33(600)36690) Unclassified report

Tube life at high temperatures: The evaluation of sealing procedures and seal surface treatments was made with the Uooden described previously (AD-217 534). The Uoode proved to be an impractical test vehicle and its use was discontinued. Five CD-19 tubes were flame-sprayed with Al_2O_3 as a deterrent against high-temperature seal failure and tested along with 5 standard CD-19 tubes. The coatings were uniform, but too porous to be effective. Grid evaluation: The grid evaluation structure was equipped with an indirectly heated unipotential grid-simulating element which provided uniform element temperature a uniform potential. Two
(over)

AD-226 294

structures incorporating Ti grid-simulating elements were made and tested. There are 2 phases to the test procedure: (1) pretreatment of the surface by cleaning or contamination, and (2) setting test conditions and plotting the data to determine the equilibrium point. Comparison of emission current density vs temperature curves will be the basis for grid material evaluation. Getter evaluation: The demountable vacuum system for evaluation of getters in ceramic tubes was completed. Tests indicated that fresh Ceralloy solutions must be used to coat the electrodes in order to avoid flaking.

AD-226 521 Div. 8
(2 Oct 59)

Amphenol-Borg Electronics Corp., Chicago, Ill.
RESEARCH ON R. F. COAXIAL CONNECTORS, BROADBAND, HIGH TEMPERATURE, RADIATION RESISTANT, by J. V. Maick. Scientific rept. no. 2, 1 June-1 Sep 59, 75p. incl. illus. tables, 24 refs. (Contract AF 33(616)6243) Unclassified report

Synthesis of connector dielectric configurations: Combination sectoral-annular insulator structures of fused silica and air were devised which yielded effective dielectric constants of 2.5 or less. A homogeneous fused silica insulator was designed with a relatively low fabrication cost and is capable of hermetic sealing. A fused silica insulator based on the application of Brewster's Angle was devised which has low insertion loss and is also capable of hermetic sealing. All three of these insulators possess various disadvantages such as cost, insertion loss, and variance of the voltage stand- (over)

AD-226 521

ing wave ratio. Metals: Experimentation with metals to be used for particular connector components led to the following recommendations: (1) the center contact (male) should be silver-plated, nickel-clad copper; (2) the center contact (female) should be silver-plated Iconel X; (3) the shell material should be stainless steel type 347, silver plated on the inside; and (4) the gasket material should be aluminum or stainless steel. Effects of radiation on materials: The use of an over-design factor of about 25% should eliminate the effect of radiation-induced air ionization on the electrical performance of the RF coaxial connector. This correction should be added to corrections for temperature and pressure. Sealing study: Although seals currently exist which will operate at temperatures of 500°C, the high-temperature seals are generally not reusable. (See also AD-217 740)

AD-228 239L Div. 14, 8
(19 Aug 59)

[illegible] Aircraft Co., Culver City, Calif.
[illegible]OPMENT OF ULTRA HIGH TEMPERATURE DIELECTRIC MATERIALS FOR EMBEDDING ELEC[illegible] PARTS, by John B. Rust and Charles L. [illegible] Quarterly progress rept. no. 1, 10 Feb-[illegible] May 59, on a Program to Develop and Evaluate Silicone or Modified Silicone Dielectric Materials. 30 May 59, 28p. incl. illus. tables, 19 refs. (Contract NObs-77054) Unclassified report

Notice: All requests require approval of Chief, Bureau of Ships, Navy Dept., Wash. 25, D. C.

This report describes the work on a program to develop and evaluate silicone or modified silicone dielectric materials which will be useful for embedding electronic parts which must function continuously at 350°C ambient temperature. Two monomeric arylenesilane compounds were prepared: 1, 4 bis(dimethylethoxysil- (over)

AD-228 239L

yl)benzene and 1, 4 bis(methylvinylethoxysilyl)benzene. These monomers were subsequently cohydrolyzed with either methylvinyldiethoxysilane or dimethyldiethoxysilane. The cohydrolysis products were viscous fluids containing a multifunctionality of vinyl groups appended to a polyarylenesiloxane chain. The monomers and 'prepolymers' were identified by infrared absorption spectroscopy. Dibutyltin dihydride was prepared for use as a potential crosslinking agent. A commercially available polysiloxane polymer was identified as being similar to the cohydrolysis products prepared during this program. The commercial resin, DC 7521, was combined with either dibutyltin dihydride or silicon oxyhydride, in the presence and absence of several catalysts, in an attempt to crosslink the polymers. Crosslinking did not appear to proceed under the conditions used in these experiments. (Author) (See also AD-130 798, AD-217 190)

AD-228 320 Div. 11, 22
(18 Nov 59)

Aberdeen Proving Ground, Md.
DESERT TEST OF COMPONENTS ON AN M48A2 TANK, by P. J. Moravec. Automotive rept. Nov 59, 1v. incl. illus. (Proj. IT-5057, rept. no. 8) Unclassified report

An expanded version is available to U. S. Military Organizations as AD-228 319L

The evaluation is reported of six of 23 non-standard components (Road Wheel Bearing Seals, Aluminized Mufflers, Sprocket Hub without Dowels, Improved End Connectors, Teflon Road Arm Bearing, and Reduced Quantity of Road Wheel Mounting Nuts) for tracked vehicles in the 50-Ton class, during 4,000 miles of M48A2 tank operation on surfaced and off-road test courses in ambient temperatures to 11[illegible] F. The concept, design, or material in the road wheel mounting nuts and the teflon road wheel arm bearing, was inade- (over)

AD-228 320

quate or unsatisfactory. Suitability and durability of the remainder were generally as good or better than the standard item. (Author)

AD-228 846 Div. 26, 14
(16 Nov 59)

Huff, M. and Co., Newark, N. J.
INVESTIGATION OF SEALING QUALITIES OF GREASEPROOF BARRIER MATERIALS. Monthly progress rept. no. 22, 1-31 Oct 59. 6p. (Contract DA 28-017-501-ORD-2699, Proj. PA4-20) Unclassified report

Results obtained on the four additional water vapor transmission tests which were run on the CM-20 reproduceability series showed no significant difference from those obtained on previous water vapor transmission tests run on the CM-20 experimental barrier composite. The examination of the interiors of each group of ten IA-1 and IA-2 packages at the conclusion of the cyclic exposure tests showed no staining and, therefore, no evidence of entry of water into the packages. This is further evidence that packages fabricated with the improved barrier (CM-20) using the newly developed closures will effect satisfactory seals. Comparison of these (over)

AD-228 846

cyclic test results obtained on Method IA-2 packages fabricated with CM-20 barrier composite with those obtained with M-6 show the performance of both to be satisfactory and comparable. (Author) (See also AD-226 516)

AD-229 115 Div. 14, 12
(4 Dec 59)

Douglas Aircraft Co., Inc., Santa Monica, Calif.
EFFECTS OF USING MPD-2067 LOW TEMPERATURE HYDRAULIC OIL IN NIKE SYSTEM. MODEL 1'10, by M. H. Holjer. 6 Feb 59, rev. 1 Apr 59, 1v. incl. illus. tables (Rept. no. SM-27931) (Subcontract to Bell Telephone Labs., Inc., Contract DA 30-069-ORD-1092, Proj. TU1-3070) Unclassified report

Descriptors: Hydraulic oils*; Hydraulic systems*; Hydraulic valves; Hydraulic cylinders; Hydraulic seals; Guided missiles; Surface to Air; Launching; Maintenance equipment.

An investigation was made of the problems associated with the field use of MPD-2067 hydraulic oil in Nike equipment. The main problems were concerned with the overheating of the hydraulic ground equipment and an excessive amount of leakage throughout the entire Nike hydraulic system. Except for color and viscosity, the physical and chemical properties are the same for MIL H5606 and MPD-2067. Tests with MPD-2067 showed that the overheating problems with this oil will be eliminated by adhering to the recommended program of oil changes and to the recommended duty cycle. Exter- (over)

nal lap leakage will be greater with MPD-2067 than with MIL-H-5606. Tests conducted on the 1001177 launcher cylinder indicate that the problems associated with spiral failure of launcher cylinder O-rings are no more severe with MPD-2067 than with MIL-H-5606. The valve wear tests conducted on a 1020773 valve assembly indicate no excessive wear associated with MPD-2067. A procedure for changing to the low viscosity oil is outlined. Systematic maintenance of the Nike hydraulic system was discussed. In general, the use of MPD-2067 should be limited to use in that equipment directly involved in checking out the entire missile. Component test equipment should use MIL-H-5606 at all times, and any components tested with MIL-H-5606 should have oil drained from them prior to installation in a system which is using MPD-2067.

Open-ended Terms: Nike.

AD-229 115

AD-229 186L Div. 14
(23 Sep 59)

Material Lab., New York Naval Shipyard [Brooklyn]
SHELF AGING OF SYNTHETIC AND NATURAL RUBBER MATERIALS; INVESTIGATION OF, by D. H. Kallas. Progress rept. no. 2 [18 May 59] [19]p. incl. illus. tables (Lab. proj. no. 5974, pts. 1 and 2) Unclassified report

Notice: Only Military Offices may request from ASTIA. Others request approval of Chief, Bureau of Ships, Navy Dept., Wash. 25, D. C.

Descriptors: Rubber*; Synthetic rubber; Gaskets; Aging*;

Packaged and unpackaged specimens of the low-temperature gasket and O-ring materials, Mare Island stocks 162-1100 and 275-31, respectively, were aged in a Geer oven at 140 ± 2°F. At weekly intervals for 17 wk, the specimens were removed from the oven and tested. In addition, specimens of both stocks were tested after they had been shelf-aged for periods up to 1 yr at 74 ± 2°F and at about -100°F. The effect of Geer oven aging at 140 ± 2°F on the tensile (over)

strength, elongation, tensile product, modulus at 300%, swelling ratio, compression set and stress relaxation of packaged and unpackaged specimens were plotted vs time. The physical properties of packaged specimens of the low-temperature gasket and the O-ring that were shelf-aged for ½- and 1-yr were tabulated. In addition, the physical properties of both elastomers after exposure for ½- and 1-yr at -100°F while in the unpackaged and packaged conditions were tabulated. Results indicated that packaging has a slightly beneficial effect on the low-temperature gasket material, and no apparent effect on the O-ring material. Packaging cannot prevent thermal effects, however, it may be beneficial in retarding or preventing the volatilization of ingredients from the elastomer by building up a high concentration of these materials within the package. Self-aging for 1 yr had less effect on the low-temperature gasket and O-ring than Geer oven aging for 1 wk at 140 ± 2°F. Conditioning specimens at -100°F apparently retards aging. (See also AD-220 970L)

AD-229 186L

AD-229 662 Div. 26
(18 Dec 59)

Feltman Research and Engineering Labs., Picatinny Arsenal, Dover, N. J.
PRODUCT ENGINEERING TO IMPROVE MATERIALS AND DESIGN MODIFICATION OF FIBER CONTAINERS AND HERMETICALLY SEALED CONTAINERS, by Theodore Fruchtman. Final engineering rept. no. FR-3-60, Oct 59, 6p. Unclassified report

Descriptors: Containers*; Packaging*; Seals; Fibers.

AD-229 700 Div. 7
(22 Dec 59)

Amphenol-Berg Electronics Corp., Chicago, Ill.
RESEARCH ON R. F. COAXIAL CONNECTORS, BROADBAND, HIGH TEMPERATURE, RADIATION RESISTANT, by J. V. Malek. Scientific rept. no. 3, 1 Sep-1 Dec 59, 52p. incl. illus. tables.
(Contract AF 33(616)6243) Unclassified report

Descriptors: Electric connectors*; Radiofrequency cables; Coaxial cables; Dielectrics; Broadband; Design; Seals; Metals; Insulating materials; Radiation effects.

Emphasis was shifted from theoretical study and literature surveys to initiation of development, testing, and evaluation of the component parts of the connector. Fused silica insulator samples were partially evaluated with respect to their physical and electrical properties. To date, fused silica is still considered the best insulator material in conjunction with the contract requirements. The design of insulator-contact structure is under laboratory refinement yielding a good VSWR of the developed structure. A study was made of the effect of contact to cable center conductor fastening methods and the resulting VSWR of (over)

the UG-21D/U connector. Crimping and staking as compared to soldering had minor effects on the VSWR of the connectors tested. An experimental outer contact of stainless steel, type 347F was high temperature and endurance tested and satisfactorily met these requirements. Sealing studies and sealing development were advanced. Silver-magnesium-nickel alloy was reconsidered as a possible center contact material because it has high conductivity, oxidation resistance, and fair mechanical strength at elevated temperatures. A survey of radiation test facilities is presented. (Author) (See also AD-226 521)

AD-229 700

AD-229 707 Div. 26, 14
(22 Dec 59)

Huff, M. and Co., Newark, N. J.
INVESTIGATION OF SEALING QUALITIES OF GREASEPROOF BARRIER MATERIALS. Summary rept. Dec 59, 1v. incl. illus. tables.
(Contract DA 28-017-501-ORD-2699) Unclassified report

Descriptors: Packaging*; Materials; Seals*; Organic coatings*; Corrosion; Adhesion; Tests.

Grade C flexible, greaseproof barrier materials are used to make method IA-1 and IA-2 packages, as prescribed by MIL-P-116C. The functional performance of these barriers is improved by subsequently dip-coat sealing the wrapped package to provide a high degree of water vaporproofness. New barrier materials were developed which make possible the fabrication of adequate method IA-1 and IA-2 packages without the need of dip-coat sealing. An initial investigation proved the feasibility of developing such materials. Further work and laboratory testing resulted in the development of a new barrier material capable of producing water vaporproof packages without the need (over)

for dip-coat sealing, grade A for method IA-1 packages, or an outerwrap for method IA-2 packages. (Author) (See also AD-217 417)

AD-229 707

AD-231 223L Div. 1, 12
(29 Jan 60)

Chance Vought Aircraft, Inc., Dallas, Tex.
MODULAR HYDRAULIC SYSTEM DEVELOPMENT (PROJECT HYDRATOY) by R. W. Burden, R. L. Evans and others. Quarterly progress rept. no. 4, pt. 1. 5 Sep-5 Dec 59, 28 Dec 59, 1v. incl. illus. tables (Rept. no. AER-E9R-12467)
(Contract NOa(s) 59-6019-c) Unclassified report

Notice: Only Government Agencies may request from ASTIA. Others request approval of Chief, Bureau of Naval Weapons, Navy Dept., Wash. 25, D. C. Attn: RAAE-322

Descriptors: Hydraulic systems*; Hydraulic actuators*; Standardization; Design; Alloys; Machining; Aircraft; Guided missiles; Space ships; Metal seals; Seals.

Work was continued on the development of a circumferential seal and on a Hi-Ceal. Tests of the Chance Vought X-seal was begun, the results were inconclusive. Tests of the High Pressure engineers Hi-seal revealed that the installation forces are higher than desirable (over)

and that there is considerable dimensional variation between seals that are supposedly identical. Preliminary test results on metallic O-rings indicated that they would be suitable as alternate seals in case of unexpected failure or delay in Hi-Ceal development and qualification. Effort is being made to establish a series of standard modular hydraulic components which will be used in packaging aircraft hydraulic systems. The package or manifold will fulfill the requirements of a particular given system. An CVS-54175-1 Ti manifold simulating an arresting gear control package was fabricated. Detail drawings for number 2 package (a manifold actuator) and layout plans for the number 3 package (second actuator) are described. An investigation is being made of methods and configurations for using airframe structure to carry fluids. The tests of fluid carrying structure continue to be encouraging, new configurations are being constructed. A mounting base for the commercial high temperature hydraulic pump was built. The pump and its driving motor were installed on the base. The remainder of the equipment required for this set-up is being installed on the base. (See also AD-231 501L)

Open-ended Terms: Project Hydratoy.

AD-231 223L

AD-231 501L Div. 1, 12
(3 Feb 60)

Chance Vought Aircraft, Inc., Dallas, Tex.
MODULAR HYDRAULIC SYSTEM DEVELOPMENT [PART II] by O. A. Moelk. Material and Process Development progress rept. no. 2, 5 June 59-5 Dec 59, 1v. incl. illus. tables (Rept. no. AER-E9R-12467)
(Contract NOa(s) 59-6019-c) Unclassified report

Notice: Only Government Agencies may request from ASTIA. Others request approval of Chief, Bureau of Naval Weapons, Navy Dept., Wash. 25, D. C., Attn: Code RAAE-322

Descriptors: Hydraulic systems*; Hydraulic actuators*; Seals; Standardization; Design; Alloys; Machining; Aircraft; Guided missiles; Space ships; Extrusion; Brazing; Metal seals.

Progress is reported on the development of modular hydraulic system components and concepts. Producibility limitations for stainless steel extrusions were established; tubular aspect in an extrusion is currently not considered extrudable. Roll-bonding of (over)

ferrous materials was considered a remote possibility and the project was terminated. A manifold of annealed Ti-8Al-4V alloy was successfully fabricated from a forged billet. No difficulties were experienced during the machining of the manifold and all specified tolerances and finishes were met. Spray brazing work is being developed. The process utilizes powdered alloys which can be applied to the work piece by spraying the powder through an oxyacetylene flame. The use of AM-350 steel for hydraulic tubing is expected to reduce the weight of the high pressure systems. Metallic seals will require additional testing to determine an optimum hardness range for a specific application. For optimum operating conditions, a proposed material and hardness range was established for the modular hydraulic system. (See also AD-231 223L)

Open-ended Terms: Project Hydratoy

AD-231 501L

AD-231 622 Div. 29, 20
(5 Feb 60)

Dewey and Almy Chemical [Co.] Cambridge, Mass.
EFFECT OF RADIATION ON THE PHYSICAL PROPERTIES AND THE SEALING OF CANS. Final rept. 1 Nov 56-30 July 58. 2p. Unclassified report

Descriptors: Radiation effects*; Physical properties; Seals; Containers*; Ionization; Sealing compounds*; Rubber; Vulcanizates; Polymerization; Gamma rays.

A study of the effect of ionizing radiation up to 5.6 Megarad on the physical properties of a variety of can sealing compounds has shown that: (1) compounds based on unvulcanized butadiene-styrene rubber, butadiene-acrylonitrile rubber, or natural rubber are cross linked and toughened; (2) compounds based on neoprene or vulcanized rubber are not noticeably changed; (3) compounds based on butyl rubber are softened, apparently as a result of depolymerization; (4) there is no measurable difference in the sealing performance of irradiated and non-irradiated cans as determined by vacuum retention; and (5) there is no measurable difference in sealing performance between cans irradiated by 2 Mev electrons and cans irradiated by gamma rays. (Author)

AD-232 073 Div. 20, 30
(12 Feb 60)

Blaw-Knox Co., Pittsburgh, Pa.
INVESTIGATION AND STUDY OF PRIMARY LOOP IMPROVEMENTS FOR NUCLEAR POWER PLANTS. Rept. for 11 Feb-11 Dec 59. Dec 59, 111p. incl. illus. tables.
(Contract AF 30(602)2006)
(RADC TR 59-235) Unclassified report

Descriptors: Nuclear power plants*; Power reactors; Pressurized water reactors; Boiling water reactors; Design; Pipe fittings; Gaskets; Valves; Pumps; Materials; Corrosion research.

Design and selection features are reviewed for pipe connectors and gaskets, valves, circulating pumps, and pipe material pertaining to the primary loop of a prepackaged nuclear power plant. Limitations of size and weight for air transport, and factors of field assembly and dis-assembly are discussed. Discussion of primary coolants indicated that the pressurized water or boiling water cycles are preferred. Pipe connectors with related gasket types are discussed. Two-bolt connectors with special forms of metallic gaskets are rec- (over)

ommended in preference to standard ASA flanged joints. Zero leakage and controlled leakage valves are described. Valve operating and control systems are discussed outlinining factors of dependability and the effect of total leakage on design of auxiliary water system. Circulating pumps of conventional design with manufacturers standard or special mechanical seals are discussed and compared with canned rotor pumps on the basis of leakage rates, dependability, maintenance and cost. Operating experience and test data for both types are reported. Simplified flow sheets of pressurized water and boiling water reactor systems are presented for discussion of corrosion. Factors affecting corrosion are emphasized. Induced radioactivity on the secondary effects on corrosion products is discussed. (Author)

AD-232 073 N

AD-232 964 Div. 8, 14, 30
(2 Mar 60)

United States Testing Co., Inc., Hoboken, N. J.
TEST PROCEDURE UNDER SPECIFICATION MIL-C-26086 (USAF) CAPACITORS, FIXED, PLASTIC OR PAPER-PLASTIC (HERMETICALLY SEALED IN METALLIC CASES). Dec 59, 1v. incl. illus. tables.
(Contract AF 33(616)6177)
(WADC TR 59-578) Unclassified report

Descriptors: Fixed capacitors*; Plastics; Paper capacitors; Paper; Seals*; Specifications; Tests; Test methods.

A step by step procedure for qualification approval testing of specific types of capacitors under Military Specification MIL-C-26086 (USAF), MIL-C-26086/1, MIL-C-26086/2, MIL-C-26086/3, MIL-C-26086/4, MIL-C-26086/5 and MIL-C-26086/6 is presented. Reference is made to each test and the specification paragraph under which the test is accomplished. The criteria for meeting or failure to meet the specification requirements are detailed. Although reference is made to specific types of equipment to be used, this does not preclude the use (over)

of other equipment which has the required accuracies and electrical characteristics. (Author)

AD-232 964

AD-232 977 Div. 14
(2 Mar 60)

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
EFFECT OF SHELF AGING ON MIL-P-5516 O-RINGS, by J. M. Holloway. Progress rept. no. 1. 29 Feb 60, 3p. table. (Proj. no. NSM-033-200; Rept. no. 92-6) Unclassified report

Descriptors: Synthetic rubber; Rubber seals*; Tests; Physical properties; Storage; Aging; Gaskets*.

O-rings originally complying with Military Specification MIL-P-5516, which had been scrapped because they had reached the maximum allowed age of 48 mo after manufacture, were tested before and after an additional 12 mo aging. During the original 48 mo they were kept in sealed envelopes; but during the subsequent 12 mo some were kept in sealed envelopes, some were exposed to air in the dark, and some were exposed to air and artificial light. No significant changes in tensile strength, ultimate elongation, modulus at 100% elongation, permanent set at 50% elongation or specific gravity were found to have occurred after the additional 12 months aging. The results were the same no matter how the O-rings were stored during this period. (Author) (See also AD-204 017)

Open-ended Terms: O-rings.

AD-232 977 . N

AD-233 471 Div. 8
(14 Mar 60)

Eitel-McCullough, Inc., San Carlos, Calif.
CERAMIC TUBE REFINEMENT, by John Richter and Jack W. White. Scientific rept. no. 5, 1 July-31 Dec 59, [20]p. incl. illus.
(Contract AF 33(600)36609) Unclassified report

Descriptors: Electron tubes*; Ceramic materials*; Cathodes (Electron tubes)*; Seals; Heating; Temperature; Design.

Seal Evaluation: Two samples of ten CD-16 tubes were subjected to life testing at ambient temperatures of 400°C and 500°C respectively. The test at 400°C was terminated after 500 hours and none of the tubes sustained air leaks during test or after cooling, subsequent to removal. In the life test conducted at 500°C, seal failures occurred after 16 hours, 23 hours, 72 hours, 88 hours and 97 hours (two). After 160 hours, all of the remaining tubes had failed. Getter Evaluation: A further revision of the getter evaluation sub-manifold was necessary, and as a final solution to the leakage problem, the original sub-manifold was replaced with a completely ballarced sub-manifold. The new sub-manifold was incorporated into the pump system, and the entire system was satisfactorily checked out. Grid (over)

Evaluation: A Keithly electrometer, capable of measuring currents as low as 10^{-14} amperes, was procured. Efforts to measure emission currents from gold grid simulating elements, at a sensitivity of 10^{-9} amperes with the Keithly electrometer, were frustrated by inherent circuit interference. To solve this problem a test circuit was designed and built to provide a means of measuring currents of the order of 10^{-13} amperes. (Author) (See also AD-226 294)

AD-233 471

AD-234 109 Div. 10, 9, 12
(30 Mar 60)

Peaco Products Div., Borg-Warner Corp., Bedford, Ohio.
STUDY AND PRELIMINARY EXPERIMENTAL EVALUATION OF MISSILE FUEL SYSTEMS AND COMPONENTS USING LIQUID HYDROGEN. Summary rept. on Phase 1. July 59, 1v. incl. illus. 23 refs. (Peaco engineering rept. no. 3094)
(Contract AF 33(616)5810)
(WADC TR 59-428) Unclassified report

Descriptors: Fluid flow*; Rocket fuels*; Hydrogen*; Liquified gases*; Heat transfer; Fuel tanks; Fuel pumps; Pressure; Thermodynamics; Guided missiles*; Induction motors; Design; Liquid rocket propellants*; Rocket motors; Bearings; Seals.

The work included in this report covers the accomplishments during the first year on an anticipated three-year program to study and evaluate SF-1 fuel system flow phenomena and system components needed between the fuel tank and the missile engine high pressure pump. The report includes an analysis of two phase flow in SF-1 transfer lines, a discussion of the fluid conditions in the (over)

supply tank, a discussion of the booster pump application to a missile fuel system, and a study of possible methods of measuring vapor-liquid ratio in SF-1. Descriptions of components and test equipment for use in later phases of the program are presented. These include: a single phase flow loop, a 7000 gallon dewar and a 1500 gpm, 250 psi circulation pump for the flow loop. The experimental phases discussed in the report include the performance of a 24,000 rpm booster pump, a 24,000 rpm submerged electric motor drive for a booster pump, and preliminary results on the effect of fluid stratification in a missile tank on the performance of a tank-mounted booster pump. (Author) (See also AD-214 257)

Open-ended Terms: SF-1

AD-234 109 N

AD-234 140 Div. 29
(30 Mar 60)

Quartermaster Research and Engineering Command, Natick, Mass.
AN INVESTIGATION TO DETERMINE A PRACTICAL, PRECISE AND RELIABLE METHOD FOR DETECTING LEAKS IN RUBBER HANDWEAR, by James C. Perkins, Jr. Mar 60, 36p. incl. illus. tables (Technical rept. no. R-6) Unclassified report

Descriptors: Rubber gloves*; Protective clothing; Guided missile personnel; Detectors; Gas leaks; Test methods; Detection.

Research indicated three methods as being potentially suitable for use in detecting leaks in rubber handwear under field testing conditions (1) high definition fluoroscopy; (2) halogen detector; and (3) infrared nitrous oxide analyzer. Each of these methods employs a tracer gas and a detector or sensing element. Some of the characteristics for which these systems were evaluated, included adequate sensitivity, reliability of response, simplicity of operation, freedom from toxicity, contamination and explosion hazards, portability, and economy of operation. The three experimental inspection methods were evaluated in comparison with the standard water bath technique (over)

in field tests at the Agency's Simulated Missile Servicing Course, and at the Agency's Accelerated Handwear Testing Course. The leak detector tracer gas (Halogen) was found to be the most satisfactory method for making determinations of rubber handwear serviceability. It is more precise and reliable and requires substantially less time per inspection than the standard technique and is superior to the other experimental methods with regard to practicability for use in field testing. (Author)

AD-234 140 N

AD-234 516 Div. 8
(6 Apr 60)

Stanford Electronics Labs., Stanford U., Calif.
ALUMINO-SILICATE GLASS-TO-MOLYBDENUM-[COAXIAL] SEAL, by James F. Margiotta. 4 Feb 60, 11p. incl. illus. (Bull. no. 103)
(Contract Nonr-22524) Unclassified report

Descriptors: Glass seals*; Metal seals*; Traveling wave tubes; Containers.

This bulletin describes a technique for the fabrication of a coaxial seal between molybdenum and an alumino-silicate glass (Corning No. 1720). The materials used in this seal are advantageous because they are non-magnetic and will withstand a subsequent brazing temperature of 780°C and bakeout temperature of 700°C. These particular seals were used as r-f connections in a traveling-wave tube. It was found necessary to maintain a minimum wall thickness of 0.017" for this size (0.187") molybdenum outer conductor in order to eliminate the frequent development of leaky lateral cracks in molybdenum tubing of lesser wall thickness. It was also found necessary to examine with a microscope the inside of the molybdenum outer conductor around the area to be sealed and to reject all parts that showed evidence of having (over)

lateral striations. This was done to prevent leaking at the glass-to-metal interface of the seals. The coaxial seals withstood subsequent furnace brazing (at 779°C) into a molybdenum-traveling-wave-tube envelope; using silver-copper eutectic solder. Corning No. 1723 glass may be substituted for Corning No. 1720 glass without changes in techniques or temperatures. (Author) (See also AD-225 855)

AD-234 516 N

AD-234 660 Div. 14, 17, 26
(11 Apr 60)

Sperry Gyroscope Co., Great Neck, N. Y.
METAL-TO-CERAMIC SEAL TECHNOLOGY STUDY, by S. Cole, Jr., H. Larisch, and E. Cheatham. Quarterly technical note no. 1, 30 June-30 Sep 59. Oct 59, 82p. incl. illus. tables, 26 refs. (Rept. no. NA-8240-8184-1)
(Contract AF 30(602)2047, Proj. 4506)
(RADC TN 59-370) Unclassified report

Descriptors: Seals*; Metal seals*; Tests; Tensile properties; Materials; Ceramic materials; Stresses; Bonding; Thermodynamics; Adhesion; Metals.

After a preliminary literature study, an intensive metallizing investigation was planned and is now in progress. Initially, four ceramic bodies of varying Al_2O_3 content have been selected and are now being used in this program. Over 300 metallizing compositions were formulated after consideration of thermodynamic principles, and chemical and physical adherence mechanisms. A testing investigation was planned, two phases of which are in progress (peel testing and com- (over)

pression testing). These programs are designed to render data which is to be correlated with standard ASTM tensile tests. New tensile test specimen des. , i have been accomplished. A stress investigation was planned and is in progress. Theoretical considerations for both thin and thick wall seals have been postulated. A test seal specimen has been fabricated and is under investigation for experimental data. (Author) (See also AD-235 301)

AD-234 660 N

AD-234 721L Div. 14, 26
(11 Apr 60)

Rock Island Lab., Ill.
THE EFFECT OF VOLATILE CORROSION INHIBITORS ON MIL-B-131 BARRIER MATERIALS, by E. S. Burke. 2 Feb 60, 23p. incl. illus. tables (Proj. TB4-006G; Rept. no. 60-294) Unclassified report

Notice: Only Military Offices may request from ASTIA. Others request approval of Diamond Ordnance Fuze Labs., Wash. 25, D. C., Attn: ORDTL 06.33.

Descriptors: Corrosion inhibition*; Packaging*; Plastics*; Vinyl chlorides; Polymers; Ethylenes; Esters; Films*; Seals*; Water vapor; Tests.

MIL-B-131 barrier materials with polyvinyl chloride, polyethylene and metallized polyethylene-polyester heat sealing films, respectively, and unsupported films of these plastics were placed in contact with paper impregnated with dicyclohexyl ammonium nitrite; a mixture of urea, sodium nitrite and monoethanolamine benzoate; dicyclohexylamine caprylate; and an amine carboxylate and exposed to temperature (over)

cycling for various periods of time. The effect of these conditions on the water resistance of the seal, moisture vapor transmission rate, delamination of the plastic film and sealability of the barrier material was measured as well as the effect of this exposure on the tensile and elongation properties of the unsupported plastic films. The results indicate that at least one of the above properties of polyvinyl chloride and polyethylene coated MIL-B-131 barriers are adversely affected by all of the volatile corrosion inhibitors except the one consisting of a mixture of urea, sodium nitrite and monoethanolamine benzoate. The MIL-B-131 material utilizing a metallized polyester-polyethylene heat sealing film was adversely affected only by the dicyclohexylamine caprylate inhibitor. (Author) (See also AD-160 468)

AD-234 721L

AD-234 860 Div. 22
(13 Apr 60)

Rock Island Arsenal, Ill.
105mm RECOIL MECHANISM (NITROGEN VENT), by E. C. Gere. 15 Feb 60, 17p. incl. illus. tables (Proj. no. TW-109)
(Rept. no. 60-302) Unclassified report

Descriptors: Recoil mechanisms*; Howitzers*; Tests; Pneumatic packings*; Pressure.

A 105-mm recoil mechanism (nitrogen vent) was manufactured and tested, which was designed to allow any nitrogen, leaking past the first high pressure seal in the floating piston, to escape rather than contaminate the oil. The recoil mechanism was tested at Rock Island Arsenal by shop hydraulic test, on powder-type testing machine and in laboratory cold room test. The 105-mm recoil mechanism (nitrogen vent) with pressure packings functioned satisfactorily at normal temperatures but not at sub-zero temperatures. Chevron packings failed at normal temperature; therefore, they were not tested at sub-zero temperature. It was recommended that a study be made to determine the adaptability of a metallic bellows-type recuperator, similar to that in use in the 115-mm, XM70 and XM70E1 boosted rocket launchers. (over)

Open-ended Terms: 105mm, M2A1 howitzer.

AD-234 860 N

AD-234 863 Div. 22, 3, 26
(13 Apr 60)

Feltman Research and Engineering Labs., Picatinny Arsenal, Dover, N. J.
PRESS-FIT CLOSURES FOR 8-INCH T174 CHEMICAL SHELL, by Walter J. Puria. Mar 60, 23p. incl. illus. table (Ord. proj. TW-125; DA proj. 504-03-074)
(Technical rept. no. 2678) Unclassified report

Descriptors: Chemical projectiles*; Seals; Projectiles; Manufacturing methods; Lubrication; Thickness; Surfaces; Pressure; Interference.

This report presents the results of a study to establish the optimum burster casing wall thickness at the press-fit for the 8-inch T174 chemical shell. Since it is known that leakage of the shell filler is usually associated with "galling," or abrasion, of the press-fit surfaces, it is the objective of this study to determine the burster casing wall thickness which produces the least amount of galling of the press-fit surfaces when assembled. As this study does not cover variations in degree of interference, surface finishes, or entrance radii, these were kept as nearly constant as possible and representative of established practice for chemical shell press-fits. Only the wall thickness (over)

was varied. Rate of pressing the parts together and lubrication of surfaces were also representative of actual practice. Traces were made of assembling load versus travel, using electrical strain gage equipment, to determine smoothness of assembling and magnitude of assembling load. The assemblies were then sectioned longitudinally, and the press-fit surfaces were photographed to note the degree of abrasion. (Author)

Open-ended Terms: 8 in., T174 Projectiles.

AD-234 863 N

AD-234 840 Div. 14, 7, 22
(14 Apr 60)

Naval Civil Engineering Lab., Port Hueneme, Calif.
PARAFFIN AS A CONDUIT SEAL AGAINST BLAST PRESSURE, by H. G. Aggas. Final rept. on type C. 18 Feb 60, 13p. incl. illus. (Technical rept. no. 074) (Proj. NY 340 030-11)
Unclassified report

Descriptors: Waxes*; Sealing compounds*; Blast; Pressure; Tests; Seals*; Explosions; Countermeasures; Resistance.

The blast resistance of paraffin as a sealing compound in Crouse-Hinds explosion-proof sealoffs was investigated. These sealoffs, which had approximately one-half of their cross-sectional area filled with standard electrical wire, were sealed with paraffin and tested in the blast simulator. Paraffin is an effective seal for resisting dynamic and static pressures to at least 150 psi at an average temperature of 88°F. The sealoffs were subjected to dynamic pressures up to 170 psi with no damage to the wax seal. (Author)

Open-ended Terms: EZS3 Sealoffs, EZS6 Sealoffs, EZS8 Sealoffs, EZS10 Sealoffs, Aristowax.

AD-235 024 Div. 26, 1, 12
(18 Apr 60)

North American Aviation, Inc., Downey, Calif.
HYDRAULIC AND PNEUMATIC FITTING AND TUBING TEST PROGRAM, by L. L. Seibel and V. W. McGillen. Final rept. Nov 59, 184p. incl. illus. tables.
(Contract AF 33(600)36168)
(WADC TR 59-267) Unclassified report

Descriptors: Pipe fittings*; Steel tubing*; Stainless steel; Pipes; Seals; Vibration; Temperature; Tests; Pressure; Pneumatic systems; Hydraulic systems; Joints.

This report describes and gives results of a 4-year fitting and tubing test program. Comparative tests were conducted of different types of tube fittings under conditions of severe vibration and extreme temperature. Most of the results are for the MS flared, the MS flareless, and the "super" flareless types. Test specimens were composed of two fittings of the same type and a tubing configuration considered typical of the needs for a missile system. Results for different combinations of materials are also given. Metal boss seals of stainless steel (cres) material were included in these tests to evaluate the effect (over)

of the test conditions on the seals. Development work accomplished on the flared-type fitting is described along with the results of a limited test program. Also, incidental studies are described dealing with flaring methods, stress analysis of the tube specimens, and metallurgical properties of tubing at high temperature. In general, it was found that the MS fittings were superior to the commercial types tested. Another finding was that specimens with MS flared fittings in the small sizes performed better than those with flareless fittings; however, a tendency for the reverse of this was found for larger size specimens. None of the specimen types were found to be satisfactory under all the test conditions imposed. (Author) (See also AD-115 248) (AD-131 645)

AD-235 024 N

AD-235 052 Div. 22, 25
(18 Apr 60)

Cornell Aeronautical Lab., Inc., Buffalo, N. Y.
M-39 COOK-OFF SUMMARY REPORT, by Gerald A. Sterbutzel. 15 Jan 60, 84p. incl. illus. tables (Rept. no. GA-948-C-49)
(Contract DAI 30-115-504-ORD(P)-596) Unclassified report

Descriptors: Automatic weapons*; Aircraft guns*; Heat transfer*; Rings; Seals*; Safety*; Temperature; High explosive ammunition*; Incendiary ammunition*; Projectiles*; Sensitivity; Tests; Flight testing; Jet fighters*; Armament*; Aircraft ammunition*.

Two systems have been developed which are in themselves complete solutions of the cook-off problem in fighter aircraft currently being considered. One of these systems utilizes air flow over the drum to make possible the firing of at least 420 rounds at an ambient temperature of 150° without cook-off. The other - and the simpler solution - requires only a modification of the drum and seal design to make possible the firing of 420 rounds without cook-off under the severest flight conditions in any of the latest fighters. The study of this type seal - known as ring seals - produced the basic solution to the cook- (over)

off problem. Flight tests were conducted on an F-101 fighter aircraft to prove out the safety of ring seals in this type of aircraft. (Author) (See also AD-312 118)

Open-end'd Terms: M39 guns, Cook off, F-100, F-101, 20 mm.

AD-235 052 N

AD-235 381 Div. 14, 26
(25 Apr 60)

Sperry Gyroscope Co., Great Neck, N. Y.
METAL-TO-CERAMIC SEAL TECHNOLOGY STUDY, by S. Cole, Jr., H. Larisch, and E. Cheatham. Quarterly technical note no. 2, 30 Sep-30 Dec 59. Jan 60, 31p. incl. illus. tables (Rept. no. NA-8240-8184-3)
(Contract AF 30(602)2047)
(RADC TN 60-53) Unclassified report

Descriptors: Seals*; Metal seals*; Materials; Tests; Stresses; Heat treatment; Tensile properties.

A study was conducted to advance ceramic-to-metal seal technology to the point where stronger seals (seals with bond strengths approaching tensile strengths of 25,000 psi, the strength of the ceramic member) can be developed and produced with a higher degree of uniformity and reliability than has been previously attained. The demands for seals with increased electrical, mechanical, and thermal requirements are being imposed by present and future high-power microwave applications, e.g., electron tubes. A literature survey of metal-ceramic seals was completed. Two hundred metallizing mixtures were (over)

milled and sintered at various temperatures. Samples sintered at 1500°C were adherence tested; relatively good results were obtained. At least 22 compositions provided better adherence than standard Mo-Mn mixtures. Although scratch testing is only a limited measure of ultimate metal-ceramic seal strength, it indicates that the metallizing composition should lead to an optimum mix to high seal strengths. Adherence tests verified the principle that low Al_2O_3 bodies (94-96%) more easily form metallizing bonds than high Al_2O_3 (99.5%), but in all bodies, metallizing composition can affect bonding within a wide range. Preliminary data indicate that Mo-oxide additions do not observably improve bonding. Other additives appear promising including Ca, Mn, SiO_2, Fe, Zr, Ti, CeO, Li, and various glass additives. The seal stress investigation has predicted the exact stress condition in a circumferential seal. Equipment to measure and predict residual seal stress was assembled and an investigation was begun.

AD-235 381 N

AD-235 681 Div. 26, 8
(2 May 60)

Bulova Research and Development Labs., Inc., Woodside, N. Y.
AUTOMATIC MACHINE FOR SEALING QUARTZ CRYSTALS IN TYPE HC-6/U, HC-13/U AND HC-18/U CRYSTAL HOLDERS, by R. F. Giglio Tos. Final engineering rept., 24 June 57-30 Apr 60. 1v. incl. illus. tables.
(Contract DA 36-039-sc-72775) Unclassified report

Descriptors: Quartz crystals*; Crystal holders*; Welding*; Seals; Machines*; Automatic; Tests; Instruction manuals; Design.

This document includes:
Raytheon Mfg. Co., Waltham, Mass.
INSTRUCTION MANUAL 2-206. MODEL K WELDING HEAD. Aug 58, 13p. incl. illus.
Raytheon Mfg. Co., Waltham, Mass.
INSTRUCTION MANUAL 2-314. MODEL 1103, DC WELDING POWER SUPPLY. Sep 56 (rev.1) 18p. incl. illus. tables.
[Dudco Div., New York Air Brake Co., Hazel Park, Mich.]
COMPOUND HIGH VACUUM PUMPS KC-15 AND KC-46. 1 June 59, 22p. incl. illus. (over)

Veeco Vacuum Corp., New Hyde Park, N. Y.
THERMOCOUPLE CONTROLLER TYPE TC-1, TC-3, 22 Oct 58, 3p. incl. illus.

An automatic sealing machine is described for the resistance welding of Quartz Crystal Holder types HC-6/U, HC-13/U and HC-18/U, while in an inert atmosphere. The machine consists of an automatic evacuation and nitrogen flushing chamber, an inert atmosphere chamber, welding head, D. C. welding power supply and controls. A history of the project is given with pilot run data, test results evaluation of 50 each HC-6/U and HC-18/U holders, as well as a discussion of design and manufacturing problems for both machine and holders. Installation, operation and maintenance instructions are given. (Author) (See also AD-231 350)

AD-235 681 N

AD-236 006 Div. 7
(5 May 60)

Bell Telephone Labs., Inc. [Whippany, N. J.]
RESEARCH AND DEVELOPMENT WITH HERMETICALLY SEALED CONTACTS, by O. M. Hovgaard. Quarterly progress rept. no. 11, 15 Sep-14 Dec 59. 14 Jan 60, 12p. incl. illus. tables.
(Contract DA 36-039-sc-72835) Unclassified report

Descriptors: Electric switches*; Design; Model tests; Electric relays*; Miniature electronic equipment; Seals.

Studies of exploratory coordinate arrays of latching relays using G29 miniature switches have been completed. A 3-by-3 array of 4-wire crosspoints showed promising transmission characteristics at 0.5 megacycles and the possibility of usefulness up to 4 or 5 megacycles. The 8-by-8 switching array exhibited restrictive margins with respect to magnetic interference between crosspoints and the need for close selection of switches for individual crosspoints. Magnetically latching relays providing 2 and 4 make contacts were also studied. (Author) (See also AD-230 935)

AD-236 067 Div. 8, 26
(9 May 60)

Eitel-McCullough, Inc., San Bruno, Calif.
PILOT LINE OF AUTOMATIC PROCESSING AND ASSEMBLY MACHINES FOR CD RECEIVING TUBES AND TESTING AND EVALUATION OF CD TUBES, by Michael M. Filmer. Final engineering rept. [1960] 1v. incl. illus.
(Contract AF 33(600)32483) Unclassified report

Descriptors: Electron tubes*; Production; Manufacturing methods; Machines; Cathodes (Electron tubes); Test equipment; Ceramic materials; Brazing; Diffusion; Seals; Spot welds.

A line of automatic processing and assembly equipment for CD receiving tubes was designed and constructed to prove the feasibility of high-volume automatic production. The four assembly machines are of the rotary indexing type, and each stacks the parts of a particular sub-assembly in jigs. The final assembly step for a complete tube is also performed on one of these machines. The additional machines constructed were a machine to size grid assemblies, a machine to pump and seal the tubes by the diffusion sealing process, an electri- (over)

cal testing machine to perform 13 tests automatically, and an electrical aging machine. A production ceramic metalizing machine and a cathode spray machine were constructed. Investigations were conducted to select the strongest available ceramic bodies, and a change was made from a very high to a lower alumina content body. Metalized ceramic rings cleaned in hot acid lost flexural strength, therefore, change was made to ultra-sonic cleaning. A barrel plating process was able to plate the ceramic within the tolerances required. Testing and evaluation of CD-18's showed that the tube performed satisfactorily except for instability of the cathode assembly which severely limits useful life, particularly under cycling and shock conditions. Design changes are necessary in the cathode structure to produce a satisfactory tube. Refinement and evaluation of the CD-18 was performed, and an objective specification in MIL-E format was prepared. Preliminary studies were made of CD-19 and CD-22 production refinement, and drawings were prepared. (Author) (See also AD-231 863)

AD-236 067 N

AD-236 184 Div. 8, 7
(10 May 60)

Texas Instruments, Inc., Dallas.
HIGH VOLTAGE SILICON RECTIFIERS, by Waylon D. Bryan. Final rept. (Life and Environmental Test rept.) on Phase 1-3, 1 Jan 58-16 Nov 59. 16 Dec 59, 96p. incl. illus. tables.
(Contract AF 33(600)36110) Unclassified report

Descriptors: Crystal rectifiers*; Silicon*; Diodes*; Voltage; Tests; Production; Manufacturing methods; Coatings; Seals; Packaging; Semiconductors; Diffusion; Phosphorous; Boron.

The first diffusion runs of the contract period were made with P-type grown silicon in the 200 Ω cm range. With 45-hour phosphorous diffusions and 6- to 8-mil π-layer thicknesses, the yield of 3,000-volt rectifiers was about 50%. The yield of high voltage cells was considerably increased by using 2,000 Ω cm float-zone starting material and increasing π-layer widths to about 18 mils. The optimum phosphorous bake time at 1250°C was found to be 24 hours. Boron diffusions were standardized as 18 hours at 1250°C. Various configurations of rectifier wafer geometry were investigated. The
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most successful configuration was found to be the round wafer of 88 mils diameter. Etch studies indicated that optimum results are obtained with a 10-second phosphate No. 1 stream etch and a 5-second re-etch when necessary. Studies of surface coatings for the rectifying cells showed that optimum results are obtained by using one full-strength coating of silicone varnish and one coating of silicone rubber. It was found unnecessary to pressurize the 3,000-volt units if they were silastic coated. A dimple closure, applied by hydraulic pressure, was found to give a higher percentage of tabulation seals with the X-286B rectifier than did resistance welded seals. (Author) (See also AD-232 837)

AD-236 184

AD-236 837 Div. 26, 22
(23 May 60)

Rock Island Arsenal, Ill.
INVESTIGATION OF HYDROPNEUMATIC RECOIL MECHANISM PACKING SPRING LOADS, by L. R. Raisbeck. Technical rept. 14 Apr 60, 31p. incl. illus. tables (Rept. no. 60-1149)
Unclassified report

Not releasable to Foreign Nationals.

Descriptors: Howitzers*; Recoil mechanisms*; Pneumatic packings*; Pistons; Tests; Seals*; Effectiveness; Springs*; Hydraulic seals*; Temperature.

Conventional packings for the 75 MM Pack, 105 MM, 155 MM and 8 in. Howitzer Hydropneumatic Recoil Mechanism floating pistons were tested in a laboratory fixture to determine the relationship between axial packing load and radial packing pressure at sub-zero, normal and elevated temperature. Tests were conducted with leather-silver-rubber, and Teflon-aluminum-rubber packings. Results indicated that: (1) the pressure factor (radial packing pressure/axial spring load) varies directly with temperature for both type packings; however,
(over)

sub-zero temperature reduces the pressure factor considerably more than elevated temperature increases it; and (2) the axial spring load on the packings varies directly with temperature if no adjustment in Belleville spring height is made. Both types of the above packings were tested in a 105 MM Howitzer Hydropneumatic Recoil Mechanism to determine the relationship between Belleville spring load and their sealing effectiveness at sub-zero, normal and elevated temperatures under dynamic conditions. Results indicated that: (1) leakage decreases as the packing spring load is increased up to 150% of standard packing spring load; however, the decrease is very small above 125% spring load; (2) leakage is greater at sub-zero temperature than at normal and elevated temperatures for any packing spring load; and (3) Teflon shrouded packings form a more positive seal than leather shrouded packings. (Author)

Open-ended Terms: 75mm, 105mm, 155mm, 8 in.

AD-236 837

AD-237 405 Div. 16, 1
(7 June 60)

Air Crew Equipment Lab., Naval Air Material Center, Philadelphia, Pa.
ENVIRONMENTAL REQUIREMENTS OF SEALED CABINS FOR SPACE AND ORBITAL FLIGHTS. A SECOND STUDY PART 6: EVALUATION OF POTASSIUM SUPEROXIDE CLOSED CIRCUIT REBREATHING SYSTEM DURING PROLONGED CONFINEMENT, by D. A. Mancinelli and E. L. Michel. 24 May 60 [17]p. incl. illus. tables (Rept. no. NAMC-ACEL-418) Unclassified report

Descriptors: Pressurized cabins*; Space flight*; Potassium compounds*; Oxides; Respiration*; Seals; Altitude chambers; Simulation; Oxygen equipment*; Moisture; Carbon dioxide; Oxygen.

An investigation was conducted to evaluate the use of potassium superoxide (KO_2) in a closed breathing system. The successful completion of an 8-day, 6-man, confinement study using KO_2 in a closed breathing system demonstrates the possible application of such a system in space flights involving sealed cabins. By utilizing a specific canister design and chemical granule size and by carefully controlling the temperature and moisture content of the inlet gas to the canister, a sys-
(over)

tem can be operated wherein the proper rate of carbon dioxide absorption and evaluation of oxygen is achieved without the use of a second expendible chemical (LiOH or cardoxide). Since the proper moisture content of the gas for this balance system is on the low side, this should ameliorate the moisture condition in the habitability compartment and in the KO_2 bed which resulted in increased pressure drop with time encountered in this investigation. Within the limits of this experiment it appears that, in a closed respiratory system, KO_2 demonstrates the following desirable characteristics: (a) evolution of oxygen; (b) absorption of carbon dioxide; (c) removal of some H_2O vapor; (d) removal of odoriferous compounds; and (e) possible removal of carbon monoxide and other contaminants. The following other characteristics were encountered: (a) increased pressure drop with time through KO_2 bed, and (b) inability to control the confined space humidity unless additional equipment is used. (Author) (See also AD-229 593)

AD-237 405 N

AD-237 500 Div. 10, 12
(23 May 60)

Borg-Warner Corp., Chicago, Ill.
INVESTIGATION OF PROPELLANT SYSTEMS AND COMPONENTS UTILIZING A HIGH-ENERGY OXIDIZER, by Thomas Lock and Clarence E. Albertson. Quarterly rept. no. 4, 16 July-15 Oct 59, May 60, 88p. incl. illus. tables.
(Contract AF 33(616)6039)
(AFFTC TR 59-44) Unclassified report

Descriptors: Fluorine*; Liquefied gases; Rocket oxidizers*; Handling; Ceramic materials; Cermets; Corrosive liquids; Fluid flow; Test equipment; Heat transfer; Cryogenics; Guided missiles*; Seals.

A series of experiments were made to determine the feasibility of rapidly sintering small fluoride (Combo) ceramic compacts which show promise as bearing and seal materials. A sintering time of about 16 hr at a maximum temperature of 1650° F is required in a continuous tunnel kiln to produce dense structures with hard, smooth surfaces, sharp edges, and high strength. X-ray diffraction studies of the sintered fluoride ceramics revealed a structure consisting of a heterogeneous mixture of Ca_2F_6 and BaF glassy matrix. Fluo-
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ride cermets with additions of Sb metal ited much improved tensile properties. An analytical heat transfer study of the one-phase flow of liquid F in tubes with friction and heat addition is presented to show that the absolute pressure available will be the principle limiting factor in test conditions selection. To maintain a single (liquid) phase heat transfer test, the temperature of the F should not exceed its vapor pressure temperature. This will limit the permissible wall temperature and determine the maximum permissible flow frictional pressure drop if simultaneous heating is applied. The second limiting factor in test condition selection will be the capability of the Lincoln Weld energy supply units. Descriptions are given of the single pass flow rig and the seal and bearing test rig. (See also AD-213 806)

AD-237 500 N

AD-237 833 Div. 14
(15 June 60)

Boeing Airplane Co. [Seattle, Wash.]
DESIGN DATA FOR O-RINGS AND SIMILAR ELASTIC SEALS, by George E. Trepus, Robert S. Roper, and William R. Hickman. Rept. for 1 July-31 Dec 59 on Non-Metallic and Composite Materials. Mar 60, 137p. incl. illus. tables, 39 refs.
(Contract AF 33(616)5722, Proj. 7340)
(WADC TR 56-272, pt. 5) Unclassified report

Descriptors: Seals*; Hydraulic seals; Plastic seals; Rubber seals; Design; Physical properties; Temperature; Tests; Cryogenics.

The study to develop design criteria applicable to O-rings, back-up rings, and other elastomeric seals, and of the groove configurations for these seals was continued. Correlation of the physical properties of the seal material with the seal design was included. Seals fabricated from silicone, Viton A, Viton A-HV, and Viton B(LD234) were used for testing. Ultimate tensile strength, ultimate elongation, and tensile modulus at 20% elongation were determined at room temperature and 400°F. Compounds which were functionally tested at 500°F were tensile tested at 500°F. Measurements were made of compres-
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sion set at 400°, 500°, and 600°F (2 hr exposures) for a number of compounds. Compression deflection and stress relaxation measurements were made on several O-ring samples with unique designs and compared with a solid Viton O-ring. Static and dynamic pneumatic seal tests, dynamic hydraulic seal tests, and high temperature static annulus seal tests were made on a variety of materials. Acceleration of 35 g for 10 min had no adverse effects on AN6290-6 O-rings. The effects of 2 levels of X-radiation [5×10^5 ergs/g carbon (C) and 6×10^5 ergs/g (C)] on O-rings conforming to AN6227B-19(MIL-P-5516A) were studied. A review of the literature concerning the effects of ozone on elastomeric seal materials, a cryogenic seal study by NBS, a literature review of age control techniques, and abstracts of current published and unpublished information on seals are appended. (See also AD-230 658)

AD-237 833

AD-237 928L Div. 22, 14
(17 June 60)

Aberdeen Proving Ground, Md.
OBTURATOR PADS, SYNTHETIC RUBBER, by Donald J. Mavis. June 60, 1v. incl. illus. tables (Proj. WVLT-1/43, rept. no. 1)
Unclassified report

Notice: Only Military Offices may request from ASTIA. Others request approval of Diamond Ordnance Fuze Labs., Wash. 25, D. C. Attn: ORDTL 06.33.

Descriptors: Synthetic rubber*; Materials*; Rings; Obturation; Howitzers*; Tests; Gas seals; Physical properties; Temperature; Corrosion.

The object of this test was to determine the suitability of pads and materials when fired at ambient temperatures (above 50°F) and -40°F. Four series of pads, each with different physical properties, were tested. Five rounds of various zones, generating pressures from 8,000 to 32,000 psi, were fired on each type pad at both the normal and low temperatures, with an additional ten rounds on the last pad of each series, if it appeared to be in good condition. The pads were
(over)

visually inspected before and after each round in order to estimate leakage, if any, and the percentage of seat contact. All pads with the exception of the Code C series (which were not fired at ambient temperature because of their failure at subzero temperatures) performed obturation satisfactorily at both ambient temperature and -40°F. However, the material of the pads was unable to resist cutting by the split rings and the erosive action of the powder gases. Although this condition is not serious after these limited firings, it is not known to what extent the pads would deteriorate upon additional firing. The special "staggered joint" type split rings were unsatisfactory, as they failed to seal the chamber adequately. (Author)

Open-ended Terms: 8 in, M2 howitzers.

AD-237 928L

AD-238 020 Div. 14, 17, 26
(20 June 60)

Armour Research Foundation, Chicago, Ill.
COMPOSITE INORGANIC RESILIENT SEAL MATERIALS, by M. Sabanas. Rept. for Dec 58-Dec 59 on Non-Metallic and Composite Materials. Apr 60, 34p. incl. illus. tables.
(Contract AF 33(616)5793)
(WADC TR 59-338, vol. 2)
Unclassified report

Descriptors: Metal seals*; Fibers; Metallic textiles*; Impregnation; Temperature; Mechanical properties; Metals; Ceramic materials; Polymers; Elastomers; Refractory materials; Manufacturing methods; Seals*; Gaskets*.

The principal objective of this research program is to investigate and develop new concepts for novel and unconventional material combinations which would have resilience, recovery, strength, and chemical resistance at temperatures up to 1000°F. Major emphasis was given to configurations that would enable these materials to be used as static and dynamic reciprocating shaft seals. Composite material combinations consisting of stainless steel and molybdenum fibers impregnated with tin, indium, magnesium ver, and polymeric
(over)

materials were produced. Composites m. d molybdenum fibers impregnated with silver were evaluated as static seals and showed good ability to seal air heated to 1000°F and retained pulsating pressures from zero to 5000 psi. The relationship between the fiber structure, impregnant, and final composite were studied. Impregnation techniques and secondary work processes like machining are outlined in this report. (Author) (See also AD-231 003)

AD-238 020

AD-238 164 Div. 25, 30
(21 June 60)

Laboratory for Insulation Research, Mass. Inst. of Tech., Cambridge.
SOME DEVELOPMENTS IN HIGH-PRESSURE TECHNIQUE, by P. W. Forsbergh, Jr. June 60, 39p. incl. illus. (Technical rept. no. 151)
(Contract Nonr-184110)
Unclassified report

Descriptors: High pressure research*; Pressure; Solids; Gases; Metals; Hydraulic seals; Seals; Measurement.

Various research problems requiring high-pressure techniques were investigated. Consideration is given to solids and gases as pressure media, typical metals used in high-pressure design and methods of making seals (hydraulic and electrically insulating seals). Examples of high-pressure design are presented which include 2-dimensional pressure holders, precompressors, systems with lapped quartz seals, systems with optically flat quartz seals, systems with hydraulically loaded closures, and systems with hydraulically loaded piston gaskets. The measurement of pressure, length, and lattice parameters is discussed.

AD-238 200 Div. 7
(22 June 60)

Bell Telephone Labs., Inc. [Whippany, N. J.]
RESEARCH AND DEVELOPMENT WITH HERMETICALLY SEALED CONTACTS, by O. M. Hovgaard. Quarterly progress rept. no. 12, 15 Dec 59-14 Mar 60. 14 Apr 60, 11p. incl. table.
(Contract DA 36-039-sc-72835, Proj. 3-26-01-000)
Unclassified report

Descriptors: Electric switches*; Electric relays*; Miniature electronic equipment; Seals; Design; Model tests.

Preliminary life tests of G29 miniature sealed contacts operated under conditions simulating those to be experienced in latching types of relays indicate that the life expectancy may be less than previously reported for the nonlatching conditions. The phenomenon appears to be related to the continuous presence of magnetic flux which may increase the opportunity for particles to bridge the open contacts. (Author) (See also AD-236 006)

AD-238 431 Div. 22
(28 June 60)

Aberdeen Proving Ground, Md.
ARCTIC WINTER ENVIRONMENTAL TEST OF OBTURATOR SYSTEM FOR BAG-LOADED CANNON, by E. W. Wagner. July 60 [21]p. incl. illus. table (Proj. OMS 5610.11.701, rept. no. 1)
Unclassified report

Descriptors: Arctic regions; Obturation*; Tests; Asbestos fiber; Guns*; Seals; Gas leaks; Silicones; Design; Effectiveness; Howitzers.

Silicon oil-asbestos pads were tested in arctic environment. No satisfactory fit was obtained even though the pads remained soft and pliable. The pads failed to seal the front split ring and were themselves only partially sealed. No firing was conducted. No further consideration should be given to testing these pads. (Author) (See also AD-203 759)

Open-ended Terms: M1 Howitzers, 8 in, 175 mm, T256 guns.

AD-238 482 Div. 7, 8
(29 June 60)

Amphenol-Borg Electronics Corp., Chicago, Ill.
RESEARCH ON R. F. COAXIAL CONNECTORS, BROADBAND, HIGH TEMPERATURE, RADIATION RESISTANT, by J. V. Malek. Scientific rept. no. 5, 1 Mar-1 June 60. 42p. incl. illus. tables, 10 refs.
(Contract AF 33(616)6243)
Unclassified report

Descriptors: Electric connectors*; Coaxial cables; Broadband; Radio-frequency cables; Radiation effects; Temperature; Design; Seals.

Study and development of hermetically sealing the shell-insulator-contact structure was expanded. In addition to fused silica 3 more materials were studied; Corning No. 1723 glass, forsterite, and alumina. New center contact design configurations were studied, developed, and partially tested and evaluated. Methods for attaching the center contact to the cable center conductor were studied. Brazing of the cable outer conductor to simulated connector shells was studied. Brazing of samples and testing were initiated. Additional study and testing of seals for use at the connector mating faces was accomplished. Experimental plating studies were conducted to obtain a silver coat-
(over)

ing over the base materials of Inconel and stainless steel for the purpose of maintaining high electrical surface conductivity at elevated temperatures. (Author) (See also AD-234 200)

AD-238 482

AD-238 832 Div. 10, 12
(1 July 60)

Borg-Warner Corp., Chicago, Ill.
INVESTIGATION OF PROPELLANT SYSTEMS AND COMPONENTS UTILIZING A HIGH-ENERGY OXIDIZER, by Clarence E. Albertson and Thomas Lock. Final rept. 1 July 58-15 Jan 60. May 60, 1v. incl. illus. tables (Suppl to AD-217 577)
(Contract AF 33(616)6019)
(AFFTC TR 60-9, vol. 2)
Unclassified report

Descriptors: Fluorine*; Liquefied gases; Rocket oxidizers*; Handling; Fluid flow; Heat transfer; Seals; Friction; Test facilities; Test equipment; Guided missiles*; Ceramic materials.

Contents:
Appendix V. Single pass flow rig. Analytical study of one-phase flow of liquid F in tubes with friction and heat addition
Appendix VI. Progress reports 1-4, Aug-Dec 1959, and Final report, 10 Feb 1960, on Determination of thermal conductivity of liquid F by the Engineering Experiment Station of The Ohio State University, Columbus, Ohio
Appendix VII. Detailed operational procedure for the static F com-
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patibility apparatus
Appendix VIII. Proposed operational sequence for friction and wear apparatus
Appendix IX. Proposed ICC specifications for F tank truck
Appendix X. Inventory of single-pass flow rig equipment
Appendix XI. Single-pass flow rig. List of drawings
Appendix XII. Instrumentation: Single-pass flow rig - flow-transfer investigation
Appendix XIII. Cleaning procedures for liquid O system components. Pesco engineering process specification P-144, June 17, 1959
Appendix XIV. Flow-transfer rig passivation
Appendix XV. Single-pass flow rig. Friction factor and pressure drop tests. Sequence of operation
Appendix XVI. Seal and bearing test rig - inventory of equipment
Appendix XVII. Seal and bearing rig. List of drawings
Appendix XVIII. Seal and bearing test rig. Passivation and operational sequence

AD-238 832 N

AD-239 180L Div. 26, 1, 12
(7 July 60)

Chance Vought Aircraft, Inc., Dallas, Tex.
MODULAR HYDRAULIC SYSTEM DEVELOPMENT (PROJECT HYDRATOY) by R. W. Burden, J. W. Browne and others. Quarterly progress rept. no. 5, 5 Dec 59-5 Mar 60. 28 Mar 60, 1v. incl. illus. tables. (Rept. no. AER-EOR-12436)
(Contract NOa(s) 59-6019-c)
Unclassified report

Notice: Only Government Agencies may request from ASTIA. Others request approval of Chief, Bureau of Naval Weapons, Navy Dept., Wash. 25, D. C.

Available on loan only.

Descriptors: Hydraulic systems*; Hydraulic actuators*; Seals; Standardization; Design; Alloys; Machining; Aircraft; Guided missiles; Jet planes; Metal seals*.

Development is continuing on the following metallic static seals (1) Bendix wedge seals, (2) Ni-Ceal (C-seal), and (3) CVA-X seals. Hydraulic package development is reported. Design and fabrication was
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completed of the Ti manifold which houses a 3-way selector valve, 2-way restrictor, and check valve. Design was completed and fabrication is progressing of the actuator with modular end-cap which houses a pressure relief valve, a shuttle valve, and 1-way restrictor. Final design is reported of the stainless steel manifold which houses a priority valve, thermal relief valve, pressure-operated shut-off, solenoid-operated sequence, and a cartridge 4-way, 3-position selector valve. The modular filter envelope was determined, and a detailed procurement specification was released. Engineering efforts on the fluid-carrying structure are reported. The brazed and seam-welded floor panel assembly was satisfactorily pressure tested. Rotary seal testing for swivel joint development was stopped because of leakage resulting from excessive seal wear. The sealed groove beam assembly was completed. Fabrication was completed of the swivel bolt assembly. Consideration was given to joining methods of tubing and sheets. (See also AD-231 223L)

Open-ended Terms: Project Hydratoy.

AD-239 180L

AD-239 394 Div. 26, 7
(13 July 60)

General Electric Co., Utica, N. Y.
RESEARCH AND DEVELOPMENT OF THERMOCOUPLE ENERGY CONVERTERS. Quarterly rept. no. 2, 1 Mar-31 May 60. 12p. incl. illus.
(Contract DA 30-069-501-ORD-2855, Proj. TH2-8100)
Unclassified report

Descriptors: Thermoelectricity*; Seals; Generators; Design; Thermocouples*; Conductivity; Materials; Temperature; Thermal expansion; Tellurides; Lead compounds.

The development of encapsulated thermoelectric (TE) components for use in a TE generator and yielding a specific electrical power output of 0.06 W. cm^3 is continuing. A hot terminal plate, for improved thermal and electrical contact to the hot junction of the capsule, was developed. Several methods for filling capsules with TE materials were investigated. Of these, a neutral atmosphere oven type filling technique, offering the best potentialities, was developed and tested successfully. (Author) (See also AD-237 964)

AD-239 509 Div. 22
(14 July 60)

Watervliet Arsenal, N. Y.
ANALYTIC DEVELOPMENT OF DESIRABLE STRESS DISTRIBUTION CHARACTERISTICS IN SEALING RINGS LEADING TO RECOMMENDATION OF RING NO. 5 FOR APPLICATION TO CLOSED BREECH LAUNCHERS, by M. A. Sadowsky. Mar 60, 1v. incl. illus. tables (DA Proj. no. 5W45-07-004-01; Technical rept. no. WVT RR 6006)
Unclassified report

Descriptors: Seals; Gas seals*; Rings; Mathematical analysis; Stresses; Design; Tests; Guns*.

Laboratory testing of closed breech sealing rings Nos. 1 and 4 and the mathematical analysis of those rings showed that they are prone to elliptic buckling and plastic flow due to stress concentration at the lips. Ring No. 5 was then designed on the basis of mathematical shell theory. It is believed that it will be as nearly free from those faults as possible. It is suggested that ring No. 5 be produced and evaluated experimentally. (Author)

AD-240 397 Div. 1, 14
(2 Aug 60)

Narmco Industries, Inc., San Diego, Calif.
RESEARCH OF TECHNIQUES AND METHODS ON WHICH TO BASE DEVELOPMENT OF HIGH TEMPERATURE GLAZING ATTACHMENTS, by Gerald K. Partain, Floyd A. Pincupo, and Roger K. Wilson. Mar 60, 63p. incl. illus. tables.
(Contract AF 33(616)5703, Proj. no. 1368)
(WADD TR 60-119)
Unclassified report

Descriptors: Transparent panels*; Glass; Windshields; Attachment; Mechanical properties; Thermal expansion; Adhesion; Metal glass adhesives*; Joints; Seals; Spaceships.

This report describes the development of processes for the integral bonding of low expansion glasses (primarily aluminosilicate) to provide for either a load carrying attachment system or a means of accomplishing an airtight seal at the glazing edge. Tests of two types of attachments (a fused glass-to-metal and a brazed glass-to-metal) for Corning Glass Works Code 1723 and Pittsburgh Plate Glass Co. Code 6695 aluminosilicate glasses had bondline shear values exceeding 400 psi at -80°F and 900°F. Thus, the feasibility of these methods
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of attachment has been demonstrated although the degree of practicality is not yet known. Discussions of various aspects of mechanically held and free floating designs are also presented. (Author) (See also AD-110 780)

AD-240 397 N

AD-240 572 Div. 14, 10
(8 Aug 60)

Material Lab., New York Naval Shipyard, Brooklyn.
THE SUITABILITY OF ELASTOMERIC MATERIALS FOR USE IN THE DEVELOPMENT OF HOSES, GASKETS, AND DIAPHRAGMS FOR HANDLING ROCKET PROPELLANTS, by J. Mironov and A. D. Delman. Final rept. 5 May 60. 8p. incl. tables (Lab. Proj. no. 8000-3)
Unclassified report

Descriptors: Rocket propellants*; Rocket oxidizers; Handling; Nitric acid; Amines; Elastomers*; Rubber hose; Rubber gaskets; Fuel hose; Fuel seals; Diaphragms (Mechanics).

Thirty-two different elastomeric materials were investigated to determine their potential suitability for use in the fabrication of hoses, gaskets, and diaphragms necessary for shipboard handling and storage of red fuming nitric acid and mixed-amines. Results indicate that a fluorinated polymer is suitable for prolonged use with red fuming nitric acid, and three other materials are suitable for short term exposure to this oxidizer. While none of the materials show suitability for use involving prolonged contact with mixed-amines, six material types are indicated to be suitable for limited or short term contact
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with mixed-amines. All of the materials are unsatisfactory for applications requiring prolonged contact with mixed-amines. In addition, these data show that three other specimens are suitable for short term use with red fuming nitric acid, while six of the elastomeric materials are suitable for interim use with mixed-amines. (Author) (See also AD-206 707)

AD-240 572 N

AD-240 838 Div. 26, 14, 22
(11 Aug 60)

Rock Island Arsenal Lab., Ill.
INVESTIGATION OF QUAD RING PACKINGS FOR 155 MM GUN CARRIAGE EQUILIBRATORS, by L. R. Rainbeck. 2 June 60, 16p. incl. illus. tables (Rept. no. 60-1785)
Unclassified report

Not releasable to Foreign Nationals.

Descriptors: Gun mounts*; Packaging*; Equilibrators*; Effectiveness; Tests; Rubber seals*; Pneumatic packings*; Synthetic rubber.

Two sets of Minnesota Rubber and Gasket Company Quad Rings were tested as dynamic packings in 155 MM Gun Carriage Equilibrators. All other components in the mechanisms were standard. The equilibrators were subjected to friction and gymnastication tests at subzero, normal and elevated temperatures to determine if the packings were superior to conventional O-rings. Both equilibrators completed 40,000 cycles of gymnastication without failure. The experimental equilibrators had slightly higher frictions than standard equilibrators with O-ring packings. All components were in good condition after completion of the tests. It was concluded that Quad Rings, when used
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in the 155 MM Gun Carriage Equilibrator: (1) seal as well or better than O-rings; (2) have slightly higher frictions than O-rings; (3) eliminate spiral twisting and thereby extend packing life; and (4) do not produce any adverse effects on other components or performance. (Author) (See also AD-35 907)

Open-ended Terms: 155 mm, Gun mounts.

AD-240 838

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compression values are higher for those bodies of the lower alumina content except mixtures of R-39 and R-50 containing SiO_2. Maximum values were slightly over 5000-lb loads. The tensile test specimen was redesigned with a radius of 21/64. Preliminary tensile stress-strain tests were performed at room temperature on several transverse specimens machined from nickel tube and both 18-8 stainless steel bar and sheet stocks. The sheet stock test results were more reproducible. A comparison of test methods indicated that a tensile value of 25,000 psi is equivalent to a compression test value of 5200 lb, drum peel of 17.25 lb, and torque peel value of 6 1/8 in.-lb. The compression test displayed the smallest coefficient of variation. Seal strengths of about 25,000 psi are expected when tensile tests are performed. (See also AD-235 361)

AD-240 980 N

AD-241 564L Div. 26, 30
(30 Aug 60)

Douglas Aircraft Co., Inc., Santa Monica, Calif.
NON-DESTRUCTIVE LEAK TESTING OF SEALED PACKAGES. Monthly progress rept. no. 2, for May 60. 15 June 60, 9p. incl. illus. tables.
(Contract NOa(s) 60-6037-c)
Unclassified report

Notice: Only Government Agencies may request from ASTIA. Others request approval of Chief, Bureau of Naval Weapons, Navy Dept., Wash. 25, D. C.

Descriptors: Electronic equipment; Packaging*; Non-destructive testing*; Test methods; Halogen compounds; Gases; Radioactivity; Scintillation counters; Test equipment; Seals; Gas leaks; Hydrocarbons; Fluorides; Diffusion; Chlorides; Benzenes; Toluenes.

Attempts are being made to establish non-destructive test methods for determining leakage in sealed packages. An investigation was made of a Radiflow unit which uses radioactive gas for leak-testing hermetically sealed electronic items. A package test chamber will be pressurized with radioactive gas and subsequent check with a geiger
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counter will determine which package leaked and, therefore, contains radioactive material. The physical and chemical properties of halogenated compounds were screened to determine those suitable to be inserted in each package just prior to sealing. A halogen detector will be used to detect the presence of halogen gas in an evacuation chamber containing the package being tested. For flexible packages only, the use of a vacuum chamber was investigated with detection of leaks made possible due to pillowing of the packages that do not leak.

Open-ended Terms: Freon, Radiflow.

AD-241 564L

AD-241 565L Div. 26, 30
(30 Aug 60)

Douglas Aircraft Co., Santa Monica, Calif.
NON-DESTRUCTIVE LEAK TESTING OF SEALED PACKAGES.
Monthly progress rept. no. 3, for June 60, 15 July 60, 4p.
(Contract NOa(s) 60-6037-c) Unclassified report

Notice: Only Government Agencies may request from ASTIA. Others request approval of Chief, Bureau of Naval Weapons, Navy Dept., Wash. 25, D. C.

Descriptors: Packaging*; Non-destructive testing*; Test methods; Gas leaks; Seals; Halogen compounds; Gases; Radioactivity; Scin-tillation counters; Hydrocarbons; Test equipment; Electronic equip-ment; Fluorides; Diffusion; Chlorides; Benzenes; Toluenes.

A diffusion cell is being fabricated to be used with a vapor fractometer in checking permeability of barrier materials to the various gases used in the leak tests. In addition to the Freons and other halogenated compounds, a non-radioactive type of Kr will be used. Toxicity de-termination of the halogenated compounds indicated that toxicity will not be a problem under the contemplated conditions. A feasibility (over)

study is to be made of the radioactive gas method. Bags made of 10 packaging materials are to be subjected to gas absorption studies. A simple lever apparatus was built to quantitatively measure the pillow-ing produced in packages upon evacuation of the surrounding atmos-phere. Sample 4- x 4-in bags were tested, some of which were left flat, some half filled, and others filled completely before sealing. All sealed bags showed significant pillowing in the test apparatus. Tests indicated that a reasonably fast evacuation rate will not produce a pillowing effect on flexible packages with small leaks, such as pin-hole or seam leaks of approximately 0.010-in. diameter. The rame also exhibited detectable pillowing under the same vacuum conditions suitable for checking flexible packages. (See also AD-241 564L)

Open-ended Terms: Freon, Radiflow.

AD-241 565L

AD-241 589 Div. 30, 14, 20, 28
(30 Aug 60)

Lockheed Aircraft Corp., Marietta, Ga.
GAMMA IRRADIATION OF AN ELECTRO-HYDRAULIC SERVO TEST LOOP USING ORONITE 8515 HYDRAULIC FLUID, by A. MacCullen. Rept. for Aug 58-June 59, on Nuclear Radiation Effects. May 60, 116p. incl. illus. tables, 13 refs.
(Contracts AF 33(600)32055 and AF 33(600)38947, Proj. 3133)
(WADC TR 59-592) Unclassified report

Descriptors: Hydraulic servomechanisms*; Hydraulic fluids*; Radiation damage; Radiation effects; Gamma rays; Plastic seals; Hydraulic seals; Rubber seals.

The effect of gamma radiation was studied on a specific high-temper-ature hydraulic fluid, Oronite 8515, and test system components operated in an electro-hydraulic servo test loop. Oronite 8515 fluid meets the requirements of MIL-H-8446A for hydraulic fluids in the -65° to $400^{\circ}F$ range. Tests were conducted for 260 hr with an average dose rate of 1.1×10^{7} ergs/g-hr (C). The fluid temperature was maintained at $275^{\circ}F$, and the nominal pressure was 3000 psi. (over)

The Oronite 8515 fluid and the hydraulic loop components with some exceptions performed satisfactorily within the temperature limits and total radiation dose received. Viton A O-rings did not perform the sealing function in a satisfactory manner. The servo valve, in as much as it was the most vulnerable of the components used, could not be considered as completely satisfactory for its intended use. Teflon backup rings in the hydraulic pump and filter performed satisfactorily after receiving radiation doses of 10^{9} ergs/g (C) and 1.3×10^{9} ergs/g (C), respectively. However, during post-irradiation examination, the Teflon rings were found to be brittle and could not be removed. Buna N O-rings performed satisfactorily; no abnormal leakage with these seals was indicated. (See also AD-211 733)

AD-241 589

AD-241 629 Div. 8, 14, 26
(29 Aug 60)

Raytheon Mfg. Co., Newton, Mass.
INVESTIGATION ON CERAMIC SEALS FOR ELECTRON TUBES, by Austin H. King. Final rept. 9 Apr 59-31 Mar 60, 25 Apr 60, 25p. incl. illus.
(Contract AF 19(604)5706)
(AFCRC TR 60-151) Unclassified report

Descriptors: Electron tubes*; Vacuum seals*; Ceramic materials; Seals*; Brazing.

CONCLUSIONS: A study of ceramic sealing techniques, separate from a device design, has greatly simplified the exploration of known sealing techniques. As a result, two practicable sealing techniques have been developed. Primarily, the program has demonstrated the use of an inexpensive test vehicle readily adaptable to any type of seal and testing. The tensile strength testing is a perfectly valid meas-ure, as the compressive force required to hold the part has been found to be very small in comparison to the tensile strengths. Secondly, a lithium-molybdenum metallizing and copper brazing method has been developed that has as good tensile strength as the (over)

ceramic itself, excellent repeatability, and ability to withstand severe environments. Fabrication is fast, at low temperature, with a minimum of steps and in low-cost equipment. This method is applicable to a very wide variety of metals and ceramics, on most types of seal configurations. The metallizing is applicable to very small holes or a flat surface, and has a negligible depth of pene-tration. Thirdly, a low vapor pressure sealing technique with excel-lent strength and reliability has been developed for butt seals, employing double nickel clad titanium in the optimum eutectic pro-portions. Troublesome parts cleaning has been minimized and again considerable tolerance of parts can be accomodated. This seal is accomplished at a lower temperature in a vacuum than the copper melting point, so that sub-assemblies can be made using the lithium molybdenum and copper brazing, and the final seals made by using the double nickel clad titanium reactive seal, without hazard of affecting the copper braze. The seals can be made at suitable tem-peratures for use with oxide-coated emitters for vacuum tube use. (Author)

AD-241 629 N

AD-241 830 Div. 26, 1, 12
(30 Aug 60)

Chance Vought Aircraft, Inc., Dallas, Tex.
MODULAR HYDRAULIC SYSTEM DEVELOPMENT. PROJECT HYDRATOY. PART I, by R. W. Burden, B. L. Evans and others. Quarterly progress rept. no. 6, pt. 1. 5 Mar-5 June 60, 1v. incl. illus. tables. (Rept. no. AER-EOP-12909)
(Contract NOa(s) 59-6019-c) Unclassified report

Available on loan only.

Descriptors: Hydraulic systems*; Hydraulic actuators*; Seals; Standardization; Design; Alloys; Machining; Aircraft; Metal seals*; Guided missiles; Spaceships.

METALLIC SEAL PROGRAM: Development is continuing on (1) a Bendix wedge seal consisting of a 2-part solid ring metallic seal; and (2) a Hi-Coal consisting of a 1-piece formed steel ring with a C-shaped 0.083 in. cross section. Qualification tests on the Bendix seal are being performed with a 6-seal assembly conforming to the 4-gpm, 4-way selector valve envelope. Hi-Coal samples were fabri-cated from Inconel and AM350 stainless steel. Tests on these seals (over)

in closely backed, minimum squeeze cavities resulted in excessive wear. PACKAGE DEVELOPMENT: Design and fabrication were completed on a titanium manifold (housing for 3-way selector valve, 2-way restrictor, and check valve) and an actuator with modular end cap (the end cap houses a pressure relief valve, a shuttle valve, and a 1-way restrictor). Design is complete and fabrication is 90% com-plete on the stainless steel manifold (housing for a priority valve, thermal relief valve, pressure-operated shutoff, solenoid-operated sequence, and a cartridge 4-way, 3-position selector valve). MOD-ULE DEVELOPMENT: Development is reported on check valves, 1- and 2-way restrictors, shuttle valves, thermal relief, pressure switches, priority valves, press shutoff valves, relief valves, 2-, 3-, and 4-way solenoid selectors, solenoid sequence valves, and filters. FLUID CARRYING STRUCTURE: Three shapes for a rotary seal in 3 different materials were fabricated for the rotary swivel joint. Various methods of permanently interconnecting 2-fluid-carrying struc-tural members and fluid-carrying members to a section of conventional tubing were studied. (See also AD-220 180L, AD-241 831)

Open-ended Terms: Project Hydratoy.

AD-241 830 N

AD-241 831 Div. 26, 1, 12
(30 Aug 60)

Chance Vought Aircraft, Inc., Dallas, Tex.
MODULAR HYDRAULIC SYSTEM DEVELOPMENT. PROJECT HYDRATOY. PART II, by J. W. Browne. Material and process development quarterly progress rept. no. 6, pt. 2 (Semi-annual rept. no. 3) 5 Dec 59-5 June 60. 30 June 60, 23p. incl. illus. tables. (Rept. no. AER-EOR-12909)
(Contract NOa(s) 59-6019-c) Unclassified report

Descriptors: Hydraulic systems*; Hydraulic actuators*; Seals; Design; Alloys; Machining; Aircraft; Guided missiles; Spaceships; Extrusion; Brazing; Metal seals.

Work has progressed on both the selection of materials and the devel-opment of satisfactory processes. It can be considered that suitable materials have been tested, evaluated and selected for various module components. For these areas it is recommended that 17-4PH steel be utilized. 440C alloy may be substituted in certain applications not involving hoop tension loads. Type 416C steel although not equivalent to 17-4PH material can be substituted for module housings. This material is somewhat less corrosion resistant than the 17-4PH alloy. (over)

The module housings will of necessity incorporate a hardened valve seat. The metallic seals for such systems will also require plating to reduce the inherent friction. A redesign of the seal to provide more bearing surface and reduce the point loading should alleviate the gall-ing and fretting corrosion problem. The manifold housing exteriors will be plated utilizing cadmium plate or chrome plate to provide the necessary corrosion resistance. The investigation of thread lubricants will be continued. However, there is a certain amount of question that a satisfactory thread lubricant can be obtained which will not flake off in the system. Therefore clearances within servo valves and close tolerance parts must be adequate or satisfactory filters developed to remove these particles. (Author) (See also AD-220 180L, AD-241 830)

Open-ended Terms: Project Hydratoy.

AD-241 831 N

AD-241 908 Div. 8, 14, 26
(6 Sep 60)

Knights, James, Co., Sandwich, Ill.
GLASS HOLDER CRYSTAL UNITS, by Charles W. Snyder. Quarterly progress rept. no. 4, 1 Dec 59-15 June 60, 30 June 60 [17]p. incl. illus. tables.
(Contract DA 36-039-sc-81274) Unclassified report

Available on loan only.

Descriptors: Seals; Crystal holders*; Quartz crystals*; Glass seals*; Vacuum seals; Industrial equipment; Industrial production.

The main effort was spent in evaluating the existing single sealing unit setup and adding improvements and techniques commonly used in the vacuum tube industry. To make a good glass seal that conforms to sizes required, a forming and alignment fixture was fabricated from boron nitride. These jigs and fixtures work well with the limited seal-ing experiments done so far. A "V. M. F." diffusion pump was in-stalled in this setup. It's pumping speed is rated at 5 liter per second, and could produce an ultimate pressure of 1×10^{-6} mm h.g. The improvements made to the existing single unit setup and the limited (over)

number of units sealed so far indicate this technique should suffice for sealing requirements stipulated in the contract. (Author) (See also AD-233 431)

AD-241 908 N

AD-242 285 Div. 10, 4, 25
(9 Sep 60)

Little, Arthur D., Inc., Cambridge, Mass.
HYDROGEN HANDBOOK. A COMPILATION OF PROPERTIES, HANDLING AND TESTING PROCEDURES, COMPATIBILITY WITH MATERIALS, AND BEHAVIOR AT LOW TEMPERATURES. Apr 60, 246p. incl. illus. tables.
(Subcontract to Parker Aircraft Co., Los Angeles, Calif., Contract AF 33(616)6710)
(AFFTC TR 60-19) Unclassified report

Descriptors: Hydrogen*; Gases; Low temperature research; Cryogenics; Nitrogen; Helium; Thermodynamics; Physical properties; Handling; Hazards; Safety; Liquefied gases*; Valves*; Gas valves; Gas leaks; Transportation; Storage; Sources; Liquid rocket propellants*; Handbooks.

This report summarizes (1) experience with and the available technical information on the development of two prototype valves, one for a cryogenic gas service and the other for a cryogenic liquid service (these valves are under development by Parker Aircraft Co. for a Government agency); (2) bibliographical information on the physical
(over)

and mechanical properties of specific construction materials for a temperature range of -420° F to +200 °F (these materials include some austenitic stainless steels and Teflon plastics); (3) the thermodynamic properties of helium, hydrogen, and nitrogen fluids with which the valves may be used or tested; (4) the hazards associated with the transportation and storage of hydrogen and with its use for testing the prototype valves for leakage across the seats; and (5) the sources and availability of hydrogen, and the Los Angeles regulations that apply to its transportation and use. (Author)

AD-242 285 N

AD-242 347 Div. 26
(13 Sep 60)

Naval Weapons Plant, Washington, D. C.
IMPREGNATION OF POROUS METAL CASTINGS, by L. F. Hatakeyama and P. Tarver. Final rept. 15 June 60, 12p. incl. illus. tables (Rept. no. TWPW-T-17-60)
(NAVORD rept. no. 6957) Unclassified report

Descriptors: Porous metals*; Castings; Impregnation*; Porosity; Polymers; Esters; Silicone resins; Seals.

This investigation was conducted to develop a specification for a single impregnant suitable for porous metal castings of various alloys, to replace several existing specifications and to correct current specification deficiencies. A study was made to determine the basic requirements for an impregnant, and tests were performed on various materials to evaluate them against these requirements. It was concluded that there were at least two commercially available resins - American Metaseal 19V5 and Dow-Corning R-7501 - which met the requirements of the proposed specification, appendix A. These materials showed curing weight loss percentages of 1.37 and 2.46, respectively.
(over)

These values were well within the 5 percent limit tentatively set forth in the proposed specification. These materials showed no curing inhibition in the presence of fine, dry chips of aluminum, copper, iron, magnesium, and zinc. The suitability of these materials also was indicated by their ability to maintain an impervious seal in the test specimens after they were conditioned in various environments and machined to reduce their wall thickness. It was further concluded that an impregnant meeting the following requirements would be satisfactory for the intended application: (1) That it be compatible with the usual casting metals - aluminum, copper, iron, magnesium, and zinc alloys. (2) That it be a liquid of low viscosity to facilitate movement of the impregnant into the casting porosity. (3) That it contain no inert solvents, viscosity reducers, or filterable solids in suspension. (4) That it produce no gaseous or liquid by-products on curing or thermosetting to form an impervious seal throughout the porosity of the casting. (Author) (See also AD-198 770)

AD-242 347 N

AD-242 744 Div. 7
(16 Sep 60)

Sonotone Corp., Elmsford, N. Y.
DESIGN AND MANUFACTURE OF SEALED NICKEL-CADMIUM CELLS OF CYLINDRICAL DESIGN TO SIGNAL CORPS TECHNICAL REQUIREMENT SCL-7504, by Francis J. Alliegro and Herbert Bilsky. Quarterly progress rept. no. 2, 1 Mar-31 May 60, 47p. incl. illus. table.
(Contract DA 36-039-sc-84496) Unclassified report

For reference at ASTIA Hq. only. This report cannot be satisfactorily reproduced; ASTIA does not furnish copies.

Descriptors: Alkaline cells*; Nickel; Cadmium; Design; Manufacturing methods; Seals; Cylindrical bodies.

The first prototype cells were completed and initial cycling data was obtained from these cells. Life cycling equipment completed. Second series of prototypes are under construction. Plates in these cells will be overloaded approximately 35%, and positive plates will contain cobalt. These cells are designed to meet a proposed general specification on sealed nickel-cadmium cells. Promising results was obtained
(over)

on substitute for paper separator. Program for testing of glass and teflon seals was begun. (Author) (See also AD-238 200)

AD-242 744 N

AD-242 962 Div. 26, 12, 27
(28 Sep 60)

Rocketdyne, Canoga Park, Calif.
DESIGN, DEVELOPMENT, AND TESTING OF ADVANCED HELIUM PRESSURE REGULATOR PART NO. 551302. VOLUME 1, by R. L. Kenyon. July 60, 1v. incl. illus. tables (Rept. no. R-2356, vol. 1)
(Contract AF 04(647)160)
(AFBMD TR 60-74, vol. 1) Unclassified report

Descriptors: Pressure regulators*; Helium; Propellant tanks; Pressure; Vibration; Temperature; Seals; Tests; Design; Vapor pressure.

The ability of the 551302 regulator to meet the requirements of the model specification was demonstrated by the successful completion of the Performance, Flight Rating, and Qualification Tests. The inability to effectively damp the controller sensor led to the development of a vibration-tolerant pilot valve to minimize the errors introduced into the controller by environmental random and sine wave vibration. The bleed regulator vibration error was minimized by a metal-to-metal coulomb friction damper to simulate the damping usually obtained from O-rings and sliding seals in conventional designs. Metal-to-metal coulomb friction damping was also effectively adapted to the pi-
(over)

loted inlet valve. The extreme ambient and gas temperature range forced the use of metal-to-metal valve seats, which proved thoroughly reliable under all conditions tested. Temperature compensation over a 500°F range was achieved with an adjustable compensator utilizing the difference in the coefficients of linear expansion of two materials. Compensation for forces exerted on the controller sensor due to acceleration was successfully achieved using a counterweighted beam. Pressure energized seals with seat forces that vary in inverse proportion to the applied pressure to provide sufficient force to seal at low pressures and to prevent destructive loading at high pressures were designed. A packing type seal was also developed for low pressures to seal the regulator adjusting screw and at the same time permit freedom of adjustment. (Author) (See also AD-242 963)

AD-242 962 N

AD-243 454 Div. 1, 4, 8
(3 Oct 60)

Air Crew Equipment Lab., Naval Air Material Center, Philadelphia, Pa.
DESIGN CRITERIA FOR EFFICIENT FILLING OF LIQUID OXYGEN CONVERTERS; DEVELOPMENT AND EVALUATION OF, by Francis A. Florio. 29 Aug 60 [15]p. incl. illus. tables (Proj. TED no. NAM AE 51143; Rept. no. NAMC ACEL 439) Unclassified report

Descriptors: Liquefied gases*; Design; Oxygen equipment*; Oxygen; Vaporization; Containers; Storage; Pressure; Temperature; Seals; Vapor pressure.

A proposed method of filling liquid oxygen converters is presented and demonstrated as being feasible in principle and operation. Filling of 5, 10, and 25 liter Dewar flasks are performed under various service and design conditions. In addition, effects of filling pressure, component pre-cooling, and Dewar volume are also investigated. The total loss of liquid oxygen resulting in transfers performed by the proposed method are extremely small and represent approximately .5 to 3.5 per cent of the total amount of liquid transferred. The proposed method is offered as a possible substitute for standard Navy methods
(over)

of filling liquid oxygen converters for aircrew breathing. (Author) (See also AD-206 917)

AD-243 454 N

AD-243 665 Div. 26, 7, 8
(5 Oct 60)

McCoy Electronics Co., Mount Holly Springs, Pa.
INDUSTRIAL PREPAREDNESS MEASURE. MANUFACTURE OF GLASS HOLDER CRYSTAL UNITS CR-(XM-17)/U. Quarterly progress rept. no. 4, 1 May-31 July 60, [11]p. incl. table.
(Contract DA 36-039-sc-81269) Unclassified report

For reference at ASTIA Hq. only. This report cannot be satisfactorily reproduced; ASTIA does not furnish copies.

Descriptors: Quartz crystals*; Crystal holders*; Glass; Production*; Manufacturing methods*; Generators*; Power supplies; Design; Industrial equipment*; Industrial production; Seals.

Processes were designed for the manufacture of the CR-(XM-17)/U glass holder crystal units in accordance with technical requirements. Multiple sealing machinery was constructed and placed into operation. The sealing equipment employs a 2.5 kw high frequency generator specifically designed to satisfy the need of an intermediate RF power generator of high production capability. Preliminary tests with experimental units were made which proved to be successful.
(over)

Open-ended Terms: CR-(XM-17)/U.

AD-243 665 N

AD-243 897 Div. 14
(12 Oct 60)

Battelle Memorial Inst., Columbus, Ohio.
CHARACTERISTICS GOVERNING THE FRICTION AND WEAR BEHAVIOR OF REFRACTORY MATERIALS FOR HIGH-TEMPERATURE SEALS AND BEARINGS, by Lewis B. Sibley, Arthur E. Mace and others. Rept. for 1 Mar 58-31 May 59 on Ceramic and Cermet Materials. May 60, 47p. incl. illus. tables.
(Contract AF 33(616)3996, Proj. 7350)
(WADD TR 60-54) Unclassified report

Descriptors: Refractory materials; Seals*; Bearings*; Thermal stresses; Friction; Erosion; Cermets*; Aluminum compounds; Oxides; Chromium; Molybdenum; Ceramic materials*; Silicon compounds; Carbides; Titanium compounds; Nickel.

Ceramic and cermet materials were evaluated for unlubricated wear resistance at high sliding speed (100 to 200 fpm) and low unit load (5 to 50 psi) in 1000 to 1800° F air. A statistical correlation was obtained between the measured wear rates under these conditions and the coefficient of friction, the thermal-stress-resistance, and the thermal diffusivity of the mated materials on which wear predominated. A
(over)

mechanism of wear has been evolved based on the above correlation and on the study of friction and wear surface temperature fluctuations using special transducers and color-motion-picture photography. During high-speed sliding, wear appears to be induced by the inability of ceramic and cermet materials to resist thermal stresses produced by temperature gradients within each rubbing surface between small asperities or hot spots in frictional contact and the body of the respective materials. The wear rate is influenced by the configuration of the rubbing parts and by the thermal-stress-resistance properties of the materials. Promising materials for high-temperature high-speed sliding bearings and seals include Al_2O_3-Cr-Mo cermets, SiC ceramics, and TiC-Ni-Mo cermets. (Author) (See also AD-202 787)

AD-243 897 N

AD-243 966 Div. 14, 29
(12 Oct 60)

Wyandotte Chemicals Corp., Mich.
DEVELOPMENT OF A FABRIC TO PROVIDE PERSONAL PROTECTION FROM TOXIC MATERIALS, by Raymond G. Spain and Alexander V. Sanger. Rept. for 1 May 59-30 Apr 60 on Air Force Textile Materials. June 60, 39p. incl. illus. tables.
(Contract AF 33(616)6501, Proj. 7320)
(WADD TR 60-198) Unclassified report

Descriptors: Protective clothing*; Dacron*; Coatings*; Jet engine fuels; Handling; Hazards; Textiles; Nylon; Cotton textiles; Elastomers; Fluorides; Seals; Synthetic fibers*; Tests; Vulcanization.

Research directed to the development of a coated fabric to resist the chemical effects of and be impermeable to high energy fuels (HEF) is described. The program included the design and fabrication of special test equipment and of laboratory fabric coating apparatus. A base fabric (Dacron) was selected on the basis of resistance to HEF, and the majority of the effort was directed toward the coating of this fabric with fluoroelastomer based compositions and the necessary subsequent
(over)

testing. A coated fabric was approved and 100 yds.[2] of this material were prepared and delivered. (Author)

Open-ended Terms: HEF fuels.

AD-243 966 N

AD-244 668 Div. 26, 8, 14
(25 Oct 60)

Corning Glass Works, N. Y.
PILOT PRODUCTION OF GLASS CRYSTAL HOLDERS (HC-XM-2)/U; HC-(XM-3)/U; HC-(XM-4)/U, by R. K. Whitney. Quarterly progress rept. no. 7, 1 Dec 59-31 Aug 60. [24]p. incl. illus. tables.
(Contract DA 36-039-sc-81255) Unclassified report

Descriptors: Crystal holders*; Quartz crystals*; Production; Industrial production; Manufacturing methods*; Glass; Seals.

Research was continued on equipment and processing techniques for the production of pilot run quantities of glass crystal holders HC-(XM-2)/U, HC-(XM-3)/U, and HC-(XM-4)/U. Samples of the HC-(XM-2)/U holders were sealed with quartz crystals having frequencies of 2.85, 40, and 70 mc; HC(XM-4)/U holders were sealed with 10 and 70 mc crystals; samples of each type were sealed without crystals. Frequency and resistance before and after sealing were tabulated. The units passed the following environmental tests per MIL-C-3098B, MIL-H-10056B, and MIL-H-10056C: visual and mechanical inspection, thermal shock, immersion, leakage (performed on units without quartz), pin alignment, seal strength, insulation resistance, leakage,
(over)

glass seal visual, mechanical shock, vibration, moisture resistance, and salt spray resistance.

AD-244 668 N

AD-245 318 Div. 8, 25
(3 Nov 60)

Keystone Electronics Co., Newark, N. J.
CRYSTAL UNITS CR-XM-30/U MINIATURE GLASS HOLDER, by Herbert Plumhoff. Quarterly rept. no. 1, 30 June-30 Sep 60. 5p.
(Contract DA 36-039-sc-85940) Unclassified report

For reference at ASTIA Hq. only. This report cannot be satisfactorily reproduced. ASTIA does not furnish copies.

Descriptors: Crystal holders*; Crystals; Semiconductors; Transistors; Glass; Seals; Production; Manufacturing methods; Quartz crystals; Miniature electronic equipment*.

Development was begun on techniques and equipment for sealing quartz crystal units in miniature glass crystal holders at a rate of 120 units per 8-hour day. Equipment for sealing miniature glass crystal holders was set up and placed into operation. A limited amount of crystal units have been sealed. (Author)

Open-ended Terms: CR-XM-30/U.

AD-245 377L Div. 26, 30
(4 Nov 60)

Douglas Aircraft Co., Inc., Santa Monica, Calif.
NON-DESTRUCTIVE LEAK TESTING OF SEALED PACKAGES. Monthly progress rept. no. 4 for July 60. 15 Aug 60 [6]p. incl. illus.
(Contract NOa(s) 60-6037-c) Unclassified report

Notice: Only Government Agencies may request from ASTIA. Others request approval of Chief, Bureau of Naval Weapons; Navy Dept., Wash. 25, D. C.

Descriptors: Electronic equipment; Packaging*; Non-destructive testing; Test methods; Containers*; Gas leaks; Diffusion; Test equipment.

Work with the vacuum chamber method of testing resulted in development of a device with which measurement of expansion of a sealed metal can in a vacuum is graphically indicated on a scale. This method of leak-testing looks very promising, since it has now been shown that it may be used on metal cans as well as on envelopes made of flexible materials. Tests were made on leak detection by electronically "sniffing" traces of various crystalline organic halogen com-
(over)

pounds which had been sealed into packages. A better appreciation of practical problems associated with the use of each has been gained. One compound melted too close to ambient room temperature and spread as a liquid, damaging the package. A diffusion cell for use with the Perkin-Elmer Vapor Fractometer in checking gas permeability of barrier materials has been made. Some minor problems involving choice of packing and carrier gas to be used with this instrument were met, but are expected to be overcome shortly.
(Author)

AD-245 377L

AD-245 424 Div. 14
(4 Nov 60)

Trionics Corp., Madison, Wis.
HEAT RESISTANT, ELASTOMERIC MATERIAL COMBINATIONS AND FORMS, by Otto L. Forchheimer. Final rept. 8 Jan 59-7 Jan 60. 31 Jan 60, 42p. incl. illus. tables.
(Contract NOa(s) 59-6083-c; Continuation of contract NOa(s) 58-437-c)
Unclassified report

Descriptors: Silver compounds*; Chlorides*; Physical properties; Chemical properties; Mechanical properties; Deformation; Elastomers*; Glass textiles; Temperature; Gaskets; Radiation effects; Heat resistant polymers.

Additions of small amounts of certain inorganic materials (e.g., potassium dichromate), in conjunction with annealing, have been found to greatly enhance the ductility of silver chloride. The resultant compositions are proposed for use with metal and inorganic (glass) fiber elastic elements to provide heat resistant-elastomeric material combinations and forms. Such composites with silver chloride matrix materials will be serviceable in the temperature range of 500 to 750 degrees F, which is a considerable extension over the service range of present metal- and glass-organic combinations. (Author)

AD-245 449 Div. 30
(7 Nov 60)

General Electric Co., Cincinnati, Ohio.
FOURTH RADIATION EFFECTS SYMPOSIUM, 15-18 SEPTEMBER 1959. VOLUME IV. [1960] lv. incl. illus. tables.
(Sponsored by Air Research and Development Command, U. S. Air Force.) Unclassified report

Descriptors: Radiation damage*; Electronic equipment; Metals; Corrosion; Fuel tanks; Seals; Sealing compounds; Organic materials; Stresses; Nuclear power plants; Nuclear propulsion*; Test methods; Symposia.

Contents:
Gamma irradiation of X-211 engine accessory systems
Airborne electronics for nuclear powered vehicles
The influence of nuclear radiation on the corrosion of metals
Dynamic radiation effects testing methods
O-ring testing in mixed field irradiation
Designing for minimum activation
Combined effects of nuclear radiation and dynamic mechanical stress fields in organic materials
(over)

Radiation effects - a major influence in designing electronic systems for use in nuclear powered aircraft
Radiation effects on aircraft pressure and fuel tank sealants

AD-245 449 N

AD-246 034L Div. 26, 30
(17 Nov 60)

Douglas Aircraft, Co., Inc., Santa Monica, Calif.
NON-DESTRUCTIVE LEAK TESTING OF SEALED PACKAGES.
Monthly progress rept. no. 5 for Aug 60. 15 Sep 60 [9]p. incl. illus.
(Contract NOa(s) 60-6037-c) Unclassified report

Notice: Only Government Agencies may request from ASTIA. Others request approval of Chief, Bureau of Naval Weapons, Navy Dept., Wash. 25, D. C.

Descriptors: Packaging*; Seals; Electronic equipment; Non-destructive testing; Test methods; Containers*; Gas leaks; Diffusion; Test equipment.

A small AN can (AN 8029-42) was subjected to vacuum conditions of approximately 1 cm. of mercury in a bell-jar and expansion was easily detectable by the same procedures previously used in testing for leakage in ordinary tin cans and flexible envelopes. More sensitive expansion detection techniques are possible and are being developed. Work is now progressing at a st- pace in the evaluation of gas transmission rates of flexible barrie- -erials. Problems (over)

with the chromatograph have been smoothed and no further delays are anticipated. In view of encouraging res- f testing either by the straight vacuum method, or by the use of halogen compounds, work on the inherently more complicated and costly method of testing with radioisotopes has, at least for the present, been discontinued. (Author)

AD-246 034L

AD-246 574 Div. 14
(29 Nov 60)

Material Lab., New York Naval Shipyard, Brooklyn.
INVESTIGATION OF VITON A ELASTOMERS FOR USE WITH OXYGEN AT A PRESSURE OF 3,000 PSI, by S. A. Eller and A. A. Stein. Final technical rept. 15 Nov 60, 9p. illus. tables.
(Lab. Proj. no. 6172-1) Unclassified report

Descriptors: Elastomers*; Gaskets*; Fluoro carbons; Oxygen equipment*; Pressure; Physical properties; Aging; Oxygen*; Test equipment; Test methods; Effectiveness; Gas seals.

Apparatus and test procedures were developed to condition specimens of elastomeric material in high pressure oxygen. Following this, 5 samples of Viton A material were subjected at room temperature to oxygen at a pressure of 3,000 psi for 500 hours. Tests made on the samples after this aging period indicated that the Viton A materials had suffered a loss in tensile strength, considerable volume swell, a decrease in hardness and an increase in compression set. These effects are apparently due to absorbed oxygen which is released from the specimens at a slow rate at room conditions but at an accelerated rate when aged in oil at 212F. In addition, three of the five samples (over)

aged in high pressure oxygen were cracked following the compression set test. As the Viton A samples tested herein are unsuitable for use as O-ring gaskets in high pressure oxygen systems, development work will continue using other types of elastomers compounded by the Material Laboratory. (Author)

Open-ended Terms: Viton A.

AD-246 574 M

AD-246 644 Div. 14, 8
(28 Nov 60)

Sperry Gyroscope Co., Great Neck, N. Y.
METAL-TO-CERAMIC SEAL TECHNOLOGY STUDY, by S. S. Cole, Jr., H. W. Larisch and others. Quarterly technical note no. 4, 31 Mar-30 June 60. July 60, 40p. incl. illus. tables (Rept. no. NA-8240-8184-4)
(Contract AF 30(602)2047, Proj. 4506)
(RADC TN 60-192) Unclassified report

Descriptors: Electron tubes*; Seals*; Ceramic materials*; Metals; Mechanical properties; Tensile properties; Metal seals*.

Adherence and torque peel tests indicated that the majority of the metallizing mixtures yield optimum strengths when sintered at 1500 C. The compression testing program indicated those mixtures which will produce high ceramic-to-metal seal strengths. An individual value of 25,400 psi and an average of 3 duplicate specimens of 21,400 psi in tension were obtained. A photoplastic investigation indicated that (1) Sperry's Tensile Design No. 1 is not subject to the shoulder breaks that are common in the ASTM tensile specimens and (2) in the seal area of all three tensile designs considered, a bending moment (over)

is present which produces nonlinear stress at the seal, giving low values in tension. It was thought that a more realistic tensile strength value from Tensile Design No. 1 would be given by a factor of 1.33 times the indicated strength.

AD-246 644 M

AD-247 023 Div. 13
(7 Dec 60)

Aberdeen Proving Ground, Md.
ROAD SHOCK, VIBRATION AND FORDING TEST OF MISSILE TEST SHOP AND ASSAULT FIRE COMMAND CONSOLE OF THE HAWK MISSILE SYSTEM, by O. B. Ford. Dec 60, 1v. incl. illus. tables
(Rept. no. DPS-103) Unclassified report

Descriptors: Guided missiles*; Test equipment; Tests; Shock, Vibration; Trailers; Accelerometers; Transportation; Moistureproofing; Surface to air; Shock resistance; Seals*; Effectiveness; Maintenance vehicles; Fire control systems*.

The missile test shop and the assault fire command console for the Hawk missile system were tested. These later models incorporate improvements over the prototypes; therefore additional vibration and fording tests were required. The objectives of the test were to measure the magnitude and frequency of acceleration imparted to the two units during transportation over the standard Munson vibration course and to determine the ability of these units to withstand the shock and vibration to which they were subjected. Through fording operation, (over)

the effectiveness of the sealing methods provided for water protection were also determined. (Author)

Open-ended Terms: HAWK.

AD-247 023 M

AD-247 173L Div. 26, 1, 13
(8 Dec 60)

Chance Vought Aircraft, Inc., Dallas, Tex.
MODULAR HYDRAULIC SYSTEM DEVELOPMENT (PROJECT HYDRATOY) by R. W. Burden, J. W. Browne and others. Quarterly progress rept. no. 7, 5 June-5 Sep 60. 1v. incl. illus. table (Rept. no. AER-EOR-13104)
(Contract NOa(s) 59-6019-c) Unclassified report

Notice: Only Government Agencies may request from ASTIA. Others request approval of Chief, Bureau of Naval Weapons, Navy Dept., Wash. 25, D. C.

For reference at ASTIA Hq. only. This report cannot be satisfactorily reproduced; ASTIA does not furnish copies.

Descriptors: Hydraulic systems*; Hydraulic actuators*; Seals; Standardization; Design; Alloys; Metal seals*; Packaging; Aircraft; Guided missiles; Spaceships.

METALLIC SEAL PROGRAM: The qualification of class A size Bendix wedge seals was completed. Indications of wear on the class (over)

A seals influenced the decision to increase the hardness of class C seals for comparison evaluation. Efforts on Hi-Ceal configurations included interim testing and qualification of the 0.015-in. wall thickness 30455 Hi-Ceal; testing with AM350 material Hi-Ceals and with electroless Ni and electrolized Hi-Ceals in an effort to reduce wear problems of the AM350 seal; and an investigation of seat material tests with 17-4 ph, 440C, and 416 stainless steels and seals for comparison of wear properties. Tests indicated that 440C and 416 materials are less susceptible to seal wear. PACKAGE DEVELOPMENT: Designs were completed for a stainless steel manifold. Additional research is being conducted on thread lubricants in an effort to reduce the installation torque required for installation of the modular valves into the packages. MODULE DEVELOPMENT: The determination of unit envelopes and preparation of a detailed procurement specification, and the receipt and evaluation of competitive proposals and selection of vendors were completed. (Author)

Open-ended Terms: Hydratoy Project.

AD-247 173L

AD-247 750L Div. 26, 14
(30 Dec 60)

Industrial Lab., Mare Island Naval Shipyard, Vallejo, Calif.
RUBBER O-RING MATERIAL FOR HYDRAULIC SERVICE, by Arthur W. Scott. Specification development progress rept. 13 Dec 60 [12]p. incl. tables (Rept. no. 9183-60) Unclassified report

Notice: Only Military Offices may request from ASTIA. Others request approval of Chief, Bureau of Ships, Navy Dept., Wash. 25, D. C.

Descriptors: Rubber seals*; Vulcanizates; Effectiveness; Tests; Hydraulic seals*; Elastomers; Hydraulic fluids; Phosphates; Esters.

The Problem: Can fluoro-rubber vulcanizates based on Viton B be used as seal materials for both triaryl phosphate ester (TPE) and petroleum base hydraulic fluids at test temperatures of 212F and 300F? How do the last three Viton B's received (up to November, 1960) compare with the best one of those tested in the previous work?
Findings: All the Viton B formulations tested in ASTM No. 3 oil were very little affected by it, so that the Code 12 Material (the best from the previous tests) is considered suitable as a seal material for either fluid at test temperature of 300F. The later three Viton B's, Codes (over)

14, 17 and 18, are considered unsuitable at 300F because of excessive compression set. For all these materials, compression set at 300F increased with the time compressed, which has not been so with some other types of polymers tested. (Author)

Open-ended Terms: Viton B.

AD-247 750L

AD-248 037L   Div. 26, 30
(28 Dec 60)

Douglas Aircraft Co., Inc., Santa Monica, Calif.
NON-DESTRUCTIVE LEAK TESTING OF SEALED PACKAGES.
Monthly progress rept. no. 6 for Sep 60. 15 Oct 60 [6]p. incl. illus.
(Contract NOa(s) 60-6037-c)   Unclassified report

Notice. Only Government Agencies may request from ASTIA. Others request approval of Chief, Bureau of Naval Weapons, Navy Dept., Wash. 25, D. C.

Descriptors: Packaging*; Materials, Paper; Plastics; Plastic coatings; Seals, Vacuum systems, Tests; Test methods; Halogen compounds, Gas leaks; Diffusion; Containers*; Non-destructive testing.

Because of the relatively high diffusion rate of gases through some plastic and plastic-coated paper barrier materials, when compared with metal foil and foil laminates, work is being done to determine these rates quantitatively and also to establish for comparison the effusion rates of the same gases through small, measured pinholes in foil. A test chamber has been fabricated to study techniques of leak detection, using heat lamps to force effusion of a halogenated (over)

gas through small measured pinholes in Kraft-foil (MIL-B-131) envelopes, and positioning the detector probe at various distances and locations relative to the package. Heat lamp positioning and length of exposure of the package to this heat source are controlled to limit the temperature rise in the package to approximately 50 F. A large vacuum reservoir tank has been put in-line with a small vacuum chamber (Bell-jar). This will permit studies of the effect of vacuum rate changes on "pillowing" of sealed packages. Detection devices for measuring vacuum, rate of change of vacuum, and package pillowing are being refined. With the refinement of detection devices it may be possible to detect package expansion at lower vacuums than heretofore. (Author)

AD-248 037L

AD-248 535   Div. 14, 8   U
(6 Jan 61)

Sperry Gyroscope Co., Great Neck, N. Y.
METAL-TO-CERAMIC SEAL TECHNOLOGY STUDY,
by S. S. Cole, Jr., H. W. Larisch and others.
Final technical rept. 22 June 59-22 Sep 60.
Oct 60, 105p. incl. illus. tables, 29 refs.
(Rept. no. NA-8240-8216)
(Contract AF 30(602)2047, Proj. 4506)
(RADC TN 60-236)   Unclassified report

DESCRIPTORS: Aluminum compounds, Analysis, *Ceramic materials, *Electron tubes, *Seals, Tensile properties, Mechanical properties, Thermodynamics, *Metal seals, Sintering, Stresses, Metals, Oxides, Brazing, Tests, Glass, Bonding.

A literature survey on ceramic-to-metal sealing techniques, adherence theory, and allied systems disclosed limited published work and no procedures for achieving ultra-high-strength seals or seals to pure high alumina. Two additional theories were formulated for this study—one proposing the migration of the glass in the ceramic (over)

properties. Parkers No. 2 silicone O-rings are not recommended for Naval ordnance applications. Linear 28-70 silicone O-rings are suitable for static seals. This latter material is resilient at temperatures as low as -73 C (-100 F). Silicone O-rings are not recommended for dynamic seals because of their low abrasion and tear resistances. (Author)

AD-248 647   N

AD-248 647   Div. 14   U
(10 Jan 61)

Naval Ordnance Lab., White Oak, Md.
O-RING MATERIALS FOR NAVAL ORDNANCE APPLICATIONS,
by Doris V. Steele and A. R. Timmins. 1 Aug 60.
13p. incl. illus. tables.
(NAVWEPS rept. no. 7284)   Unclassified report

DESCRIPTORS: *Rubber seals, Temperature, *Silicone resins, Military requirements, Elastomers, Hydraulic seals, Silicones, *Seals, Silicon compounds, Tests.

A study was made of O-ring materials conforming to Military Specification MIL-P-5516A to determine their suitability for pneumatic applications in Naval ordnance. Two types of silicone elastomers were also included in the study. Results indicate that MIL-P-5516A O-rings may be suitable for many applications but that low temperature stiffening at -54 C (-65 F) could permit leakage or breathing of ordnance under adverse conditions. It is recommended that MIL-P-5516A O-rings be lubricated with DC-55 silicone pneumatic grease (MIL-L-4343A) to obtain optimum aging (over)

into the metallizing mixture, and the other recognizing the need for promoting metallized sintering. Three sintering temperatures were chosen, depending on composition, for each of the 200 metallizing mixtures. Each mixture was applied to specimens of 94-, 96-, and 99.6-percent alumina. Testing involved a screening technique whereby the most promising compositions were carried through to increasingly refined test techniques (scratch and peel, circumferential seal, and finally tensile tests). The tensile test specimen was redesigned to eliminate shoulder breaks when evaluating ultra-high-strength seals. Extremely strong seals were developed for all the ceramic bodies considered. A wide variety of sealing compositions was disclosed which produced seals stronger than those previously reported. A method to calculate stresses in ceramic-to-metal seals is theorized. Measurements of the properties of the metal and of residual stresses in seals were made, showing excellent agreement with calculated stresses. (Author)

AD-248 535   N

AD-249 144L   Div. 14   U
(17 Jan 61)

Douglas Aircraft Co., Inc., Santa Monica, Calif.
NON-DESTRUCTIVE LEAK TESTING OF SEALED PACKAGES.
Monthly progress rept. no. 8 for Nov 60.
15 Dec 60, 8p. incl. table.
(Contract NOa(s) 60-6037-c)
    Unclassified report

Notice: Only Government Agencies may request from ASTIA. Others request approval of Chief, Bureau of Naval Weapons, Navy Dept., Wash. 25, D. C.

DESCRIPTORS: *Packaging, *Non-destructive testing, Seals, Effectiveness, Test methods, Test equipment, Gas diffusion, Tests, Halogen compounds, Materials, Gas leaks.

Various packaging materials were compared to their permeability to Freon 12. Materials which showed no diffusion on the basis of a 2-min diffusion time were tested for longer periods of time. Tests were made on p-dichlorobenzene, one of the most volatile solid halogenated hydro- (over)

carbons, in the vapor fractometer to see whether it could be detected; negative results were obtained. Six-to seven-mil polyethylene was found to be more permeable to Freon 12 than was approximately 1/2-mil polyethylene layer in Poly-Kraft. Kraft paper was about 136 times as permeable as Poly-Kraft. Further investigation of the halogen method of leak detection indicated that the synthetic Freon compounds provide the highest degree of detectability, and the highest degree of reliability and reproducibility. Tentative conclusions were: (1) that in a package containing 100% Freon-22 it is possible to detect a hole as small as 0.000001 inch, (2) it is possible by using Freon-114, to distinguish between a leak in, and diffusion through a Poly-Kraft package; and (3) on a statistical basis, the size of the smallest hole detectable can be varied by changing the amount of heat absorbed by the package, the molecular weight of the Freon used, and the Freon concentration, based on the chamber volume.

AD-249 144L

AD-249 262L   Div. 14   U
(17 Jan 61)

Nonmetallic Materials Lab., Wright Air Development Div., Wright-Patterson Air Force Base, Ohio.
HIGH TEMPERATURE FUEL TANK SEALANTS,
by Philip A. House. Rept. on Nonmetallic and Composite Materials Sep 60. 15p. incl. tables.
(Proj. 7340)
(WADD TN 60-154)
    Unclassified report

No automatic release to Foreign Nationals.

DESCRIPTORS: *Seals, *Sealing compounds, *Fuel tanks, Fluorides, Elastomers, *Production, Materials, Temperature, High temperature research.

The Air Force has investigated the area of integral fuel tank sealants for advance design air weapons and has obtained materials which significantly advance the state-of-the-art. Fuel tank sealants are now available that will withstand considerable periods of aging in fuel and (over)

air at temperatures up to 600 F. A room temperature cure for fluoroelastomers has been discovered and efforts are now being directed toward suitable sealants using this curing system. (Author)

AD-249 262L

AD-249 349   Div. 7   U
(19 Jan 61)

Gelton Industries, Inc., Metuchen, N. J.
THE TESTING OF HERMETIC SEALS AND SEPARATORS, POSITIVE PLATE STUDIES, AND INVESTIGATIONS ON THE UNIFORMITY OF SEALED NICKEL-CADMIUM CELLS,
by H. N. Seiger. Quarterly progress rept. no. 1, 1 July-30 Sep 60 on Investigations Leading to the Development of Improved Sealed Nickel-Cadmium Batteries. 27p. incl. illus.
(Contract DA 36-039-sc-85390, Proj. 3G1P-03-001)
    Unclassified report

For reference at ASTIA Hq. only. This report cannot be satisfactorily reproduced; ASTIA does not furnish copies.

DESCRIPTORS: *Alkaline cells, Seals, Battery separators, Electrodes, Ceramic materials, Electrolytes, Absorption, Reliability.

Investigations are to be conducted which will lead to improved hermetically sealed nickel-cadmium batteries having a high level of reliability (over)

and performance. Efforts will be directed toward: (1) development of a ceramic-to-metal seal for 5 ampere-hour cells; (2) development of a good separator; and (3) development of greater uniformity between cells, and test methods for rapidly determining uniformity; (4) a fundamental study of the positive electrode directed at increasing its stability and reducing self-discharge; and (5) fabrication of improved 5 ampere-hour hermetically sealed cells. Work has been started on the hermetic seal, separators, and uniformity of cells. Twenty separators were wetted with electrolyte and subjected to an acceleration of 25 G for 2 minutes. The results of this test are tabulated in the report. Some separators can absorb up to 10 times their own weight of electrolyte, and retain a goodly portion after acceleration. (Author)

AD-249 349   N6

AD-249 387L Div. 14 U
(19 Jan 61)

Goodrich, B. F., Co., Brecksville, Ohio.
NUCLEAR RADIATION RESISTANT POLYMERS AND POLYMERIC COMPOUNDS,
by John N. Born. Rept. for 1 Apr 59-31 Mar 60, on ANP Airframe and General Support. Oct 60, 101p. incl. illus. tables, 56 refs.
(Contract AF 33(616)6442, Proj. 1448; Continuation of Contract AF 33(616)5646)
(WADC TR 55-58, Pt. 6) Unclassified report

No automatic release to Foreign Nationals.

DESCRIPTORS: Polymers, *Elastomers, Radiation damage, *Synthetic rubber, Temperature, *Heat resistant polymers, Sensitivity, Synthesis, Gamma rays, Isomeric transistors, *Rubber seals, Additives, Butadienes, Acrylic resins, Mechanical properties, Molecular structure.

This research includes the following: fundamental studies of radiation energy transfer and of the (over)

mechanisms of radiation damage in high polymers; the selection, design, and synthesis of special new monomers to produce new high polymers with outstanding inherent radiation resistance and heat stability; polymerization of said monomers, identification of the resulting polymers, and evaluation of their radiation and heat stabilities; applied studies of the effect of gamma irradiation in air at room temperature on the compression set properties of various rubber compounds, with and without potential antirads present; and static and semi-dynamic radiation testing of O rings with attempts to protect the military-approved rubber compounds against radiation damage with antirads. (Author)

AD-249 387L

AD-249 542 Div. 10 U
(24 Jan 61)

Bell Aerosystems Co., Buffalo, N. Y.
STORABLE PROPELLANT DATA FOR THE TITAN II PROGRAM,
by Ralph R. Liberto. Quarterly progress rept. no. 1, 15 June-15 Sep 60. Oct 60, 70p. incl. illus. tables, 51 refs.
(Contract AF 04(611)6079, Proj. 6753)
(AFFTC TR 60-62) Unclassified report

DESCRIPTORS: *Methylhydrazines, Nitrogen compounds, *Tetroxides, Handling, Storage, *Liquid rocket propellants, *Rocket fuels, *Rocket oxidizers, Safety devices, Hazards, Physical properties, Containers, Metals, Plastics, Elastomers, Lubricants, Sealing compounds, Gaskets.

Open-ended Terms: Titan II.

Physical properties, materials compatibility, handling techniques, flammability and auto-ignition limits, and procedures for storing, (over)

cleaning, and flushing of 50/50 UDMH-N2H4 blend and N2O4 are summarized in this report. The source material of this information was obtained during a one-month literature and industry survey. During this time users of these propellants and companies participating in the Titan II program were contacted. The known materials compatibility data for N2O4 was found to be reasonably complete. The known materials compatibility data for the 50/50 UDMH-N2H4 blend is limited and is presented with a summary of metals compatible with N2H4. Data regarding the handling of N2O4 is based on user experience, but limited information is available for the 50/50 UDMH-N2H4 blend. Information presented on the handling of the fuel blend is based on experience in dealing with UDMH and N2H4 individually. The appendix is a U. S. Bureau of Mines report covering work done under a working agreement with Bell Aerosystems to determine the flammability and auto-ignition limits of the propellants. (Author)

AD-249 542 N6

AD-250 028 Div. 14 U
(1 Feb 61)

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
EFFECT OF SHELF AGING ON MIL-P-5516 O-RINGS,
by J. N. Holloway. Progress rept. no. 2. 26 Jan 61, 4p. incl. table (Proj. no. S-F013-01; Rept. no. 92-7)
Unclassified report

DESCRIPTORS: *Rubber seals, *Gaskets, Aging, Mechanical properties, Physical properties, Military requirements, Synthetic rubber, Light, Oxygen, Tests, Storage.

O-rings originally complying with Military Specification MIL-P-5516 and which had reached the maximum allowed age of 48 months, were tested after an additional 24 months of shelf aging. During the original 48 months the rings were kept in sealed envelopes, but during the subsequent 24 months, some were kept in sealed envelopes, some exposed to air in the dark, and some exposed to air and artificial light. No significant (over)

changes in tensile properties, permanent set, or specific gravity due to the additional 24 months shelf aging were observed. It is concluded that O-rings meeting the requirements of Specification MIL-P-5516 will give satisfactory service after at least 6 years shelf aging at moderate room temperatures. (Author)

AD-250 028 N1

AD-250 163L Div. 31 U
(7 Feb 61)

Coast Guard, Washington, D. C.
STRENGTH TEST OF VINYL PAD INSERTS.
6 Jan 61, 8p. incl. illus. table. (Proj. CGTD 328-3/3-5-6; Rept. no. 219)
Unclassified report

Notice: Only Military Offices may request from ASTIA. Others request approval of U. S. Coast Guard, Wash. 25, D. C.

DESCRIPTORS: *Life preservers, Sealing compounds, Seals, Bonded joints, Mechanical properties, Adhesion, Test methods, Tests, Effectiveness, Vinyl radicals, Polymers, Plastics.

Coast Guard Specification for Kapok Buoyant Vests (160.047) calls for a test on the forming and sealing of the vinyl pad inserts. The requirement states that the heat-sealed pad seams shall show an adhesion of not less than 8 pounds when one-inch strips cut across and perpendicular to the seams are pulled apart at a rate of separa- (over)

tion of the clamping jaws of the test machine of 12 inches per minute. This report covers the above strength tests on four samples of vinyl pad inserts manufactured by the Atlas Manufacturing Company. (Author)

AD-250 163L

AD-250 231 Div. 22, 33 U
(6 Feb 61)

Ogden Air Materiel Area, Hill Air Force Base, Utah.
T-46 AMMUNITION BOX,
by Lipman H. Singer. Airmunitions test rept. Jan 61, 30p. incl. illus. tables (Rept. no. OOT TM 61-2)
Unclassified report

DESCRIPTORS: Ammunition, *Containers, Air transportation, Airborne, Tests, Seals.

Open-ended Terms: T-46 ammunition box.

Altitude chamber tests were performed on T-46 Ammunition Boxes for the purpose of determining if this container was suitable for airborne ammunition transportation. Six instrumented box assemblies were tested in an altitude chamber to study the following parameters: reduced pressure at constant temperature and reduced pressure at reduced temperature. It was found that combined low temperature and pressures (e.g. air trans- (over)

portation) affect the plasticity and sealing ability of the rubber gasket, therefore, the box will leak resulting in a vacuum developed in the container. Upon return to sea level conditions, the vacuum will be retained in the box and may render it difficult to open. In most cases this vacuum may be broken on release of the cover hasp. After a short warming up period, the box will again leak and equilibrium pressure will be attained. No modification to the T-46 Ammunition Box is required, however, it is recommended that the ammunition be protected from air leakage by: utilization of heated air cargo space to improve the sealing properties of the rubber; insertion of desiccant with the contents, to absorb water vapor (e.g. top and bottom cardboard spacers impregnated with desiccant). If a new box is developed, it is recommended that a pressure relief device be installed and altitude chamber tests be made to insure that the box will be suitable for air lift. (Author)

AD-250 231 N1

AD-250 399 Div. 8 U
(8 Feb 61)

McCoy Electronics Co., Mount Holly Springs, Pa.
PRODUCTION ENGINEERING MEASURE. MINIATURE GLASS HOLDER CRYSTAL UNITS CR-(XM-31)/U.
Combined progress rept. 1 June-30 Nov 60. 14p. incl. illus.
(Contract DA 36-039-sc-85939; Continuation of Contract DA 36-039-sc-74942)
Unclassified report

DESCRIPTORS: *Crystal holders, *Miniature electronic equipment, Quartz crystals, Production, Manufacturing methods, Seals, Crystals.

Development of efficient production methods for the crystal unit type CR-(XM-31)/U along with equipment and techniques for sealing 120 or more per eight hours is reported. Fabrication of the preproduction units was nearly completed; both material and cultured quartz plates were processed in identical fashion. No significant differences between natural and cultured quartz (over)

plates were found; this opinion is based upon a study of 30 units at 40 megacycles. The units appeared to meet specification requirements with a margin of safety; frequency calibration after sealing measured, on the average, between 0.001 and 0.002% above pre-seal readings; at 40 megacycles the resistance decreased approximately 5 or 6 ohms, going from atmosphere to a vacuum of 20 to 40 microns. The sealing equipment met the production requirements.

AD-250 399 N1

AD-250 736 Div. 22 U
(15 Feb 61)

Atlas Powder Co., Tamaqua, Pa.
DEVELOPMENT OF MINIATURE, HERMETICALLY-SEALED SQUIB SWITCH,
by G. W. Bills. Narrative summary rept. (Final) Aug 60, 111p. incl. illus.
(Contract DA 36-034-501-ORD-86, Proj. TN 2-8107)
Unclassified report

DESCRIPTORS: *Electric detonators, Design, *Electric switches, Materials, Switches, Miniature electrical equipment, Seals.

The report covers the progress of the design and development of hermetically-sealed miniature squib switches. This includes two types of switches, a 4-pole and an 8-pole switch. The problems encountered in the general design during development, assembly and testing are discussed. The test results included indicate the 4-pole and 8-pole miniature squib switch designs meet military requirements. (Author)

AD-250 742 Div. 30, 14 U
(17 Feb 61)

Battelle Memorial Inst., Columbus, Ohio.
FUEL O-RING TESTING PROCEDURES,
by W. A. Hartig, R. A. Clark, and R. G. Heiligmann. Final rept. 31 Mar 54, 1v. incl. illus. table.
(Contract AF 33(038)25618)
Unclassified report

DESCRIPTORS: Seals, Test equipment, Test methods, *Fuel seals, *Gaskets, Military requirements, Pressure, Tests, Specifications, Rubber gaskets, Temperature, Rubber seals.

Studies were made of the rates of swelling and drying of fuel-type O-rings at 250 degrees F and the effects of pretreatment on the leakage of such O-rings at -65 F when subjected to fluid pressures from 15 to 1500 psi. In the swelling of fuel O-rings at 250 F, about the same maximum and equilibrium (constant) swell values were obtained as at -65 to 212 F, except that the rate of swelling increased with temperature. The same trend was found for drying O-rings at 250 F. Preconditioning O-rings at 250 F, by both swelling and drying to equilibrium conditions, was more harmful to rubber physical properties than at 160 to 212 F. Two qualification test procedures for fuel O-rings are detailed. The one procedure utilizes preconditioning at 250 F and tests low-temperature leakage in a jig meeting Spec MIL-P-5315, except that the gland in the rotating jig stem is eccentric by 0.0005 inch. The other procedure utilizes preconditioning at 250 F and involves eccentric jig-stem glands which were made shallower to provide more O-ring squeeze and fluid pressures of 15 to 1500 psi.

AD-250 742 M1

AD-250 749 Div. 7 U
(16 Feb 61)

Bell Telephone Labs., Inc., Whippany, N. J.
RESEARCH AND DEVELOPMENT WITH HERMETICALLY SEALED CONTACTS,
by O. M. Hovgaard. Final rept. 15 Feb 57-14 Aug 60. 14 Oct 60, 57p. incl. illus. tables.
(Contract DA 36-039-sc-72835, Proj. 3-26-01-000)
Unclassified report

DESCRIPTORS: Design, *Electric relays, Miniature electronic equipment, Seals, Life expectancy, Switching circuits, *Electric switches, Switches, Miniature electrical equipment.

Consideration was given to the selection from several experimental designs of dry-reed switches of the one most suitable to use as the basis for this study. This proved to be the Bell Laboratories G-29 design and its life performance at 28 volts dc was explored in detail. It also served as the means for studying a large variety of possible miniature relays. Models examined contained from 1 to 4 switches within a single operating coil and the problems of providing make, break and transfer contact combinations were explored. Magnetic latching types of relays providing make contacts were studied as was their application to switching arrays for frequencies up to 10 megacycles. This contract has provided fundamental information on the characteristics and life expectancy of G-29 switches and indicated many of the problems which must be considered in the design of relays and switching structures. (Author)

AD-250 749 M1

AD-251 130L Div. 14 U
(23 Feb 61)

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
EVALUATION OF EPOXY AND THIOKOL VULCANIZATES FOR GASKETS OF SOLVENT CONTAINERS.
14 Feb 61, 4p. tables. (Rept. no. 119-7)
Unclassified report

Notice: Only Government Agencies may request from ASTIA. Others request approval of Bureau of Ships, Navy Dept., Wash. 25, D. C.

DESCRIPTORS: *Plastic gaskets, Elastomers, Vulcanizates, Epoxy resins, Containers, Hydrocarbons, Ketones, Petroleum, Esters.

The purpose of this investigation was to find an elastomeric material for container gaskets which would be suitable for use with all of the following types of fluids: aromatic compounds, ketones, mixtures of aromatic compounds with esters and ketones, and petroleum fuels and oils. Three epoxy elastomeric compounds and three vulcanizates of Thiokol FA and Thiokol ZR-300 were evaluated for this purpose. It was tentatively concluded on the basis of the results of swelling and compression set tests that none of these materials would be suitable. (Author)

AD-251 130L

AD-251 145L Div. 11 U
(24 Feb 61)

Aberdeen Proving Ground, Md.
AUXILIARY ENGINE-GENERATOR KIT AND CARGO HATCH DOOR SEALS FOR M59 ARMORED PERSONNEL CARRIER,
by L. F. Garner. Feb 61, 1v. incl. illus. tables (Rept. no. DPS/OTA-101)
Unclassified report

Notice: Only Military Offices may request from ASTIA. Others request approval of Diamond Ordnance Fuze Labs., Wash. 25, D. C. Attn: ORDTL 06.33

DESCRIPTORS: *Internal combustion engines, *Cargo vehicles, *Armored vehicles, *Seals, *Motor generators, *Power supplies, *Hatches, Electrical equipment, Voltage regulators, Maintenance, Reliability, Effectiveness, Installation, Operation, Direct current.

Open-ended Terms: M59 personnel carriers.

A 100-ampere auxiliary engine-generator kit was tested to determine the ease of installation, adequacy of installation instructions, ease of maintenance, mechanical reliability and operating characteristics. The engine failed to cool when the kit was installed in the M59 vehicle. A similar portable unit was tested outside the vehicle and delivered approximately 30% of the rated output. It is concluded that the command post vehicle power requirement exceeds 100 amperes. It is recommended that a portable unit of 150-ampere capacity be developed. An improved cargo hatch door seal tested to determine sealing ability was found to be satisfactory. (Author)

AD-251 145L

AD-251 453 Div. 30 U
(1 Mar 61)

Naval Research Lab., Washington, D. C.
THE NRL MODEL E-5753 LEAK DETECTOR,
by C. C. Minter, S. F. Andersen, and R. W. Dolk. 18 Jan 61, 7p. incl. illus. (NRL rept. no. 5593)
Unclassified report

DESCRIPTORS: *Gas leaks, Gas detectors, Thermoelectricity, Circuits, Design, Heat transfer, Sensitivity, Detection, *Gases.

Open-ended Terms: Freon-12.

A 21-pound portable Freon leak detector has been developed. Air from the vicinity of a probe is pumped through tubing so as to first pass two of the four cells of a thermal conductivity bridge and then through a length of tubing around to the other pair of cells. Thus if the probe is moved past a leak, the bit of Freon-containing air unbalances the bridge first in one direction and then in the other. These unbalances cause the successive flashing of a red light and a white light in the lucite probe head, where the operator's attention is focused. While the indicator gives a clear response for a Freon concentration of 4 to 5 ppm, such sensitivity is not needed under service conditions, where detection of the signal produced by changing the concentration of Freon by 50 ppm is satisfactory in most cases. (Author)

AD-251 453 M2

AD-251 669 Div. 31, 14 U
(? Mar 61)

Material Labs., Puget Sound Naval Shipyard, Bremerton, Wash.
LATEX MASTIC INJECTED AS SEALANT UNDER FLIGHT DECKING. 1st TRIAL INSTALLATION,
by W. H. Miles and D. S. Petersen. Jan 61, 7p. incl. illus. (Development rept. no. E-179-3)
Unclassified report

DESCRIPTORS: *Flight decks, *Sealing compounds, Seals, Rubber seals, Weather proofing, Deck coverings, Coatings, Effectiveness, Tests.

Open-ended Terms: Yorktown, CVS-10.

Methods were investigated to stop leaks through flight decks, without plank removal, by use of a sealant injected between wood deck and underlying steel pan. A survey of possible sealing materials including polysulfides, epoxies, epoxy-tar, latex mastic, and bitumen compounds indicated neoprene latex-microballoon-hydraulic cement mastic of BUSHIPS NOTICE 9140 to be the most suitable from standpoint of sealing ability, weight, cost, and handling characteristics under adverse weather and temperature conditions. In a trial installation on the flight deck of the USS YORKTOWN (CVS-10), all leaks over a 6,000 sq. ft. area were stopped by pneumatic injection of latex mastic through the wood deck. The latex mastic injection system represents a low cost repair for leaks in wood flight decks. Due to the presence of an interfering coating of rust preventive grease between planks and steel pan, continuous bonding to both surfaces is not obtained. Subsequent movement of the deck may open new water channels requiring repetition of the injection in some areas. There is evidence that complete filling of void areas may not be essential due to the ability of the latex portion itself to penetrate cracks and seal leaks. (Author)

AD-251 669 N2

AD-252 077 Div. 25, 8 U
(10 Mar 61)

Philco Corp., Philadelphia, Pa.
ELECTROQUENCHABLE PHOSPHOR INVESTIGATION.
Interim technical rept. no. 3, 1 Sep-30 Nov 60, 31 Dec 60, 13p. incl. illus. (Rept. no. 9005-3)
(Contract NObsr-81189)
Unclassified report

DESCRIPTORS: *Display systems, *Phosphors, *Fluorescence, Luminescence, Sulfides, Ultraviolet radiation, *Electric fields, Cadmium compounds, Vacuum systems, Films, Electrical properties, Silicones, Seals.

A vacuum system was revised to permit electrical contact to be made to a cell within the bell jar of the vacuum system both during and after evaporation of the final contact. Tests were performed to determine if water vapor or oxygen is responsible for changes in cell characteristics. The results indicate the need for sealing completed cells against atmospheric effects. Silicone varnishes were used to coat completed cells but had a deleterious effect on quenching and in some cases, have disrupted top contact conductivity. Excellent extinction characteristics have been obtained by using films of sodium fluoride and magnesium fluoride. Cells with a final copper oxide contact show far less moisture sensitivity than those with final indium-tin oxide contacts. (Author)

AD-252 077 N3

AD-252 607 Div. 8, 14, 20 U
(20 Mar 61)

Radiation Effects Information Center, Columbus, Ohio.
THE EFFECT OF NUCLEAR RADIATION ON ELECTRON TUBES AND TUBE MATERIALS,
by W. E. Chapin. 15 Feb 61, 36p. incl. illus. tables, 79 refs. (REIC rept. no. 14)
(Contract AF 33(616)7375), Proj. 4458)
Unclassified report

DESCRIPTORS: *Radiation damage, Gamma rays, *Electron tubes, *Miniature electron tubes, *Subminiature electron tubes, Thyratrons, Photomultipliers, Voltage regulator tubes, Triodes, Beam power tubes, Pentodes, Magnetrons, Traveling wave tubes, Photometers, Iconoscopes, Klystrons, Cathode ray tubes, Glass, Ceramic materials, Metals, *Glass seals, *Metal seals, Bibliography.

The report concerns studies of radiation effects as reported for electron tubes and tube materials during the past 8-10 years. These studies indicate that the primary damage mechanism which limits prolonged use or storage of electron tubes in high-intensity fluxes is the failure of the glass envelope. Much has been written concerning failures in the glass-to-metal seals. These damages were observed at integrated neutron fluxes in the range of 10 to the 15th power to 10 to the 18th power (nv)t, for most borosilicate envelopes and Kovar seals. Boron-free glasses are more radiation resistant and can withstand integrated fluxes as high as 10 to the 19th power (nv)t. Tubes necessitating large glass envelopes are more prone to radiation damage. Miniature and sub-miniature tubes sealing areas and use less glass, thus enabling the tube to withstand prolonged exposures. Ceramic tubes exhibit one disadvantage; the level of residual radioactivity after irradiation is much higher than for the glass-tube type. Gamma radiation does not cause any appreciable effect on most electron tubes. The exception to the case is mainly tubes that rely on light transmission. Magnetrons and similar devices are not suited for use in high-intensity radiation fields. (Author)

AD-252 607 N3

AD-252 784 Div. 26, 8 U
(20 Mar 61)

Bulova Research and Development Labs., Inc., Woodside, N. Y.
OPERATION AND MAINTENANCE MANUAL. SEALING MACHINE,
1961, 1v. incl. illus.
(Contract DA 36-039-sc-54667)
Unclassified report

DESCRIPTORS: *Quartz crystals, Instruction manuals, Operation, Maintenance, *Manufacturing methods, Processing, Seals, Machines, Automatic, Crystal holders.

A base sealing machine a component of the quartz crystal manufacture mechanization system, is described. It was designed to seal the assembled crystal bases and covers automatically. During this process all air is evacuated and the inert gas nitrogen, with a trace of helium added, is introduced into the cover prior to sealing. (Author)

AD-252 992L Div. 14, 22 U
(24 Mar 61)

Ammunition Group, Picatinny Arsenal, Dover, N. J.
INVESTIGATION OF THREADED JOINT SEALANTS AND CRIMPED JOINT SEALANTS,
by W. Gaston. Final engineering rept. Dec 60, 1v. incl. illus. tables, 20 refs.
(Technical rept. no. DC-TR:2-60)
Unclassified report

Notice: Only Military Offices may request from ASTIA. Others request approval of Diamond Ordnance Fuze Labs., Wash. 25, D. C. Attn: ORDTL 06.33

DESCRIPTORS: *Seals, *Sealing compounds, Rubber seals, *Ammunition, Production, Joints, Manufacturing methods, Oxygen, Design, Metals, Torque, Rubber, Polymers, Sulfides, Storage, Explosives.

None of the anaerobic thread sealants submitted were suitable for general use in ammunition items. At present, the formulation of an anaerobic thread sealant that is generally applicable to ammunition items is impracticable. The commercial grades of one thread sealant are suitable for some ammunition applications. However, for each application considered, it will be necessary to make a separate determination of compatibility with the explosive used, method of application to obtain satisfactory penetration, and acceptable setting time and permissible torque range with the metals involved. (Author)

AD-252 992L

AD-252 996 Div. 26, 14 U
(24 Mar 61)

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
SUITABILITY OF VITON B O-RINGS FOR USE IN 3000 psi HYDRAULIC SYSTEMS CONTAINING PETROLEUM BASE OR PHOSPHATE ESTER FLUIDS,
by A. E. Barrett. Progress rept. no. 1, 10 Mar 61, 7p. illus. tables. (Rept. no. 92-4)
(Proj. S-F013-13-01)
Unclassified report

DESCRIPTORS: *Hydraulic seals, Materials, Rubber, Tests, *Hydraulic fluids, Phosphates, Esters, Petroleum, Hardness, *Hydraulic systems, Polymers, Ethylenes, Fluorides, Synthetic rubber, Cyclohexanes.

Open-ended Terms: Viton.

The Bureau of Ships is interested in determining whether or not seals made from Viton B will be suitable for use in 3000 psi hydraulic systems containing either petroleum base or phosphate ester fluids. Accordingly, a simulated dynamic service test was devised and Viton B O-rings of 70 and 90 Shore-A hardness were tested therein in comparison with O-rings conforming with Specification MIL-P-5516B, Class A (90 Shore-A) and Class B (70 Shore-A). The results obtained to date indicate that Viton B O-rings of either 70 or 90 Shore-A hardness with Teflon backup rings will perform satisfactorily at 190 F in 3000 psi hydraulic systems containing phosphate ester fluids. The Teflon backup rings do not stand up at higher temperatures under conditions of this test. (Author)

AD-252 996 N4

AD-253 211   Div. 8   U
(20 Mar 61)

Westinghouse Electric Corp., Elmira, N. Y.
SCAN-CONVERSION STORAGE TUBE BASED UPON THE PERMACHON,
by W. S. Rial. Quarterly progress rept. no. 2, 1 Oct-31 Dec 60. 31 Dec 60, 21p. incl. illus.
(Contract DA 36-039-sc-85051)
Unclassified report

DESCRIPTORS: Photoconductivity, *Storage tubes, Cathode ray tubes, Coatings, Test equipment, Electron guns, *Image tubes, Design, Electron tubes, Tests *Image converter tubes, Seals.

Open-ended Terms: Permachon.

Two experimental tubes of the second design version were started, both of which were completed through tip-off. RF sealing techniques for envelope assemblies were established. Three experimental CRT's were constructed to study writing gun characteristics. The test equipment is capable of 800 TV line resolution overall using magnetically deflected write and read guns. (Author)

AD-253 211   N4

AD-253 527   Div. 30   U
(3 Apr 61)

Louisiana State U., Baton Rouge.
HIGH-PRESSURE AND VACUUM SEALS AT LIQUID HELIUM TEMPERATURES,
by K. S. Balain and C. J. Bergeron. Aug 59. 1p. incl. table.
(OOR rept. no. 1579:5)   Unclassified report

Reprint from the Review of Scientific Instruments 30:1058-1059, Nov 59. (Copies not supplied by ASTIA)

DESCRIPTORS: Epoxy resins, *Plastic seals, *Vacuum seals, Design, High pressure research, *High temperature research, *Liquefied gases, Helium, Temperature.

AD-253 753   Div. 14, 26   U
(13 Apr 61)

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
DEVELOPMENT OF RUBBER GASKETS WHICH ARE RESISTANT TO NUCLEAR RADIATION,
by R. E. Morris. Final rept. 28 Feb 61, 1v. incl. illus. tables. (Rept. no. 149-4)
(Proj. NS-033-200)
Unclassified report

DESCRIPTORS: Gaskets, *Rubber gaskets, Seals, Radiation damage, Radiation tolerance, *Synthetic rubber, *Rubber, Additives, *Butyl rubber, *Vulcanizates, *Rubber seals, Mechanical properties, Tests, Test methods, Gamma rays, Packaging.

An investigation was conducted to develop rubber gaskets and seals capable of operating for extended periods in nuclear radiation fields. The tendency of a compressed vulcanizate to undergo compression set when exposed to gamma radiation was chosen as a criterion. Vulcanizates have been developed which are capable of satisfactory gasket service in gamma radiation fields to a dosage of 10 to the 8th power roentgens and beyond. The rubbers which are the most satisfactory for manufacture of gaskets to be used in gamma radiation fields are natural rubber, styrene/butadiene copolymers and acrylonitrile/butadiene/methacrylic acid terpolymers. The resistance of these vulcanized rubbers to compression set engendered by gamma radiation can be enhanced by compounding the rubbers with certain antioxidants or antiozonants and/or with certain chemicals containing aromatic rings or condensed ring structures. The resistance of a styrene/butadiene vulcanizate to gamma radiation is better if dicumyl peroxide is used as the vulcanizing agent in lieu of [illegible] (Author)

AD-253 753   N5

AD-254 020   Div. 7   U
(13 Apr 61)

Gulton Industries, Inc., Metuchen, N. J.
THE TESTING OF HERMETIC SEALS AND SEPARATORS, POSITIVE PLATE STUDIES, AND INVESTIGATIONS ON THE UNIFORMITY OF SEALED NICKEL-CADMIUM CELLS,
by H. N. Seiger. Quarterly progress rept. no. 2, 1 Oct-31 Dec 60, on Investigations Leading to the Development of Improved Sealed Nickel-Cadmium Batteries. 31 Dec 60, 36p. illus.
(Contract DA 36-039-sc-85390, Proj. 3G18-03-001)
Unclassified report

For reference at ASTIA Hq. only. This report cannot be satisfactorily reproduced; ASTIA does not furnish copies.

DESCRIPTORS: *Alkaline cells, Test methods, *Seals, *Battery separators, Electrodes, Ceramic materials, Metal seals, Design, *Anodes (Electrolytic cells), *Storage batteries, Nickel, Cadmium.

The use of the halogen leak detector for testing of the ceramic-to-metal seal was not satisfactory, and a helium leak detector was resorted to after it was deemed satisfactory. Six new separators were included into the testing program. Difficulties encountered with the measurement of electrical resistance were overcome using a voltage drop method. Separators were tested after storage at -40 F and also at 125 F for chemical resistance, dimensional stability and electrical resistance. Separators were also tested in 3-plate cells at 125 F. The voltage stabilizes more rapidly than the pressure during overcharge of sealed cells. During discharge the pressure in a cell decays more rapidly than it would on open circuit. From a cell with an extra positive and extra negative electrode (not electrically connected to the other electrodes within the cell) it was found that the negative electrode could not be used as a reference electrode for the sealed cell. Difficulties encountered when the gas evolution apparatus for the positive plate studies was placed in a thermostatted bath were overcome by modification of the apparatus to eliminate a ball joint and changing two stopcocks from glass to teflon. (Author)
AD-254 020

AD-254 493   Div. 7   U
(21 Apr 61)

Eagle-Picher Research Labs., Joplin, Mo.
RESEARCH INVESTIGATIONS LEADING TO THE DEVELOPMENT AND EVALUATION OF A CADMIUM - SILVER OXIDE BATTERY HAVING A HERMETICALLY SEALED CONSTRUCTION,
by J. K. Wilson. Quarterly progress rept. no. 2, 1 Oct-31 Dec 60. 31 Dec 60, 1v. incl. illus.
(Contract DA 36-039-sc-85370, Proj. 3G18-03-001-02)
Unclassified report

DESCRIPTORS: *Storage batteries, Design, Seals, Cadmium, Cadmium compounds, Silver compounds, Oxides, Electrolytes, *Battery separators, Recombination reactions, Electrodes, Alkaline cells.

Seven hundred shallow cycles have been obtained from cadmium - silver oxide sealed cells. The data obtained indicate that 0.040 ampere is the maximum continuous overcharge rate that can be applied to the cells of the present design. Cell containers and covers were designed for use in the final design sample cells. (Author)

AD-254 496   Div. 14, 22   U
(21 Apr 61)

Frankford Arsenal, Philadelphia, Pa.
STANDARDIZATION OF ELASTOMER FOR BOOT SEAL AND DUST COVER. EVALUATION OF CHLOROPRENE COMPOUNDS,
by S. M. Kalan. Mar 61, 17p. incl. illus. tables.
(Memo. rept. no. M61-19-1)
Unclassified report

DESCRIPTORS: *Chloroprenes, *Elastomers, *Seals, *Sealing compounds, *Rubber seals, Mechanical properties, Temperature, Tests, Fire control, Materials, Protective coverings, Standardization, Test methods, Silicones, Manufacturing methods.

An investigation was made of elastomeric materials used in the manufacture of boot seals and dust covers for fire control material operating at temperatures between -40 degrees F and 125 degrees F. The investigation of chloroprene rubber shows that MIL-STD-417 Type S, Class SC 615 A1B1C2F2S1 material, which is commercially available, will provide the required sealing and stability in air, oil, and ozone over this temperature. It is recommended that this investigation be continued for all configuration designs, since it appears that the modulus at low extensions, low temperature elastic recovery rate and the -40 degrees F Gehman stiffness limiting values require establishment to guarantee satisfactory performance. (Author)

AD-254 496   N6

AD-254 607 Div. 14 U
(24 Apr 61)

Material Lab., New York Naval Shipyard, Brooklyn, N. Y.
INVESTIGATION OF THE EFFECTS OF PHOSPHATE ESTER-PETROLEUM OIL MIXTURES ON THE PHYSICAL PROPERTIES OF NITRILE RUBBER O-RINGS,
by L. G. Moss. Final rept. 20 Apr 61, 8p. table. (Lab. proj. no. 6326)

Unclassified report

DESCRIPTORS: *Phosphates, *Esters, *Petroleum, Mixtures, *Nitrile rubber, *Oils, Hydraulic fluids, *Rubber gaskets, Physical properties, *Hydraulic seals, Submarines, Rubber, Gaskets, Seals, Rubber seals, Synthetic rubber.

The effects of phosphate ester-petroleum oil mixtures on the physical properties of nitrile rubber O-rings, were determined. Mixtures containing 5% and 10% concentrations of Cellulube 220 in each of the oils 2190 TEP and 2110 H, as well as the pure Cellulube 220, 2190 TEP and 2110 H oils, were used as the immersion media. It was found that 5% concentrations of the phosphate ester in each of the oils produced tolerable swelling and softening of the rubber. However, 10% concentrations of the ester in each of the oils, as well as the pure ester itself, produced excessive swelling and softening to a degree which is expected to render the rubber O-rings unserviceable for use in petroleum oil hydraulic systems aboard FBM submarines. It was recommended that phosphate ester impurities in these systems be kept to a minimum concentration of less than 5%. It was also recommended that in view of the observed extraction effects of the petroleum oils, themselves, on the O-rings, that the Laboratory be authorized to develop an improved elastomer O-ring compound. (Author)

AD-254 607 N6

AD-254 614L Div. 27, 31 U
(24 Apr 61)

Naval Boiler and Turbine Lab., Philadelphia, Pa.
LABORATORY PERFORMANCE OF DD-931 CLASS HP-IP TURBINE,
by L. F. VanSciver and D. N. Cooke. Evaluation and development rept. 1960, 1v. incl. illus. tables (NBTL rept. no. T-181)

Unclassified report

Notice: All requests require approval of Bureau of Ships, Navy Dept., Wash. 25, D. C.

DESCRIPTORS: *Destroyers, Marine engines, Propulsion, *Steam turbines, Instrumentation, Diaphragms (Mechanics), Seals, Design, Tests, Operation, Thermal stresses, Deformation, Steam turbine blades, Steam turbine rotors.

The performance and reliability of the DD-931 Class HP-IP turbine were investigated. Tests indicated that the original performance was 11.2 and 12.8% below the expected levels at 20 kn and full power, respectively. With modifications, the turbine developed the original design HP-IP turbine share of the guaranteed full power output at the design flow. However, the 20-kn performance (top of series range) remained about 8% deficient. Nearly half of the original full power deficiency was caused by diaphragm packing rings blowing open radially due to lack of sufficient closing pressure. This was corrected by an improved ring design. The major cause of original and remaining deficiency was unexpected wear of interelement, diaphragm and end gland packing. The irregular wear pattern of the packing indicates local distortion other than that caused by casing hog and sag. The minimum average radial diaphragm and interelement clearance that can be held is 18 to 20 mils. Considerable by-pass leakage between the HP and IP elements due to thermal distortion of the packing rings and/or sealing surfaces was evident. This is responsible for about 1/4 of the remaining deficiency at 20 knots. The reliability and maneuverability of the turbine are good, with no mechanical weaknesses. (Author)

AD-254 614L

AD-254 803L Div. 1, 12 U
(25 Apr 61)

Chance Vought Aircraft, Inc., Dallas, Tex.
MODULAR HYDRAULIC SYSTEM DEVELOPMENT. PROJECT HYDRATOY. PART I,
by R. W. Burden, D. R. Chamberlain and others. Quarterly progress rept. no. 8, pt. 1, 5 Sep-5 Dec 60. 5 Dec 60, 1v. incl. illus. tables. (Rept. no. AER-EOR-13270, pt. 1) (Contract NOa(s) 59-6019-c)

Unclassified report

Notice: Only Government Agencies may request from ASTIA. Others request approval of Bureau of Naval Weapons, Navy Dept., Wash. 25, D. C.

DESCRIPTORS: *Hydraulic systems, *Aircraft, *Satellite vehicles, *Spaceships, Design, *Hydraulic actuators, *Hydraulic valves, Seals, Packaging, Metal seals, Materials, Test methods, Rotary seals, *Guided missiles, Tests.

Open-ended Terms: Hydratoy project.

Progress was continued on modular hydraulic system development. The 2 static metallic seals presently being qualified are the Bendix Wedge Seal and the Ni-Seal (C-Seal). Four hydraulic test packages were designed and fabricated. They are the titanium manifold, the actuator with modular end cap, and 2 stainless steel manifolds. The integrated system design was concerned with review of tube to structure joining methods, evaluation of bending, welding, and brazing applications, study of actuator technology, ball joint actuator design and evaluation, and study of connection methods for permanently attached fluid lines.

AD-254 803L

AD-254 804L Div. 1, 12 U
(25 Apr 61)

Chance Vought Aircraft, Inc., Dallas, Tex.
MODULAR HYDRAULIC SYSTEM DEVELOPMENT. PROJECT HYDRATOY. PART II. MATERIAL AND PROCESS DEVELOPMENT PROGRESS,
by J. W. Browne and E. Jimenez. Quarterly rept. no. 8, pt. 2, 5 June-5 Dec 60. 5 Dec 60, 26p. incl. illus. (Rept. no. AER-EOR-13270, pt. 2) (Contract NOa(s) 59-6019-c)

Unclassified report

Notice: Only Government Agencies may request from ASTIA. Others request approval of Bureau of Naval Weapons, Navy Dept., Wash. 25, D. C.

DESCRIPTORS: *Hydraulic systems, Materials, Packaging, Metal seals, *Hydraulic valves, Lubricants, *Guided missiles, *Spaceships, *Satellite vehicles, *Aircraft, Processing, *Hydraulic actuators, Rotary seals, Tests, Brazing, Coatings, Seals.

Materials and process investigations were continued. Investigations were conducted upon static seals of the Ni-Seal type. The investigation of various lubricants, coatings and surface treatments to reduce friction and increase wear resistance was continued. One coating material, a Teflon, appeared to have promise of reducing galling tendencies during installation. No further investigations were conducted with stainless steel extrusions for use as fluid-carrying structures. Various methods of fabricating fluid-carrying structures are under consideration. The various materials being considered for such construction are magnesium, aluminum, titanium, and stainless steel alloys. (Author)

AD-254 804L

AD-254 806 Div. 14, 30 U
(26 Apr 61)

Naugatuck Chemical Div., United States Rubber Co., Conn.
DEVELOPMENT OF SELF-LUBRICATING AND NON-ADHERING ELASTOMERIC MATERIALS,
by A. S. Krivitsky and W. E. Galwardy. Quarterly rept. no. 1, 1 June-1 Sep 60. 31 Oct 60, 26p. incl. illus. table, 18 refs. (Contract NOw 60-0123-c)

Unclassified report

DESCRIPTORS: Butyl rubber, *Elastomers, Rubber, *Synthetic rubber, Vulcanizates, Butadienes, Acrylonitriles, *Friction, Mechanical properties, Lubrication, Seals, Adhesion, Bearings, Metals, Measurement, Stainless steel, Surface properties, Test equipment, Test methods, Rubber seals, Nitrile rubber, Antiseize compounds.

A simple test apparatus which measures quantitatively the static surface friction properties of elastomeric materials in contact with metals was developed. Preliminary tests were made investigating the effect of variable applied loads on the coefficient of friction of a butadiene-acrylonitrile O-ring compound. A comprehensive literature search on the friction properties of all materials, particularly those with elastomeric properties, was conducted. (Author)

AD-254 806

AD-255 145 Div. 10, 26, 27 U
(4 May 61) OTS price $7.60.

Wyle Research Corp., El Segundo, Calif.
PROBLEMS ENCOUNTERED DURING INSTALLATION AND OPERATION OF A STORABLE PROPELLANT FACILITY FOR TESTING OF TITAN II COMPONENTS AND SYSTEMS.
7 Mar 61, 66p. incl. illus. 27 refs. (Technical note no. 10235) (Contract AF 04(647)398) (AFBMD TN 61-32)

Unclassified report

DESCRIPTORS: *Liquid rocket propellants, *Storage, Handling, Safety, Protective clothing, *Hydrazines, *Methyl hydrazines, Nitrogen compounds, Tetroxides, *Rocket oxidizers, *Rocket fuels, Contamination, Corrosion, Lubricants, Seals, Gaskets, Polymers, Ethylenes, Fluorides, Materials, Test equipment, Test facilities.

Open-ended Terms: Titan.

The design and operation of a storable propellants test facility utilizing N2O4 and the Titan II UDMH-N2H4 fuel blend as the fluid media are discussed. Two 6,000-gal stainless steel tanks provided fuel blend storage and similar tanks to store the N2O4. The blending system consisted of a 6,000-gal tank, two 260-gpm pumps for recirculation, and a high turbulence blender. The toxicity of the storable propellants necessitated the inclusion of aspirators, N purge connections, liquid drain valves and isolation valves to facilitate removal of components from the test sections. Similar precautions were provided for component removal from the fuel and oxidizer systems. Waste propellant liquid and vapor neutralization, material compatibility, contamination, leakage, lubricants, seals, and materials are discussed.

AD-255 145 N1

AD-256 161 Div. 2, 3, 12 U
(22 May 61) OTS price $3.60

Geophysics Corp. of America, Bedford, Mass.
DEVELOPMENT AND TESTING OF IGNITION SYSTEM FOR ROCKET-BORNE SODIUM VAPORIZER,
Nov 60, 33p. incl. illus. tables. (GCA Technical rept. no. 60-1-N)
(Contract NAS5-215)

Unclassified report

DESCRIPTORS: *Igniters, *Ignition systems, Explosives, Pyrotechnics, Thermochemistry, Pellets, *Thermite, Tests, Seals, Rockets, *Sodium, Vaporization, Upper atmosphere, Flares, Booster rockets, Meteorology, Wind.

Open-ended Terms: Sodium vaporizers, Nike-Asp.

The ignition failure of two thermite vaporizer payloads, resulted in an investigation of the ignition parameters of the systems. It was quickly determined that the basic starter thermite ignition system depended upon the integrity of the vaporizer seals and would fail at low pressure. A self-pressurizing ignition system was therefore devised which would be initiated at ambient pressure in order to avoid the seal problem. A commercial hermetically-sealed igniter is used to initiate a large booster which, in turn, initiates the main charge. Limited venting is provided for both booster and canister. The system was test-fired successfully at 3 mm Hg after vibration and shock tests. Canister pressure measurements indicated the ignition sequence to be independent of ambients below 3 mm Hg. It is concluded that the ignition system now developed will prove reliable. (Author)

AD-256 161 N2

AD-256 214 Div. 26, 8, 14 U
(18 May 61) OTS price $4.60

Radio Corp. of America, Somerville, N. J.
MICRO-MODULE CRYSTAL UNITS,
by R. R. Bigler, E. Cohen, and W. H. Liederbach. Final rept. 1 July 60-31 Mar 61. 31 Mar 61, 1v. incl. illus. tables.
(Contract DA 36-039-sc-85046, Proj. 3A 99 15-001-02)

Unclassified report

DESCRIPTORS: *Quartz crystals, Crystals, *Piezoelectric crystals, *Subminiature electronic equipment, Packaging, *Containers, Ceramic materials, Electroplating, Nickel plating, Copper coatings, Seals, Soldering, Tests, Aging, Manufacturing methods, Design.

A ceramic crystal-microelement package was demonstrated to be a practical package capable of meeting all requirements of the Signal Corps Micro-Module program; The packages were made from metallized-ceramic layers. Electrical connections from the notches to the crystal plate were laminated laterally between the layers of ceramic. After firing, the electrical connections and the seal ring were electroplated with nickel, then with copper. The units were tested for hermeticity and solder dipped. Quartz plates with a nominal frequency of 10 mc were used to evaluate the package. The plates were mounted; then, metal caps were ultrasonically soldered in place to seal the packages. A tip-off hole was required in the cap to allow entrapped gas to escape during sealing. This hole was subsequently sealed in a hand-soldering operation. Ultrasonic welding of the cap was also demonstrated. Sealed microelements successfully passed vibration and shock, and temperature-characteristic tests. Two units were encapsulated in element-test Micro-Modules and subjected to 30-day aging tests. Both units remained within the specified limits. (Author)

AD-256 214 N2

AD-256 218 Div. 14, 26, 31 U
(18 May 61) OTS price $1.60

Materials Testing Lab., Portsmouth Naval Shipyard, N. H.
DEVELOPMENT OF SYNTHETIC RUBBER MATERIAL FOR GASKETS AND PACKINGS IN DEEP SUBMERGENCE SUBMARINES,
by Billy H. Swift and Frederick L. Downs. Progress rept. no. 1, 28 Mar 61, 9p. incl. tables.
(Proj. no. SF 013 13 01)

Unclassified report

DESCRIPTORS: *Gaskets, *Rubber gaskets, *Synthetic rubber, Rubber, Nitrile rubber, Elastomers, Submarines, Pressure, Tests.

A total of 9 rubber stocks were tested to determine their various physical properties pertinent to the function of gasket material for deep submergence submarines. Of these 9 stocks, 5 were formulations already available, and the other 4 stocks were formulations developed in the laboratory. An analysis of test data compiled at this time indicates that one of the stocks already available and 2 of the newly developed stocks should be put through a pressure cell, or simulated service test for evaluation as deep submergence gasket material. (Author)

AD-256 218 N2

AD-256 590L Div. 14 U
(25 May 61)

Industrial Lab., Mare Island Naval Shipyard, Vallejo, Calif.
RUBBER O-RING MATERIAL FOR HYDRAULIC SERVICE,
by Arthur W. Scott. Specification development Final rept. 18 May 61, 8p. incl. tables (Rept. no. 5163-60)

Unclassified report

Notice: Only Military Offices may request from ASTIA. Others request approval of Bureau of Ships, Navy Dept., Wash. 25, D. C.

DESCRIPTORS: Seals, *Rubber seals, Tests, *Hydraulic fluids, *Specifications, Esters, Elastomers, Phosphates, Petroleum.

Open-ended Terms: Viton B.

THE PROBLEM: Can Precision Rubber Products Corporation's fluro-rubber vulcanizate No. 19007, based on Viton B, be used as a seal material for both triaryl phosphate ester and petroleum base hydraulic fluids at temperatures approaching 300 F. How does this Viton B compound compare with the best reported one (coded No. 12) tested in previous work. FINDINGS: This formulation is considered unsuitable for use at both temperatures in both fluids, because of excessive softening and modulus decrease. In addition, the hardness as received (15-sec durometer readings) is lower than the generally recommended value. In all these respects, this compound compares unfavorably with the best one (coded No. 12) reported on in previous work, and also does not conform to the limits of the Tentative Specification proposed in Report No. 0879-60 dated 27 June 1960 and added to in the present report. (Author)

AD-256 590L

AD-256 649 Div. 26, 22 U
(26 May 61) OTS price $8.10

Lake City Arsenal, Independence, Mo.
HYDRAULIC CRIMPING UNIT FOR LARGE CALIBER SMALL ARMS AMMUNITION,
by H. L. McLean, May 61, 85p. incl. illus. tables. (IED rept. no. 60-12)
(Proj. 56-242)

Unclassified report

DESCRIPTORS: Hydraulic fluids, Pressure, Seals, Test equipment, Machining, *Crimping tools, *Small arms ammunition, *Projectiles, *Cartridge cases, Rubber, Synthetic rubber, Statistical analyses, Automatic, *Quality control, Ammunition, Attachment, Hydraulic systems, Production.

Open-ended Terms: 20mm, Disogrin.

An investigation was made of the feasibility and adaptability of the basic principles of hydraulic crimping unit for use in mass production in crimping 20mm cartridges. Test equipment was designed and constructed for testing various types of resilient materials. Several design variations were produced and materials tested prior to obtaining a suitable design and material for the crimp ring and seal ring. It was demonstrated that the principles of hydraulic crimping were feasible for adaptation to the V&O Automatic Machine for use in mass production in crimping 20mm cartridges. This method of crimping 20mm cartridges maintains projectile extraction loads, torsion requirements between case and projectile, and moisture sealing qualities within control limits; minimizes cold-working of the case metal. (Author)

AD-256 649 N2

AD-256 678 Div. 10 U
(29 May 61) OTS price $18.50

Bell Aerosystems Co., Buffalo, N. Y.
RESEARCH AND DEVELOPMENT ON THE BASIC DESIGN OF STORABLE HIGH-ENERGY PROPELLANT SYSTEMS AND COMPONENTS,
by Ralph R. Liberto, Final rept. 6 July 59-30 Sep 60. 19 May 61, 281p. incl. illus. tables, 92 refs.
(Contract AF 33(616)6689, Proj. 3058)
(AFFTC TR 60-61) Unclassified report

DESCRIPTORS: *Liquid rocket propellants, Storage, *Nitrogen compounds, *Tetroxides, *Hydrazines, *Methyl hydrazines, Decomposition, Stability, Contamination, Impurities, Temperature, Stainless steel, Titanium alloys, Aluminum alloys, Physical properties, Bearings, Lubrication, Seals, Pumps, Handling, Solubility.

Studies were made of nitrogen tetroxide (N_2O_4), unsymmetrical dimethylhydrazine (UDMH), and hydrazine (N_2H_4). Pertinent physical property data is presented for each propellant. Data concerning propellant decomposition caused by such contaminants as aluminum and stainless steel chips, lint, constituents of air and hot pressurizing gases is included. High temperature, low temperature, and cycling temperature tests were conducted with each propellant in tanks made of type 347 stainless steel, 6061-T6 aluminum, PH 15-7 Mo stainless steel and C 120 AV titanium. Three explosions were encountered with N_2H_4 in stainless steel tanks (347 stainless steel and PH 15-7 Mo stainless steel) at temperatures above 290 F. Other than slight decomposition, no problems were encountered with UDMH stored at temperatures up to 300 F. If N_2O_4 is maintained in the anhydrous state and high vapor pressure can be tolerated, no problems will be expected with N_2O_4 stored in type 347 stainless steel, PH 15-7 Mo stainless steel, and 6061-T6 aluminum tanks up to 270 F. (Author)

AD-256 678 N2

AD-257 025 Div. 14 U
(6 June 61) OTS price $4.60

Convair, Fort Worth, Tex.
THE EFFECTS OF REACTOR RADIATION OF ELASTOMERS AND SEALANTS. IV.
by L. L. Morgan. 9 May 61, 46p. incl. tables.
(Document no. NARF-61-6T; Rept. no. MR-N-174-4)
(Contract AF 33(600)38946)
Unclassified report

DESCRIPTORS: *Elastomers, *Synthetic rubber, Nitrile rubber, Rubber, Urethanes, Polymers, Cyanates, Polymerization, Synthesis, Radiation damage, Gamma rays, Temperature, Mechanical properties, Tensile properties, Neutron bombardment, Tests, Seals.

Polyurethane elastomers are produced through the reaction of certain diisocyanates and polyhydroxy compounds (polyesters and polyethers). Thirteen polyurethane elastomers from 3 manufacturers were irradiated to various doses at several temperatures. The highest dose was 7 x 10 to the 10th power ergs/gm(C) gamma plus accompanying neutrons. The irradiation temperatures ranged from -45 F to 260 F. Following irradiation, stress-strain measurements were made on all of the materials. Compression-set tests were performed on certain selected elastomers. The data tend to indicate that materials from any particular manufacturer were sufficiently similar so that their percentage change from control values were of the same order of magnitude under the same conditions. (Author)

AD-257 025 N3

AD-257 228 Div. 14 U
(13 June 61) OTS price $4.60

Aberdeen Proving Ground, Md.
TX-200 TRANSMISSION OIL COMPATIBILITY PROGRAM, by R. C. Kleinfeld. June 61, 1v. incl. illus. tables (Rept. no. DPS-235)
Unclassified report

DESCRIPTORS: Transmission, Fluids, *Oils, Transmission gears, *Automatic transmissions, Lubrication, Specifications, Tests, Test methods, Effectiveness, Plastic seals, Failure (Mechanics), Acrylic resins, Clutches, Gaskets, Lubricants.

A procedure was developed for laboratory compatibility tests with the TX-200 transmission. Using this procedure a referee base-line condition was established for the TX-200-2X transmission with MIL-L-2104A, OE-10, M14500, REO-148-61 oil. The objectives of the tests were to evaluate MIL-O-10295, OE-S arctic engine oil and MIL-L-2104A, OE-30 engine oil as automatic transmission fluids and to compare the results of the laboratory tests with actual cross-country test-course operation. These tests showed that with reference to the established base line, MIL-O-10295, OE-S, REO-127 oil is incompatible. However, the laboratory test cycle is less severe on clutch wear and more severe on low-range gearing than the 5000-mile cross-country test. Thus it is recommended that a 5000-mile cross-country test be run in a cold climate with arctic engine oil as the transmission fluid to determine whether or not low-sun-gear failures could occur during actual field operation. The MIL-L-2104A, OE-30, M14604 oil was compatible with the transmission except for breakdown of the polyacrylate piston seals. Bench-type investigations should be conducted to determine the cause of seal failure. (Author)

AD-257 228 N3

AD-257 411L Div. 14, 29 U
(12 June 61)

Douglas Aircraft Co., Inc., Santa Monica, Calif.
NON-DESTRUCTIVE LEAK TESTING OF SEALED PACKAGES.
Final rept. 1 Feb 61, 1v. incl. illus. tables, 11 refs.
(Contract NOs(s) 60-6057-c)
Unclassified report

Notice: Only Government Agencies may request from ASTIA. Others request approval of Bureau of Naval Weapons, Navy Dept., Wash. 25, D. C.

DESCRIPTORS: *Packaging, Seals, Tests, *Non-destructive testing, Test methods, Gas diffusion, Instrumentation, Gas leaks, Test equipment, Non-leakage probability, Vacuum pumps, Vacuum systems, Materials, Plastics, Laminates, *Containers, Pressure.

The feasibility was investigated of proposed non-destructive test methods for detection of leaks in sealed packages. These were: (1) Insertion of a detectable contaminant in packages prior to sealing; (2) Use of a pressure chamber to force a radioactive gas into leaking packages, with subsequent detection of contaminated packages with a scintillation counter; and (3) Use of a vacuum chamber to effect detectable expansion of tightly sealed packages. The first method, using Freon gas and a General Electric Halogen Detector, has practical application for testing: (1) long high-production runs of a single type and size of sealed package and (2) large sealed containers, and particularly pressurized containers, at relatively low production rates. The third method can be adapted for use for packaging operations where short runs of a wide variety of sealed packages (excluding large, re-usable type containers) are produced. In the use of the second method, it could be determined that cost factors involved in development and use would be prohibitive, as compared with those for other methods evaluated. (Author)

AD-257 411L

AD-257 940 Div. 26, U
(26 June 61) OTS price $10.50

Republic Aviation Corp., Farmingdale, N. Y.
DESIGN AND DEVELOPMENT OF A 1000 F HYDRAULIC SYSTEM. PART III. PHASE II INVESTIGATION, by William E. Mayhew. June 60, 125p. incl. illus. table
(Contract AF 33(616)6411, Proj. 8128;
WADC TR 59-430, pt. 3) Unclassified report

DESCRIPTORS: *Hydraulic actuators, *Hydraulic systems, *Hydraulic valves, Design, High temperature research, Hydraulic fluids, Servomechanisms, *Hydraulic servomechanisms, Control systems, Aircraft equipment, *Hydraulic seals, Hydraulic reservoirs, *Hydraulic fluid filters, *Pumps, Materials, Test facilities.

Information is presented on the design and fabrication of the components to be used in a hydraulic system where a significant portion of the system will operate at a fluid temperature of 1000 F in an ambient temperature of 1200 F. Further work with regard to the phenoxyphenyl ether previously chosen as the most promising fluid for this system is described. The results of a seal test program, including static and dynamic seals, are also included. In addition, the facilities designed, fabricated, and procured to evaluate the system mockup in the required environments are described. (Author)

AD-257 940 N4

AD-257 978 Div. 14, 7, 26 U
(19 June 61)

Army Signal Research and Development Lab., Fort Monmouth, N. J.
SEALING MOLYBDENUM LEADS TO ALUMINO-SILICATE GLASS,
by Morris Liebson. 5 Oct 60, 1p.
Unclassified report

Reprint from Review of Scientific Instruments 32:204-205, Feb 61. (Copies not supplied by ASTIA

DESCRIPTORS: *Glass, Aluminum compounds, Silicates, Molybdenum compounds, Conductors, Silicon compounds, Seals, *Glass seals, *Metal seals, High temperature research, Molybdenum, Manufacturing methods, Bonding, Molybdenum wire.

A method was devised for fabricating Mo to alumino-silicate glass leads without the use of ovens or elaborate plating methods applied to the Mo. In this method, Mo leads are first beaded with alumino-silicate glass in the usual manner. Close fitting Ni sleeves are then slipped over the Mo wire and pinched into place so that no part of the Mo is exposed. These beaded leads are then sealed into the glass in the usual manner of glassworking. After forming the seal, the Ni sleeve is removed, and the Mo is cleaned. The glass part forming the seal may now be sealed to another piece of alumino-silicate glass; the insulating properties of the glass already in place about this seal will prevent the Mo from reaching the temperature at which copious quantities of molybdenum oxide vapors form. (Author)

AD-257 978

AD-258 535L Div. 14 U
(27 June 61)

Cleveland Graphite Bronze Div., Clevite Corp., Ohio.
SILICONE LUBRICANT-CARBON SEAL INTERACTION AT ELEVATED TEMPERATURES,
by Thomas H. Koenig and E. James Vargo. Rept. for July 59-Sep 60 on Aviation Lubricants. Dec 60, 41p. incl. illus. tables.
(Contract AF 33(616)6714, Proj. 3044)
(WADD TR 60-757) Unclassified report

No automatic release to Foreign Nationals.

DESCRIPTORS: *Lubricants, Seals, *Silicones, Oils, *Carbon seals, High temperature research, Chemical reactions, Esters, Viscosity, Tests, Stability, Carbon, Temperature, Deposits, Deterioration.

Static immersion tests using 14 grades of C in combination with 3 silicone oils, one ester, and one silicone-ester blend at both temperatures of 400, 500, and 550 F gave no indication that C has an effect on viscosity increase or high temperature breakdown of the oils tested. In [illegible] tests where radical changes occurred in samples containing C, there were also radical changes in control samples not containing C; this point was also confirmed in dynamic tests. In a test setup simulating sealing conditions, silicone sludges were formed at indicated interface temperatures of 290 F. This temperature is considerably below the oxidation and thermal decomposition temperatures of the lubricants. Localized hot spots were found to be the source of the high temperature. In one case, temperatures 300 F higher than the average were recorded. Bench rig tests of shaft and face seals produced silicone sludge under conditions far less severe than those encountered in the field. Tests run with an N blanket show that excluding O from the seal interface inhibits or stops the formation of silicone sludge. This method of seal protection will be satisfactory until the seal interface temperature reaches the thermal decomposition limit of the silicone oil, approximately 620 F. (Author)
AD-258 535L

AD-258 784   Div. 8, 7, 14   U
(29 June 61) OTS price $3.60

Battelle Memorial Inst., Columbus, Ohio.
SERVICES, INVESTIGATIONS, AND TESTS ON STANDARD AND NONSTANDARD PARTS AND MATERIALS,
by W. E. Chapin. Final summary rept. 1 May-30 Nov 60 on Resistors. Evaluation of Body Enclosures. 30 Nov 60, 38p. incl. illus. tables. (Contract DA 36-039-sc-85294)
Unclassified report

DESCRIPTORS: *Resistors, Films, *Seals, Encapsulation, Ceramic capacitors, Ceramic materials, Plastic seals, Glass seals, Glass resistors, Soldered joints, Moisture-proofing, Test methods, Phthalates, Tests, *Sealing compounds, Metal seals.

An investigation was made (1) to determine which moisture-resistance-test method or test sequence of pretreatments is most sensitive in differentiating between molded plastics and ceramic resistor body enclosures, (2) to determine whether leak-detection procedures outlined in military specifications can be used with moisture-resistance cycling as a pretreatment and a performance test, and (3) to explore the possibility of whether the static-humidity test can discriminate effectively between 2 distinctly different types of resistor body enclosures. Film-type resistors from 6 sources were subjected to (1) extensive moisture-resistance tests, (2) lead-strength tests, (3) solder-dip application, (4) temperature cycling, (5) salt-water immersion, (6) lead-strength test followed by salt-water immersion, and (7) static-humidity effects. Ten moisture-resistance cycles were sufficient to indicate that resistors with epoxy-tube enclosures and conformal coatings were different from and inferior to other body-enclosure types. Separating the glass, ceramic, and diallylphthalate enclosures in accordance with their differences in degradation rate under repeated moisture-resistance cyclings was not feasible.

AD-258 784   N5

AD-258 865   Div. 14   U
(29 June 61)

Frankford Arsenal, Philadelphia, Pa.
A PRELIMINARY INVESTIGATION OF A THREAD LOCKING COMPOUND,
by James H. Wiland. Mar 61, 25p. incl. illus. tables (Technical memo. M61-20-1)
Unclassified report

For reference at ASTIA Hq. only. This report cannot be satisfactorily reproduced; ASTIA does not furnish copies.

DESCRIPTORS: Screw threads, *Sealing compounds, Seals, Effectiveness, Tests, Metal screws, Self-locking fasteners, Vibration, Impact shock, Test methods, Test equipment, Bolted joints, Self-locking nuts.

An investigation was made to determine the efficiency of Loctite, a liquid sealant, in preventing screws from loosening under shock and vibration. The Loctite was applied to screws in test plates of aluminum, steel, and bronze which were then subjected to combined shock and vibration tests. Breaking and running torque values were measured on the screws both before and after the tests. None of the bonded screws walked out during the tests. The results show that Loctite is a very efficient thread locking device, but a more severe test condition must be found to allow a quantitative evaluation. (Author)

AD-258 865

AD-258 893   Div. 26   U
(3 July 61) OTS price $1.60

Coast Guard, Washington, D. C.
TESTS OF PARKER RING SEAL FOR BUOY POCKETS.
26 June 61, 10p. incl. illus. (Rept. no. 242) (Proj. CGTD J24-2/1-2-6(a))
Unclassified report

DESCRIPTORS: *Seals, *Gaskets, *Rubber gaskets, *Rubber seals, Aluminum, *Buoys, Rings, Tests, *Metal seals, Design.

Tests were conducted to determine the suitability of a Parker Ring Seal for use as a closure gasket in buoy pockets. The gasket assembly consists of an aluminum flat ring having a 24-inch I. D. and 30-1/2-inch O. D. with a neoprene rubber ring insert. The ring seal was installed in a mock-up buoy pocket and air tested to one atmosphere. Fire hose and water submergence tests were also performed. It was concluded that the Parker Ring Seal gasket was adequate for use in Coast Guard buoy pockets. (Author)

AD-259 086   Div. 27, 10   U
(6 July 61) OTS price $5.60

Parker Aircraft Co., Los Angeles, Calif.
DESIGN, DEVELOPMENT, AND TESTING OF A LIQUID HYDROGEN TANK VENT AND RELIEF VALVE,
by A. E. Neal and R. H. Warren. Dec 60, 143p. incl. illus. 10 refs. (Rept. no. 560814-14) (Contract AF 33(616)6710, Proj. 6753) (AFFTC TR 60-71)
Unclassified report

DESCRIPTORS: *Valves, *Propellant tanks, Liquid rocket propellants, Rocket propulsion, Design, Tests, Liquefied gases, Hydrogen, Control valves, Helium, Gas valves, Gas leaks, Gas seals, Low temperature research, Materials, Reliability, *Safety valves.

A summary is presented of the design and development of a liquid H propellant tank vent and relief valve for use in rocket propulsion systems, missiles, and space vehicles. A study was made of the physical properties of liquid H and the properties of materials at liquid H temperatures. Materials were chosen to ensure the reliability, minimum weight, and low leakage of this component. The validity of the valve concept as a pressure control device was well established. A valve was developed with a high degree of steady state accuracy, adequate response characteristics, and consistent crack and reseat characteristics within the specified operating temperature range. An adequate and simple means of thermal compensation of an absolute pressure reference device was developed. Satisfactory operation at -420 F of a valve seat with low seating force and of mylar diaphragms was demonstrated. (Author)

AD-259 086   9   N6

AD-259 160   Div. 7   U
(11 July 61) OTS price $1.10

Army Arctic Test Board, Fort Greely, Alaska.
EVALUATION OF IMPROVED BATTERY FILLER CAPS FOR USE WITH WINTERIZATION KIT FOR M48A2 TANK.
9 July 61, 4p. illus. (Proj. ATB 2-331)
Unclassified report

DESCRIPTORS: *Storage batteries, Arctic regions, *Tanks, Seals, Tests.

Open-ended Terms: M48A2 tank.

Service tests were made to determine the extent the improved battery filler caps corrected the battery heating system deficiency. The Improved Battery Filler Cap is a battery cell cover that provides an acid tight seal of the cell opening. It contains two vent openings that baffle the gases and prevents loss of electrolyte when splashed against the underside of the cap. The cap is threaded and contains a rubber gasket that seals between the top of the cell opening and the cap when the cap is screwed into the opening. It is fabricated from a hard rubber material capable of withstanding more heat than the standard cap. (Author)

AD-259 160   9   N6

AD-259 550L   Div. 22   U
(14 July 61)

Rock Island Arsenal Lab., Ill.
INVESTIGATION OF SOME FACTORS AFFECTING HYDRO-PNEUMATIC RECOIL MECHANISM FLOATING PISTON PACKING PERFORMANCE.
Technical rept.,
by F. K. Blessin and L. R. Raisbeck, 10 Mar 61, 29p. incl. illus. tables, 10 refs. (Rept. no. 61-998)
(Projs. 80304231-15-48401 and 484-58 (ORD))
Unclassified report

Notice: Only Military Offices may request from ASTIA. Others request approval of Rock Island Arsenal Lab., Rock Island, Illinois.

DESCRIPTORS: *Recoil mechanisms, *Howitzers, *Hydraulic seals, *Pneumatic packings, Plastic seals, Tests, Lubrication fittings, Ethylenes, Fluorides, Polymers, Nitrogen, Rubber seals, Silicones, Nitrile rubber.

Open-ended Terms: 105 mm.

An investigation was made to evaluate the effect of packing configuration, filler material, shroud material, lubricators and venting the floating piston void on the performance of floating piston packings in 105 mm howitzer recoil mechanisms. Gymnastication tests were conducted, at ambient temperatures of 225, 75, and -65F. Evaluations, were made of the packing wear and the volume of nitrogen leakage. Results indicated the following: (1) A 3 piece shroud configuration without cup rings wore less and permitted less nitrogen leakage than the standard configuration, (2) More packing shroud wear and nitrogen leakage occurred with fillers made of silicone rubber than of standard nitrile rubber, (3) Packing shrouds made of a filled Teflon material wore less than those of unfilled Teflon, but did not reduce nitrogen leakage, (4) The addition of polyester felt lubricators to the floating piston reduced packing wear and nitrogen leakage, and (5) Less packing wear and nitrogen leakage occurred when the floating piston void was vented to the atmosphere than when 525 psi pressure was maintained. (Author)
AD-259 550L   9

AD-259 574 Div. 24 U
(14 July 61) OTS price $1.[illegible]

Aerospace Technical Intelligence Center, Wright-Patterson Air Force Base, Ohio.
CENTRIFUGAL PUMP FOR AMALGAM,
by L. I. Gel'man. 27 Mar 61, 8p. incl. illus.
(Trans. fr. MCL-884 of Energomashinostroyeniye 9:23-25, Sep 56)

Unclassified report

DESCRIPTORS: USSR, Technological intelligence, *Centrifugal pumps, *Mercury alloys, Test methods, Pumps, Shafts, Hydraulic seals, Mercury, Construction, Tests.

A description is given of a pump for the pumping of amalgam in which the principle of airtight shaft packing of original construction was employed. Data on testing shaft packing of this pump are given. The construction of shaft packing for a centrifugal pump was made for the purpose of eliminating contact between the amalgam and atmospheric air and a test was made to determine the efficiency of pump shaft packing. The obtained results were satisfactory. (Author)

AD-259 653 Div. 7 U
(17 July 61) OTS price $2.60

Eagle-Picher Research Labs., Joplin, Mo.
RESEARCH INVESTIGATIONS LEADING TO THE DEVELOPMENT AND EVALUATION OF A CADMIUM - SILVER OXIDE BATTERY HAVING A HERMETICALLY SEALED CONSTRUCTION.
Quarterly progress rept. no. 3, 1 Jan-31 Mar 61, by J. K. Wilson. 31 Mar 61, 23p. incl. illus. tables.
(Contract DA 36-039-sc-85370, Proj. 3G18-03-001-02)

Unclassified report

DESCRIPTORS: *Storage batteries, *Battery separators, Seals, Alkaline cells, Cadmium, Cadmium compounds, Silver compounds, Oxides, Electrolytes, Design, Recombination reactions, Electrodes.

Several types of regenerated separator materials have become available for test, and some appear promising. Preliminary capacity tests indicate a longer cycle life for the present series of control cells than was obtained from the series of cells previously tested. Preliminary tests indicate that negative plate configurations are available that will increase the maximum allowable overcharge rate many times. The present design of cells is not suitable for long life operation at 165 F. (Author)

AD-259 653 9 N6

AD-259 803 Div. 20 U
(19 July 61) OTS price $1.60

Army Engineer Reactors Group, Fort Belvoir, Va.
PRELIMINARY INVESTIGATION OF SM-1 CONTROL ROD SEAL FAILURE,
by G. H. Knighton. 18 Apr 61, 13p. incl. illus.
(Rept. no. OSB 15)

Unclassified report

DESCRIPTORS: Research reactors, *Control rods, Rods, Neutron absorbers, *Seals, *Metal seals, *Reactor safety systems, Reactor control, Chromium plating, Corrosion, Failure (Mechanics), Nuclear power plants.

Open-ended Terms: SM-1 reactors.

On 26 March, the SM-1 plant experienced excessive control rod drive seal leakage rate. During a vapor container entry for instrumentation check, the leakage rate from each seal was measured. The maximum rate was found to be 6.3 gallons per hour compared to a normal value of approximately .7 gph. Two seals were replaced with rebuilt assemblies. This reduced leakage rates to an allowable operating rate. Shim no. 3 seal was found plugged, and back flushing eliminated the plugging. Visual inspection of the disassembled Rod-B seal indicated leakage increase was due to the increased clearance probably caused by overheating due to loss of cooling water at various times in the operating period. (Author)

AD-259 803 9 N6

AD-260 114 Div. 9, [illegible] U
(25 July 61) OTS price $[illegible]

RCA Defense Electronic Products, Camden, N. J.
DEVELOPMENT OF DIODE MICROELEMENTS.
Formal engineering rept. on Micro-Module Production Program.
17 Mar 58, 101p. incl. illus. tables.
(Contract DA 36-039-sc-75968)

Unclassified report

DESCRIPTORS: *Diodes, Semiconductors, *Subminiature electronic equipment, Electron tubes, Germanium, Silicon, Variable capacitors, Computers, *Manufacturing methods, Electrical properties, Mechanical properties, Tests, Design, Reliability, Seals, Production, Life expectancy, Specifications.

The task of preparing diodes to meet the objectives of the Micro-Module Program was centered around four basic diode types: general-purpose germanium diodes (F-1), diffused silicon computer diodes (F-2), silicon junction zener diodes (G), and silicon-alloy variable-capacitance diodes (H). Specifications were prepared for each group, delineating the electrical, mechanical and environmental performance requirements necessary for a complete evaluation of the diodes. The F-1 diodes were constructed by bonding a fine gold wire to a germanium pellet. The pellet was cleaned and the junction coated with a protective silicone resin. The device was then mounted on the metallized ceramic wafer and soldered to a metal cap. Following this, the top cap was ultrasonically sealed, providing the second diode electrode. The diode microelement was then tested for hermeticity, and aged and inspected for electrical and mechanical properties. Prototype and final-grade F-2, silicon diffused diodes featured a surface passivation treatment. The F-2 diode program was completed satisfactorily. The F-2 and Group H diodes were processed from silicon pellets. After the processes of bonding the leads and baking, the pellets were subjected to surface-passivating treatment and high-temperature aging. The diodes were then mounted on metallized wafers and the electrical and mechanical properties of the devices were determined. The preparation of type G regulator diodes was discontinued. (Author)

AD-260 114 9 X1

AD-260 235 Div. 31, 4 U
(27 July 61) OTS price $2.75

Naval Research Lab., Washington, D.C.
THE PRESENT STATUS OF CHEMICAL RESEARCH IN ATMOSPHERE PURIFICATION AND CONTROL ON NUCLEAR-POWERED SUBMARINES.
Annual progress rept.,
ed. by V. R. Piatt and E. A. Ramskill. 14 July 61, 134p. incl. illus. tables, 128 refs. (NRL rept. no. 5630)

Unclassified report

DESCRIPTORS: *Submarines, Nuclear propulsion, *Carbon dioxide, *Oxygen, Contamination, Atmosphere, Purification, Controlled atmospheres, Sampling, Chemical analysis, Carbon compounds, Monoxides, Aerosols, Hydrocarbons, Photosynthesis, Algae, Sea water, Ventilation, Chromatographic analysis, Gases, Arsines, Ozone, Antimony compounds, Hydrides, Ammonia, Ethylenedinitrile tetraacetates, Hydrogen, Gas leaks, Detection.

OXYGEN SOURCES: The sulfate-cycle system; Algae in a photosynthetic gas exchanger; Chlorate candles; Weight economies of CO2 and O equipment for closed spaces; The analysis of sea water for atmospheric gas content. CO2 REMOVAL: The use of versene to improve MEA performance; Alkazid M. ATMOSPHERE SAMPLING AND ANALYSIS: Gas chromatography in the study of trace organic vapor contaminants; The Mk III and Mk IV atmosphere analyzers; Concentrations of hydrocarbon vapors in submarine atmospheres by carbon sampling; Arsine and stibine analysis; Analysis for ozone; Liquid-O contaminants; Determination of monoethanolamine and ammonia in submarine atmospheres; Description of NRL Model E-5793 leak detector. ELIMINATION OF CONTAMINANTS: The CO/H2 burner; Water-thinned paints for submarine interiors; Aerosols in nuclear submarines; Catalytic combustion of aerosols; Thermal degradation of triaryl phosphates.

AD-260 235 9 N1

AD-260 368 Div. 7, 29 U
(31 July 61) OTS price $5.60

General Electric Co., Syracuse, N. Y.
OPTIMIZATION OF THERMOELECTRIC ENERGY CONVERTERS,
Bimonthly progress rept. no. 2, 15 June-14 Aug 60, ed. by Philipp H. Klein. 14 Aug 60, 49p. incl. illus. tables.
(Contract NObs-78403)

Unclassified report

DESCRIPTORS: *Thermoelectricity, *Generators, *Thermal conductivity, Encapsulation, Oxides, Magnesium compounds, Titanium compounds, Seals, Dioxides, Ceramic materials, Lead compounds, Tellurides, Antimonides, Materials, Digital computers, Thermal insulation, Electric power production, Convection, Cooling, Simultaneous equations, Coolants, Sea water.

A preliminary analysis of the first thermal system to be investigated was nearly completed. This system consists of saturated steam at 533 K (500 F), a thermoelectric generator utilizing thermoelectric materials representative of the present state of the art, and a sea-water heat sink at 291 K (65 F). A magnesia-titania ceramic body was tested and found to be a material suitable as a high-temperature encapsulating medium in conjunction with iron end-caps. Improvements in sealing technology include the construction of an improved sealing apparatus and its use for closing an easily-filled cell that will facilitate the study of thermoelectric properties of encapsulated materials. Encapsulation of p-type lead telluride was successfully accomplished. (Author)

AD-260 368 9 N1

AD-260 385 Div. 13, 31 U
(31 July 61) OTS price $.75

Naval Research Lab., Washington, D. C.
A PORTABLE THERMISTOR-BRIDGE GAS LEAK DETECTOR, by A. Hafner. 3 July 61, 18p. incl. illus. (NRL rept. no. 5647)

Unclassified report

DESCRIPTORS: *Gas leaks, Detection, *Detectors, Mobile, Sensitivity, Gases, *Thermal conductivity, Reliability, *Submarines, Safety devices, Design, *Thermistors, Measurement, Gas flow, Refrigerants, Refrigeration systems, Air conditioning equipment, Shipborne.

A new portable gas leak detector useful for locating leaks in pressurized systems containing any gas having a thermal conductivity different from that of air is described. The detector is uncoupled and hence does not respond to the gas concentration in the ambient air but only to the change in concentration encountered in the vicinity of a leak. Investigations and experiments led to a choice of a thermistor bridge in which optimum response time and minimum sensitivity to gas flow rate variations is obtained by using thermistors shielded from the turbulence of the direct gas flow by glass envelopes but with a hole in each envelope to admit the gas by diffusion. Care was taken in the circuit design to prevent blocking of the high-gain amplifier by noise. Sensitivity of the gas leak detector was checked against a calibrated leak of Freon-12, and the device proved to be useful in locating Freon F-12 leaks of as small as 2 ounces per year. It was used with marked success during submarine operation. (Author)

AD-260 385 9 N1

AD-260 465 Div. 26, 14 U
(31 July 61) OTS price $1.10

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
SUITABILITY OF VITON B O-RINGS FOR USE IN 3000 psi HYDRAULIC SYSTEMS CONTAINING PETROLEUM BASE OR PHOSPHATE ESTER FLUIDS.
Progress rept. no. 2, by A. E. Barrett. 24 July 61, 4p. tables (Rept. no. 92-10)

Unclassified report

DESCRIPTORS: *Hydraulic seals, Materials, *Hydraulic systems, Hydraulic fluids, Rubber, Phosphates, Esters, Petroleum, Ethylenes, Fluorides, Polymers, Tests, Effectiveness, Synthetic rubber, Cyclohexanes.

Open-ended Terms: Viton, Teflon.

Results of dynamic performance tests of Viton B O-rings indicate that Viton B O-rings of 70 Shore A hardness with Teflon backup rings will perform satisfactorily at 180 F in 3000 psi hydraulic systems containing phosphate ester fluids. In addition, the results indicate that the O-rings will perform equally as well under these conditions in hydraulic systems containing petroleum base fluids. As reported previously, the Teflon backup rings do not stand up at 300 F under the test conditions. Backup rings made of a 90 Shore A Viton B stock showed even greater wear. (Author)

AD-260 465 9 N1

AD-260 526L Div. 26, 14 U
(2 Aug 61)

Douglas Aircraft Co., Inc., Santa Monica, Calif.
NON-DESTRUCTIVE LEAK TESTING OF SEALED PACKAGES.
Monthly progress rept. no. 7 for Oct 60. 15 Nov 60, 8p. incl. illus. table. (Contract NOa(s) 60-6037-c)

Unclassified report

Notice: Only Government Agencies may request from ASTIA. Others request approval of Bureau of Naval Weapons, Navy Dept., Wash. 25, D. C.

DESCRIPTORS: Gas leaks, Gas diffraction, *Packaging, Seals, Plastics, Films, Resins, Paper, Fluorides, Methanes, Gases, Vacuum Seals, Halogens, Test methods, Detection, Reliability, *Non-destructive testing.

Equipment and instrumentation for running the various study programs is now available and in operation. Accumulation of data is progressing at a much accelerated pace. Studies in progress include: gas transmission through flexible barriers and through measured holes in foil; vacuum effect studies with equipment available for varying the rate of vacuum application and for accurate detection and measurement of very slight expansion of packages; and studies of chamber design and probe positioning for detection of halogen leakage through measured holes in flexible barrier envelopes. (Author)

AD-260 526L 8

AD-260 629 Div. 7 U
(4 Aug 61) OTS price $5.60

Gulton Industries, Inc., Metuchen, N. J.
THE TESTING OF HERMETIC SEALS AND SEPARATORS, POSITIVE PLATE STUDIES, AND INVESTIGATIONS ON THE UNIFORMITY OF SEALED NICKEL-CADMIUM CELLS.
Quarterly progress rept. no. 3, 1 Jan-31 Mar 61, on Investigations Leading to the Development of Improved Sealed Nickel-Cadmium Batteries, by H. N. Seiger. 31 Mar 61, 45p. incl. illus. tables. (Contract DA 36-039-sc-85390)

Unclassified report

DESCRIPTORS: *Alkaline cells, *Seals, Design, *Battery separators, Electrodes, Ceramic materials, Metal seals, Test methods, *Anodes (Electrolytic cells), Nickel, Cadmium, *Storage batteries.

Ceramic-to-metal seals are now produced consistently. Accelerated testing of separator materials was completed, and the properties of separators are reviewed. Overcharge data were obtained on 6 cells. The data consists of the capacity, Tafel slope and pressure slope. Several cells with a ratio of negative to positive capacity near unity, were brought into several different states of charge. These cells were then pressured with oxygen and the pressure decrease obtained. The question of oxygen adding capacity to a cell was investigated, and a negative answer obtained. The oxygen evolution data was analyzed on a phenomenological basis. The theory of the analysis was developed. It appears that pure nickel has 2 sources of oxygen which have half-times of 9 and 102 minutes respectively. A third self-discharge process is also occurring, but no data has been obtained on this. The doped electrodes have 2 self-discharge processes which are suspected of being one oxygen source undergoing sequential decay. The potential decay data is reported as the Tafel slopes. (Author)

AD-260 629 9 N1

AD-260 791 Div. 25 U
(4 Aug 61) OTS price $1.60

Oklahoma U. Research Inst., Norman.
THE MELTING CURVES OF TIN AND SELENIUM TO 10,000 BARS.
by Stanley E. Babb, Jr. June 61, 18p. incl. illus. 10 refs. (Technical rept. no. 1) (Contract Nonr-98205)

Unclassified report

DESCRIPTORS: *High pressure research, Laboratory equipment, Pressure vessels, Pressure gages, Metal seals, Heaters, Circuits, *Tin, *Selenium, *Melting, Temperature, Pressure, Design.

Melting studies were made on Sn and Se, and equations are given which can be used to calculate the melting pressure of these substances at any temperature to pressures of 10,000 bars. The intensifier for the generation of pressure consisted of a ram of about 4.5-in. inner diameter and 6-in. outer diameter which drives a 0.654-in.-diam piston. The steel pressure cell has a 0.5-in. inner diameter and a 3.0-in. outer diameter with a usable bore of 6-in. Pressure measurements were obtained with a manganin wire resistance gage. A 1000-w electric heater was used to heat a bath of Na, K, and Li nitrates which remained fluid below 150 C and did not cause corrosion of the steel parts immersed in it. Details of the heater and gage circuits and pressure seals are presented.

AD-260 791 4 N1

AD-260 920 Div. 14, 26, 31 U
(8 Aug 61) OTS price $1.60

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
DEVELOPMENT OF SEALS FOR HIGH PRESSURE AIR SYSTEMS IN SUBMARINES.
Progress rept. no. 5, by R. D. Ford. 31 July 61, 12p. incl. table. (Rept. no. 28-7) (Proj. no. S-F013-13-01)

Unclassified report

DESCRIPTORS: *Seals, *Rubber seals, Synthetic rubber, Submarines, *Pneumatic systems, High pressure research, Effectiveness, Tests, Physical properties, Mechanical properties, Air, Compressed air, Materials.

Studies were conducted to develop a formulation for rubber seals to be used in the high pressure air systems of FBM submarines. The best stock which the Laboratory had developed previously for this service was a Viton A-HV stock identified as Compound A. An objectionable feature of this stock was its tendency to backrind when molding. A new Viton A-HV stock (377-89) was developed which does not backrind and which is also suitable for use in seals for high pressure gas service. Four sets of O-rings submitted by the Parker Seal Company were tested for resistance to swelling caused by rapid release of 5000 psi air pressure. Two of the sets of O-rings were fairly resistant to swelling, but were not equal to Compound A or 377-89 in this regard. O-rings made by Linear Incorporated and conforming to Classes A and B of specification MIL-P-5516B were tested for swelling. Both the Class A (88 Shore) and Class B (68 Shore) O-rings were seriously swollen, and the Class B O-rings exhibited many small blisters after decompression. The Class A O-rings were considered to be of marginal utility in high pressure air systems. The Class B O-rings were considered to be unsuitable for this purpose. (Author)

AD-260 920 9 N2

AD-261 561   Div. 26, 14, 31   U
(21 Aug 61) OTS price $1.60

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
DEVELOPMENT OF SEALS FOR HIGH PRESSURE AIR SYSTEMS IN SUBMARINES.
Progress rept. no. 1,
by R. D. Ford and R. E. Morris. 28 Dec 60, 11p. incl. tables (Rept. no. 28-3)
(Proj. S-F013-13-01)

Unclassified report

DESCRIPTORS: *Rubber seals, *Seals, Rubber, Synthetic rubber, Submarines, High pressure research, High pressure compressors, Lubricants, Petroleum, Esters, Phosphates, Fluorides, Vulcanizates, Nitrile rubber, Elastomers, Mechanical properties, Fluorocarbons, Hardness, Tests, Materials.

Rubber O-ring seals for high-pressure submarine air systems were tested for their ability to withstand sudden pressure drops from 4500 to psig. Stocks were prepared from nitrile, carboxylated nitrile, polyurethane, and polysulfide rubbers for service with petroleum-based fluids; stocks prepared from fluoro rubbers were also tested for service with phosphate-ester fluids. Hardness ranged from 83 to 96 Shore A. Damage was appraised by the volume changes of O-rings from these vulcanizates. The results indicated that nitrile, carboxylated nitrile, polyurethane, and fluoro rubber vulcanizates would be satisfactory seals. It appeared that seal stock for service with either type of lubricant could be developed from the fluoro rubber.

AD-261 561   9   N3

AD-261 562   Div. 26, 14   U
(21 Aug 61) OTS price $1.60

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
DEVELOPMENT OF SEALS FOR HIGH PRESSURE AIR SYSTEMS IN SUBMARINES.
Progress rept. no. 2,
by R. D. Ford. 2 Feb 61, 6p. incl. tables (Rept. no. 28-4)
(Proj. S-F013-13-01)

Unclassified report

DESCRIPTORS: *Rubber seals, *Seals, Rubber, Synthetic rubber, Submarines, High pressure compressors, High pressure research, Lubricants, Petroleum, Esters, Phosphates, Fluorocarbons, Vulcanizates, Nitrile rubber, Elastomers, Carbon black, Carboxylic acids, Mechanical properties, Hardness, Materials, Polymers, Tests.

Stocks were prepared from Hycar 1071 loaded with different high-structure carbon blacks. These were intended for use with petroleum-based lubricants. The hardness of the various vulcanizates ranged from 70 to 90 Shore A. The damage to the vulcanizates caused by a sudden drop of air pressure from 5000 to 0 psig was appraised by measuring the change in volume of O-rings made from the vulcanizates. The results of the tests indicated that all of the stocks are satisfactory for this service. These findings are to be tested by simulated service tests. The development of stocks for O-rings for use with either petroleum or phosphate-ester lubricants is being continued. (Author)

AD-261 562   9   N3

AD-261 563   Div. 26, 14, 31   U
(21 Aug 61) OTS price $1.60

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
DEVELOPMENT OF SEALS FOR HIGH PRESSURE AIR SYSTEMS IN SUBMARINES.
Progress rept. no. 3,
by R. D. Ford. 11 Apr 61, 13p. incl. illus. tables (Rept. no. 28-5)
(Proj. S-F013-13-01)

Unclassified report

DESCRIPTORS: *Rubber seals, *Seals, Rubber, Synthetic rubber, Vulcanizates, Nitrile rubber, Elastomers, Submarines, High pressure compressors, High pressure research, Lubricants, Petroleum, Esters, Phosphates, Fluorides, Fluorocarbons, Carbon black, Mechanical properties, Hardness, Materials, Tests.

Stocks were prepared from both Viton A-HV and Viton B loaded with various high-structure carbon blacks. All stocks had a nominal hardness of 80 Shore A. The damage to the vulcanizates caused by a combination of immersion in either the phosphate-ester or petroleum-base oils and a sudden drop in air pressure from 5000 to 0 psig. was appraised by measuring the change in volume of O-rings made from the vulcanizates. Vulcanizates of Viton B and Viton A-HV appeared equal in resistance to these conditions. After conditioning for 46 hr or more in 5000 psig air followed by rapid decompression, the best Viton A-HV O-rings developed to date were observed on occasion to develop blisters on the surface of the O-rings. The development of improved Viton stocks having greater resistance to swelling and blistering is being continued. (Author)

AD-261 563   9   N3

AD-261 564   Div. 26, 14, 31   U
(21 Aug 61) OTS price $1.60

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
DEVELOPMENT OF SEALS FOR HIGH PRESSURE AIR SYSTEMS IN SUBMARINES.
Progress rept. no. 4,
by R. D. Ford and R. E. Morris. 9 June 61, 16p. incl. illus. tables (Rept. no. 28-6)
(Proj. S-F013-13-01)

Unclassified report

DESCRIPTORS: *Rubber seals, *Seals, Rubber, Synthetic rubber, Vulcanizates, Nitrile rubber, Elastomers, Fluorocarbons, Submarines, High pressure compressors, High pressure research, Lubricants, Petroleum, Esters, Phosphates, Silicones, Fluorides, Carbon black, Mechanical properties, Tensile properties, Materials, Tests.

Viton A-HV (Compound A) stocks and a Viton B stock were tested for tensile properties, and for swelling after lengthy exposure to air at 5000 psi pressure followed by rapid decompression.

One of these Viton A-HV stocks proved to be outstanding in all respects. Its swelling after decompression was again checked, this time after prior immersion in petroleum-base oil and in phosphate-ester oil. The swelling was found to be very moderate (10% at most). A formulation prepared from fluorosilicone rubber was tested for tensile properties and for decompression swelling after prior immersion in the oils and lengthy exposure to air at 5000 psi pressure. Its tensile properties were poor, but its resistances to the oils and to decompression swelling were excellent. It was considered to be suitable for static service only. O-rings made from a Hycar 1071 stock were installed in the air compressor on the USS THEODORE ROOSEVELT, which is lubricated by petroleum-base oil. The O-rings have sealed satisfactorily for at least 349 hours. (Author)

AD-261 564   9   N3

AD-261 886   Div. 22   U
(11 Sep 61)

Aberdeen Proving Ground, Md.
PRELIMINARY ENGINEERING TEST OF AN O RING AS A LOADING-PLUG SEAL FOR CASE, CARTRIDGE, 90-mm, XM114E1,
by R. N. Dempsey. Aug 61, 15p. incl. tables. (Rept. no. DPS-320)

Unclassified report

No automatic release to Foreign Nationals.

DESCRIPTORS: Projectiles, *Cartridge cases, Loading, Cartridges, Tests, *Seals, *Cartridge case plugs, High explosive ammunition, Antitank ammunition, Obturation, Fin stabilized ammunition.

Open-ended Terms: M41 Gun, T300 Projectile, M114 cartridge case, O Rings, 90-mm, M36 Gun.

The 90-mm, XM114E1 cartridge case was designed for use with the T300 series projectile. Gas leakage, limiting the retention of the loading plug (with possible resultant damage to the breechblock), was experienced in case-acceptance and complete-round firings. To eliminate this problem an O ring used as a loading-plug seal was tested. One hundred and eighty cases were fired at chamber pressures of from 34,000 to 56,000 psi and at conditioning temperatures of -40 F, 70 F, and +140 F. Less than 5% of the cases failed to eject, and even these cases extracted from 12 to 22 inches and were hand free. There was slight extrusion of the O ring on all rounds fired over 50,000 psi chamber pressure, but this office deems it not objectionable. There was no gas leakage around the loading plug or the primer, no cracks or splits, internal or external fluting in the case side wall, no loosening of the loading plugs, no damage to the breechblock, or other defects. It is recommended that the O ring be immediately incorporated in the manufacture of the XM114E1 cartridge cases. (Author)

AD-261 886   9

AD-261 958   Div. 22, 26, 13, 30   U
(29 Aug 61) OTS price $9.60

Armour Research Foundation, Chicago, Ill.
TESTS ON ATLAS BLAST VALVES.
Final test rept. no. 20 (Preliminary) on Blast Effects on Buildings and Structures and Protective Construction Operation of Six-foot and Two-foot Shock Tubes,
by James J. Swatosh, Jr., 30 Apr 61, 112p. incl. illus. tables.
(Contract AF 29(601)796, Proj. 1080)
(AFSWC TN 61-16)   Unclassified report

DESCRIPTORS: *Underground structures, *Shock tubes, *Air blast, *Blast, Seals, *Valves, Nuclear explosions, Atomic bomb explosions, Pressure vessels, Loading, Shock waves, *High pressure valves, Hydraulic systems, Pistons, Plastic seals, High temperature research, Low temperature research, Test facilities, Tests, Failure (Mechanics).

Open-ended Terms: Atlas.

The MS-1073-1 Silo Blast Closures, 16- and 42-in. were subjected to a series of performance experiments. These performance experiments were designed to simulate the actual operating conditions and/or environments to which the closures would be subjected during normal operation. The 16-in. closures reacted satisfactorily to all the performance experiments with the exception of cold environments. They were subjected to maximum test blast load without failure. During the cold environment experiments, pressure leaks in the hydraulic system of the closures developed at approximately ° F. These pressure leaks resulted from the shrinkage of the teflon seals in the system. The 42-in. closures responded satisfactorily only to the hot environment experiments. The temperature of the 42-in. closures was raised to 3°° F before any failures occurred. Pressure leaks in the hydraulic system occurred when the closures were air-blast loaded. Extensive damage developed during repeated cycling experiments. (Author)

AD-261 958   9   N3

AD-262 0?? Div. 14, 26 U
(28 Aug 61) OTS price $3.60

Frankford Arsenal, Philadelphia, Pa.
THREADED FASTENERS LOCKING DEVICES EVALUATION OF LOCTITE SEALANT,
by S. H. Kalen. May 61, 34p. incl. illus. tables (Memo. rept. no. M61-29-1)
Unclassified report

DESCRIPTORS: *Seals, *Sealants, Effectiveness, Tests, Shock resistance, Vibration.

This report covers a study of the effectiveness of LOCTITE sealant as a locking method for the prevention of loosening of threaded fasteners under shock and vibration conditions. The weakest grade, LOCTITE H, was found to be as effective as the dab type materials MIL-V-173 varnish and MIL-M-11011 sealing compound under the test conditions used. However, LOCTITE sealant compounds should not be relied upon for fastening under high frequency, high amplitude vibration conditions. Application after fastener insertion is undesirable since extent of penetrability is unpredictable. (Author)

AD-262 196 Div. 27, 10 U
(30 Aug 61) OTS price $1.60

Thompson Ramo Wooldridge, Inc., Rocky Mount, Va.
PUMPING HYDRAZINE WITH A TURBOPUMP,
by I. Sheer. 14 Aug 61, 17p. incl. illus.
(ER-4152)
(Contract AF 33(616)6774)
(AFFTC TR 60-53)
Unclassified report

DESCRIPTORS: Pumps, *Fuel pumps, *Centrifugal pumps, *Hydrazines, Feasibility studies, Temperature, Materials, Stainless steel, Polymers, Fluorides, Ethylenes, Seals, Cavitation, Design, Tests, Liquid rocket propellants, Rocket fuels.

Feasibility studies of pumping hydrazine with a turbopump are reported. The results show that vapor lock, cavitation, and dead ending can be tolerated with hydrazine. Careful attention must be paid to the temperature rise which results from dead ending or flow stoppage due to vapor lock, however the rise in the case of the pump tested was gradual enough to allow corrective action by an observer. Materials chosen for construction of impeller, housing, seals, and shaft stood up well under a wide range of conditions of operation for a total running time of 55 minutes and 33 seconds. (Author)

AD-262 196 9 N3

AD-262 398 Div. 22 U
(5 Sep 61)

Aberdeen Proving Ground, Md.
SPECIAL BALLISTIC TEST OF 105-MM CARTRIDGE CASES, M115B1, XM148E1 AND XM148B1,
by R. N. Dempsey. Aug 61, 16p. incl. tables.
(Rept. no. DPS-315)
Unclassified report

No automatic release to Foreign Nationals.

DESCRIPTORS: *Cartridge cases, Projectiles, Tests, Seals, Cartridge case plugs, Ballistics, Steel, Loading, High explosive ammunition, Antitank ammunition.

Open-ended Terms: O rings, 105-mm, M115 cartridge case, M148 cartridge case, T384 projectiles.

The M115 and XM148 105-mm cartridge cases were designed for use with projectile, APDS-T, M392E1 and projectile, HEAT, T384, respectively. This ballistic test was conducted to determine the feasibility of substituting steel for the currently used brass M115 and XM148 cases, the effectiveness of an O ring as a loading-plug seal on the XM148 cases, and the most satisfactory loading plug for the XM148 cases with regard to material and thread design. The M115B1 (steel) cases performed satisfactorily when fired at excess pressure (68,000 psi) at 70 F. There was slight gas leakage around the primer of one case at -40 F. The XM148E1 (brass) cases generally performed satisfactorily, but there was slight gas leakage around the locking plug on six of the cases when they were fired at a pressure of 54,000 psi at 70 F. All the XM148B1 (steel) cases performed satisfactorily in every respect. The use of O rings reduced gas leakage significantly. (Author)

AD-262 398 9

AD-262 566 Div. 12, 31 U
(7 Sep 61) OTS price $4.60

Naval Weapons Lab., Dahlgren, Va.
TARTAR STRUCTURAL FIRING TEST ABOARD THE USS CHARLES F. ADAMS (DDG-2),
by J. W. Loving, T. I. Hodson, and others.
15 Aug 61, 47p. incl. illus. tables (NWL rept. no. 1770)
Unclassified report

DESCRIPTORS: *Guided missiles, Surface to air, Guided missile launchers, Test vehicles, Launching, Shipborne, Rocket motors, *Blast, Measurement, Tests, Test methods, Structures, Gas leaks, Safety, Destroyers, Guided missile personnel, Damage control, *Explosion damage, Poisonous gases, Noise, Temperature, Exhaust flames, Ship decks, Hazards, Booster rockets, Guided missile boosters.

Open-ended Terms: Tartar, DDG-2.

Eight TARTAR Mk 1 Mod 0 dual-thrust rocket motors with concrete slugs were fired aboard the USS Charles F. Adams (DDG-2) to investigate the adequacy of the protection for the Tartar launching system personnel against blast effects and to determine the effects of the booster blast on the ship's structure. The test vehicles were fired at various angles such that the exhaust stream was directed at areas thought to impose the most severe conditions on the ship's structural components and equipment. Measurements were made of structural strains, toxic gas concentrations, noise levels, flame penetrations at door and port seals, temperature changes inside the ship and in the blast area of the 01 level and the main deck. High-speed motion pictures were taken on all tests. The results indicated severe gas leakage into mount 52 and other areas adjacent to the launcher and minor damage to mechanical parts of the launching system. A detailed description of all data obtained and ship damage incurred is included. (Author)

AD-262 566 9 N4

AD-262 580 Div. 8 U
(6 Sep 61)

Sperry Gyroscope Co., Great Neck, N. Y.
INVESTIGATION OF MICROWAVE WINDOW FAILURE MECHANISMS AND THEIR ELIMINATION.
Quarterly progress rept. no. 2, 1 Jan-1 Apr 61,
by D. B. Churchill and E. W. Cheatham. Apr 61, 29p. incl. illus. (Rept. no. NA-8220-8235-2)
(Contract DA 36-039-sc-87389; Continuation of Contract DA 36-039-sc-78314)
Unclassified report

No automatic release to Foreign Nationals except Gt. Britain and Canada.

DESCRIPTORS: *Waveguide windows, *Microwave equipment, Seals, *Electron tubes, Dielectrics, Materials, Waveguide filters, Electron bombardment, Electron transitions, Cavity resonators, Sapphires, Measurement, Reflection, Wave transmission, Beryllium, Waveguides, Waveguide irises, Design, Tests, Ceramic materials, Propagation.

The high-energy electron bombarder was repaired and reassembled. Comparative breakdown tests showed that the rectangular block window has a threshold of breakdown lower than that of a disc window with suitably contoured transition corners. Breakdown on the dielectric surface of the block is a disadvantage and may be a factor responsible for having a lower threshold than the disc, in which breakdown occurs at the transitions rather than at the dielectric surface. The graphical design of a double-resonator filter window using a sapphire disc was confirmed by measurements on an experimental model. Measurements of the reflection coefficient of a seated-slab window using a beryllia slab indicated that the structure has somewhat less bandwidth than a self-resonant block window in rectangular waveguide. (Author)

AD-262 580 9

AD-262 718L Div. 1 U
(11 Sep 61)

Aeronautical Engine Lab., Naval Air Material Center, Philadelphia, Pa.
EVALUATION OF WIGGINS PART NO. GS51218-3 COUPLING FOR USE WITH AIRCRAFT GROUND COMPRESSOR DUCTS.
Rept. on Phase 1,
by Donald B. McCormick. 16 Aug 61, 9p. illus.
(Rept. no. NAMC-AEL-1689)
Unclassified report

Notice: Only Government Agencies may request from ASTIA.

DESCRIPTORS: *Couplings, Disconnect fittings, Tests, Test methods, Compressors, Ducts, Aircraft, Gas turbines, Ground support equipment, Seals.

The purpose of the assignment was to determine the suitability of quick-disconnect couplings for use in connecting flexible ducting to normally ground stationed gas turbine compressors and to aircraft aboard which pneumatically powered equipment is located. It was concluded from the results of the tests that: (1) the coupling is unsafe for service use because the mechanism which locks the coupling to the mating nipple periodically malfunctions and because the end fitting attached to the coupling body is not secured positively; (2) O-ring seal durability and pressure lock functioning are unsatisfactory; and (3) the pressure lock assembly will function as a safety device despite failure of the O-ring seal. (Author)

AD-262 718L 9

AD-263 029L Div. 1, 29 U
(19 Sep 61)

Air Crew Equipment Lab., Naval Air Material Center, Philadelphia, Pa.
MERCURY SPACE SUIT. TEST AND EVALUATION OF MERCURY HELMET VISOR SEAL BOTTLE.
Test rept.,
by Donald F. Hughes. 21 Aug 61, 21p. incl. illus. tables (Rept. no. NAMC-ACEL-458)
(Problem Assignment no. C44AE22-29)
Unclassified report

Notice: All requests require approval of Bureau of Naval Weapons, Navy Dept., Wash. 25, D. C.

DESCRIPTORS: *Pressure suits, *Helmets, *Vacuum bottles, Oxygen, *Hoses, Materials, *Rubber hose, Silicones, Tests, Vibration, Pressure, Vacuum seals, Heat, Spin, Deterioration, Storage.

Twelve silicone hose and bottle units were submitted to an oxygen weight leakage loss over a fixed time period by filling the bottles with oxygen to 1800 psig pressure, weighing them and setting them aside for the required 72 hour time period and then reweighing them. The condition of these units while being tested was that eight units were tested with a capped male fitting on the hose disconnect, one without the cap, three with the regulator in the off position, and one with the hose removed from the swivel fitting and the fitting opening soldered closed. By deduction from analysis of the data obtained from this above mentioned oxygen weight leakage loss over a 72 hour time period test, it was found that the silicone hoses had excessive porosity and exceeded the allowable loss of 5 per cent by weight of oxygen in 72 hours for the complete hose, regulator and bottle assembly as one integral unit. (Author)

AD-263 029L 9

AD-263 298 Div. 26, 14 U
(TIPSH/BRW) OTS price $1.10

Aeronautical Materials Lab., Naval Air Material Center, Philadelphia, Pa.
MIL-F-22191 (AER) QUALIFICATION TEST RESULTS ON THE RAP-IN-WAX COMPANY PRODUCT FILM-O-RAP 5500,
by E. J. Lion. 20 Apr 61, 1p. (Rept. no. NAMC-AML-1230)
Unclassified report

DESCRIPTORS: *Packaging, Materials, Films, Thin films, Ethylenes, Esters, Polymers, Military requirements, Specifications, Tests, Seals, Storage.

The test results indicate that the transparent material, designated Film-O-Rap 5500, is in conformance with the Type II requirements of MIL-F-22191 (Aer) except the one year storage which will be completed March 1962. The sealing conditions are as follows: 350 F, 30 psi, 3-sec dwell time for Packaging Industries, Inc., Portable Heat Sealer, Model 6, 1.5-sec impulse time, 3.0-sec cooling time, 30% of nominal 110 v, 25-lb line pressure, 30 psi effective jaw pressure for Packaging Industries, Inc., Impulse Sealer, Model 12SC and a dial setting of 8 for Vertrod Corporation, Impulse Sealer, Model 13A. (Author)

AD-263 298 9 M5

AD-263 306 Div. 14 U
(TIPSH/BRW) OTS price $1.10

Aeronautical Materials Lab., Naval Air Material Center, Philadelphia, Pa.
QUALIFICATION TEST REPORT ON DODGE CORK COMPANY'S 711-BN AND 100-N GASKET MATERIALS TESTED IN ACCORDANCE WITH MIL-G-6183, AMENDMENT-4, GASKETS AND SHEET GASKET MATERIAL; SYNTHETIC RUBBER AND CORK COMPOSITION. CLASS I. FIRM; AND CLASS II. MEDIUM, RESPECTIVELY OF 22 AUGUST 1947,
by E. J. Celuziono. 8 Dec 60, 3p. (Rept. no. NAMC-AML-22090-61)
Unclassified report

DESCRIPTORS: (*Gaskets, Sheets, Seals, Military requirements, Specifications, Tests.) (*Cork gaskets, Gaskets, Materials, Synthetic rubber.)

Test data indicated that Dodge Cork Company's sheet gasket materials 711-BN and 100-N conform with all the qualification requirements of Military Specification MIL-G-6183, Amendment-4, Class I, Firm; and Class II, Medium, respectively. (Author)

AD-263 391 Div. 8, 14 U
(TIPSE/CNJ) OTS price $3.60

Sperry Gyroscope Co., Great Neck, N. Y.
HIGH AVERAGE-POWER R-F WINDOW STUDY.
Quarterly memo. no. 2, 26 Apr-26 July 61,
by D. B. Churchill and E. M. Cheatham. Aug 61, 29p. incl. illus. (Sperry rept. no. NA-8220-8261-2)
(Contract AF 30(602)2428, Proj. 5571)
Unclassified report

DESCRIPTORS: (*Waveguide windows, Dielectrics, Ceramic materials, Single crystals, Synthetic gems, Sapphires, Aluminum compounds, Beryllium compounds, Wave transmission, Electron bombardment, Seals, Design, Tests.) Microwave equipment, Broadband, Electron tubes, Waveguides. (Cavity resonators, Waveguide couplers, Design.)

The literature study and survey of the commercially available dielectric materials for window applications was completed. It was found that single-crystal synthetic sapphire, cut with the axis of the disc parallel to the C-axis of the crystal, is the best available material from the standpoints of mechanical strength and dielectric loss factor. High-energy electronic bombardment tests on sample windows of sintered alumina and crystalline synthetic sapphire were completed. The final oven brazes of the hybrid coupler assembly, a component of the high-power resonator, were completed. Also, the high-directivity monitor coupler was designed, and a prototype model was satisfactorily tested. The high-power water load was designed, and a prototype model was completed. A nearly perfect match at the specified operating frequency was exhibited by the model. An investigation was made of the problems of sealing rectangular block windows. Satisfactory transmission characteristics were achieved on 2 prototype models of broadband disc windows. The operation frequency bands overlap slightly, with the test frequency of 8.35 kmc in the region of overlap. (Author)

AD-263 391 4 M5

AD-263 457 Div. 10, 27, 30 U
(TIPSH/RED)

Beech Aircraft Corp., Boulder, Colo.
HIGH TEMPERATURE FUEL SYSTEMS INVESTIGATIONS.
Summary rept. 1 Apr 60-31 May 61 on Propulsion Fluid and Heat Transfer Subsystems,
by R. W. Burgess, V. E. Isakson and others.
Aug 61, 54p. incl. illus. tables (Engineering rept. no. 11152)
(Contract AF 33(616)7073, Proj. 3064)
(ASD TR 61-312) Unclassified report

No automatic release to Foreign Nationals.

DESCRIPTORS: (*Jet engine fuels, Aviation fuels, Hydrocarbons, Fuel additives, High temperature research, Thermal stresses, Fuel filters, Deposits, Chromatographic analysis, Spectrographic analysis, Tests.) (Jet engine fuels, Vapors, Corrosion, High temperature research, Tests, *Fuel systems, *Fuel tanks, Valves, Seals, Gaskets, Plastics, Metals, Steel, Aluminum alloys, Titanium alloys.)

High-temperature hydrocarbon fuels were subjected to coker, gas chromatographic, and IR spectrophotometric analysis to determine the effects of various additives and thermal stressing. Aircraft fuel system components were installed in the preconditioning system to determine their suitability for operation under the test conditions, particularly with respect to elastomeric seals. Metal samples were physically stressed and subjected to the higher temperature ranges, both immersed in liquid and suspended in the vapor space. The tendency of the fuels to produce solids and the accumulation of the solids in the preconditioning system were continuously monitored. Appreciable solids accumulation was encountered. No accelerated stress corrosion occurred. It is thought that the duration of the exposure was insufficient to produce concrete results. Proper containment and restraint can extend the operating range of elastomeric seals considerably over their normal high temperature rating. (Author)

AD-263 457 9

AD-263 485L Div. 1 U
(TIPSE/CBM)

Chance Vought Aircraft Inc., Dallas, Tex.
INTEGRATED FLIGHT CAPSULE COCKPIT SEALING AND ENCLOSURE STUDY.
Interim rept., 1 June 59-1 Mar 60,
by W. F. Goebring and W. R. Dobels. 25 Mar 60, 16p. incl. illus. (Rept. no. AER-EOR-12826)
(Contract NOa(s) 59-6150-c)
Unclassified report

Notice: Only Government Agencies may request from ASTIA. Others request approval of Bureau of Naval Weapons, Navy Dept., Wash. 25, D. C. Attn: Code RAAE-2312.

DESCRIPTORS: (*Jettisonable cockpits, Naval aircraft, Jet fighters, Jet planes.) (Cockpits, Seals, *Jettisonable cockpits, Design, *Aircraft canopies.) (Airframes, Sandwich construction, Aluminum, Steel, Glass textiles.)

Open-ended Terms: Integrated flight capsule, FBU.

A study was made of zero cavity sealing within the cockpit and what is involved in the system in terms of occupant safety, arrangement and design, construction and materials required, and recommended tests and development work in conjunction with the program. Various types of cockpit enclosures are discussed with emphasis placed on the one which appears to meet all the objectives necessary for a desired cockpit enclosure. (Author)

AD-263 485L 0

AD-263 640 Div. 8 U
(TIPSE/DLW)

Westinghouse Electric Corp., Elmira, N. Y.
HIGH TEMPERATURE MATCH-BOX ELECTRON TUBES.
Quarterly progress rept. no. 2, 1 May-31 July 61 on Phase 2,
by Joel M. Fink. 31 July 61, 14p. incl. illus.
(Rept. no. 8)
(Contract DA 36-039-sc-87404; Continuation of Contract DA 36-039-sc-75076)
Unclassified report

No automatic release to Foreign Nationals.

DESCRIPTORS: *Electron tubes, *Triodes, Glass seals, *Glass, Cathodes (Electron tubes), Metal seals, Oxide cathodes, Materials, Structures, Test equipment, *High temperature research, Design, Tests.

Open-ended Terms: Pyroceram, Nucleated glass.

Investigations were continued on the use of pyroceram as a universal high temperature (500 C) tube envelope material. The cathode structure was modified to provide better thermal efficiencies and more uniformity from tube to tube. Some difficulties were encountered in sealing the tubes without ruining the cathode emissivity but it is believed that this can be overcome in a newly built sealing and exhaust position. Only preliminary trials were made on this equipment. (Author)

AD-263 640 9

AD-263 722 Div. 14, 7 U
(TIPSN/EET)

Burgess Battery Co., Freeport, Ill.
PLASTIC SEALS.
Final rept.,
by C. J. Broughton. 28 Apr 61, 23p. incl. illus.
(Contract DA 36-039-sc-72758)
Unclassified report

For reference only at each of the ASTIA Offices. This report cannot be satisfactorily reproduced; ASTIA does not furnish copies.

DESCRIPTORS: Dry cells, Seals, *Plastic seals, Thermosetting resins, Ethyl cellulose, Production, Manufacturing methods, Automation, Machines.

A summary is presented of efforts to produce a plastic sealing material suitable for mechanical sealing dry battery cells. Both thermo setting and thermo plastic compounds were investigated. Investigation and experimentation proved that the thermo plastic compound ethylcellulose was found to be effective. The material is identified by Dow Chemical as Dow PG-N60-I-5168-Flow - no. 2 formula no. 860. Production equipment and special tooling was used to produce pilot run quantities of 'N', 'A', and 'B' cells using ethylcellulose seals. No difficulty was encountered in making the pilot run quantities of the dry cell seals or cell blocks. (Author)

AD-263 722 3 ref.

AD-263 990 Div. 26, 17 U
(TIPSN/BMW)

Battelle Memorial Inst., Columbus, Ohio.
INVESTIGATION OF HYDROSTATIC EXTRUSION.
Interim technical engineering rept. no. 1, 1 June-31 Aug 61,
by R. J. Fiorentino, A. M. Sabroff, and F. W. Boulger. Sep 61, 12p. incl. illus. (Interim rept. no. 61-7-916 vol. 1)
(Contract AF 33(600)43328)
(ASD TR 61-7-916) Unclassified report

DESCRIPTORS: (*Steel, *Titanium alloys, Aluminum alloys, Vanadium alloys, *Extrusion, Hydrostatic pressure, Feasibility studies, *Machine tools, Hydraulic presses, Design.) (High pressure research, Hydraulic presses, Hydraulic fluids, Dies, Hydraulic seals, Metal seals, Rubber seals.)

Tooling is being designed for hydrostatic extrusion of SAE 4340 steel and Ti-6Al-4V Ti alloys into T-sections which can be inscribed in a 1-in. circle. The tooling is expected to be capable of withstanding extrusion pressures up to 250,000 psi at temperatures up to 500 F. Initial extrusion tests with the equipment will be made with materials such as lead and aluminum which can be extruded at lower pressures. Effective seals must be maintained at the die-liner and ram-liner interfaces. The seal design being considered employs variations of Bridgman's principle of unsupported area. Water will probably be the liquid medium used for the initial tests. Fluid for extrusion temperatures near 500 F will be selected on the basis of their stability at both temperature and high pressure. (Author)

AD-263 990 9

AD-264 762L Div. 7, 27 U
(TIPSN/BMW)

Aeronautical Engine Lab., Naval Air Material Center, Philadelphia, Pa.
QUALIFICATION TESTS OF GENERAL ELECTRIC MODEL 2CM27OD3 ELECTRIC STARTERS, BUAER MODEL E-5.
Final rept.,
by J. R. Cipparrone. 5 Oct 61, 1v. incl. illus. tables (Rept. no. NAMC-AEL-1691)
Unclassified report

Notice: Only Government Agencies may request from ASTIA.

DESCRIPTORS: (*Airplane engine starters, Power supplies, Clutches, Seals, Armatures, Electric motors.) (Tests, Helicopter engines, Turbopropeller jet engines.)

Open-ended Terms: T58 engines.

The results of qualification tests of General Electric Co. model 2CM27OD3 electric starters designed for use on General Electric model T58 turbo-shaft engines are discussed. The model 2CM27OD3 starter incorporated modifications which were designed to correct deficiencies which existed in model 2CM27OD2 starters. The results of tests established that the starter meets all of the performance and durability requirements specified for a BUAER type E-5 starter in Amendment 2 of Military Specification MIL-S-7780B(Aer) of 25 June 1958. However, suitable modifications to permit disassembly of the clutch are considered mandatory; in addition, improvement in the stability of the clutch assembly is considered desirable. (Author)

AD-264 762L 9

AD-265 170 Div. 7 U
(TIPSN/EET)

Gulton Industries, Inc., Metuchen, N. J.
THE TESTING OF HERMETIC SEALS AND SEPARATORS, POSITIVE PLATE STUDIES, AND INVESTIGATIONS ON THE UNIFORMITY OF SEALED NICKEL-CADMIUM CELLS.
Quarterly progress rept. no. 4, 1 Apr-30 June 61 on Investigations Leading to the Development of Improved Sealed Nickel-Cadmium Batteries,
by N. N. Beiger. 30 June 61, 56p. incl. illus. tables.
(Contract DA 36-039-sc-85390, Proj. 3618-03-001)
Unclassified report

No automatic release to Foreign Nationals.

DESCRIPTORS: (*Seals, *Alkaline cells, *Battery separators.) (Ceramic materials, Metal seals, Soldering alloys, Design, Manufacturing methods, Tests.) (Electrodes, Anodes (Electrolytic cell), Nickel, Cadmium, *Storage batteries.)

Ceramic-to-metal seals are being made using a batch process using two furnaces. Plate formation in the pilot plant has been made automatic. Additional methods of making the cover to can closure are being investigated. Separator tests on polypropylene from American Felt and Pellon's PVC and polypropylene at 125 and -40 F were completed. The separators which seemed most promising were tested for strength and resistance after storage in electrolyte at 165 F. New cells were fabricated to improve uniformity between cells. Pressure and voltage characteristics were determined. Additional experiments on the oxidation of Cd were made. (Author)

AD-265 170 9

AD-265 360 Div. 8 U
(TIPSE/MD) OTS price $8.10

Westinghouse Electric Corp., Pittsburgh, Pa.
HEAT DIODE CONVERTER.
Interim scientific rept. no. 1, 1 May-31 July 61
20 Aug 61, 86p. incl. illus.
(Contract AF 33(616)8262)
Unclassified report

DESCRIPTORS: (*Thermoelectricity, Electric power production, Space flight.) (Diodes, Seals, Life expectancy.) (Life expectancy, Space charges, *Thermionic emission, *Diodes, Voltage, Distribution, Cesium.) (Electrons, Scattering, Cesium, Atoms.) (Refractory materials, Metals, Cesium, Vapors, Electrons, Thermionic emission, Tantalum, Molybdenum.)

Contents:
emission of cesium coated refractory metals
Scattering of electrons by Cs atoms
Space charge limited currents in thermionic diodes at high cesium pressure
Potential distributions in high pressure cesium thermionic diodes
Thermionic diodes, carnot efficiency, and anomalous Richardson constants
Theoretical efficiencies of thermionic diodes
Life testing of seals for thermionic diodes

AD-265 360 9 N1

AD-265 443 Div. 14 1
(TIPSM/SEH)

Boeing Airplane Co., Seattle, Wash.
DESIGN HANDBOOK FOR O-RINGS AND SIMILAR ELASTIC SEALS.
Rept. for June 59–Jan 61 on Materials Applications.
by William R. Walker. Apr 61, 105p. incl. illus. tables, 24 refs.
(Contract AF 33(616)5722)
(WADC TR 59-428, pt. 2)
Unclassified report

No automatic release to Foreign Nationals.

DESCRIPTORS: (*Seals, *Fuel seals, *Hydraulic seals, *Plastic seals, *Rubber seals, *Vacuum seals, Gaskets, High temperature research, Low temperature research, Cryogenics, Storage, Aging, Dynamics, Physical properties, Mechanical properties, Impact shock, Sensitivity, Design.) (Hydraulic systems, Pneumatic systems.) (Materials, Elastomers, Polymers, Synthetic rubber, Silicones, Fluorides.) *Handbook.

Open-ended Terms: Compatibility, Teflon.

Data is presented on the mechanism of O-ring sealing in hydraulic and pneumatic systems utilizing static and dynamic type seals at temperatures exceeding 275 F and static applications at cryogenic temperatures. Fuel system seals are discussed in relation to material compatibility with hydrocarbon fuels and propellants. The effects of vibration, pressure cycling, seal materials and fluids on the operational efficiency of seals and back-up rings also are discussed. The limitations of organic seals and the design of vacuum seals are presented. Physical properties of fluorinated rubber elastomers at high temperatures are given. Aging and procedures for controlling aging are discussed in terms of storage. (Author)

AD-265 443 9

AD-265 555 Div. 14, 26, 31 U
(TIPTB/CCH) OTS price $2.60

Materials Testing Lab., Portsmouth Naval Shipyard, N. H.
DEVELOPMENT OF SYNTHETIC RUBBER MATERIAL FOR GASKETS AND PACKINGS IN DEEP SUBMERGENCE SUBMARINES.
Progress rept. no. 2,
by Billy H. Swift. 25 Oct 61, 21p. incl. illus.
(Rept. no. RD61-TOM/a)
(Proj. SF-013-13-01)
Unclassified report

DESCRIPTORS: (Design and Tests of *Gaskets, *Rubber gaskets, *Synthetic rubber, Nitrile rubber, Elastomers in Submarines.) (Test methods, Pressure, Underwater.)

A second phase in the development of deep submergence gasket material has been completed. Three rubber stocks, selected on the basis of tests described in Progress Report No. 1, have now been evaluated in a high-pressure test tank. One of the stocks performed satisfactorily in all aspects, but may be too high in hardness for certain applications. A third phase of the project is nearly complete. A group of seven stocks in the 60-80 durometer hardness range are being tested in the Laboratory, and current data indicate that at least two of them should be evaluated in the pressure test tank as candidate materials. (Author)

AD-265 555 9 N2

AD-265 582 Div. 20 U
(TIPTM/MRW) OTS price $2.60

Alco Products, Inc., Schenectady, N. Y.
INSPECTION OF SM-1 CONTROL ROD SEAL COMPONENTS.
Final rept. on Investigation of SM-1 Control Rod Seal Failure,
by R. A. Chittum, R. N. Saccocio and others. 16 June 61, 16p. incl. illus. tables (AP Note no. 362)
(Contract DA 44-192-eng-17)
(Rept. no. 05B 15, rev. 1) Unclassified report

DESCRIPTORS: (Nuclear power plants, Research reactors, Liquid cooled, *Control rods, Neutron absorbers, Drive shafts, Seals, *Metal seals, Failure (Mechanics) of Chromium plating on Stainless steel, Shafts, Corrosion, Microscopy, Seals, Chemical analysis, Radiographical analysis.) (Seals, Coolants, Water, Pressure, Tests.)

Identifiers: SM-1 reactors.

A laboratory test was completed to check leakage rates of a control rod drive seal assembly (shaft, seal rings and diaphragms) which had been in operation in the SM-1 since its original startup. The testing was performed at simulated reactor operating pressure (1200 psi) and room temperature to determine the leakage rate with the seal correctly assembled and with it assembled as it had been during reactor operation i.e., the first diaphragm and seal ring turned around on the seal shaft. No measurable wear was found when the components were measured. Micrographical examination of the Cr plate on a seal shaft removed from the SM-1 showed considerable pitting, corrosion and erosion under and adjacent to some of the stellite ring areas. The most likely cause of removal of the Cr plate was the poor quality plating and inadequate preplate cleaning which resulted in non-metallic inclusions in the material. (Author)

AD-265 582 9 N2

AD-266 158 Div. 8, 14, 26, 17 U
(TIPTE/NTM) OTS price $2.60

Sperry Gyroscope Co., Great Neck, N. Y.
CERAMIC-METAL SEALS FOR HIGH-POWER TUBES.
Technical note no. 1, 10 Apr-7 July 61,
by S. S. Cole, Jr. and J. E. McLinden. July 61, 18p. illus. (Rept. no. NA-8250-8268-1)
(Contract AF 30(602)2371, Proj. 5573)
(RADC TN 61-217) Unclassified report

DESCRIPTORS: (Seals, *Metal seals, *Ceramic materials, *Electron tubes.) (Materials, Aluminum compounds, Ceramic materials, Oxides, Rupture, Tensile properties, Heat treatment, Bonding, Metals, Stresses, Reliability.)

The ceramic-to-metal seal technology study is a continuation study in which 200 metallizing compositions were examined. Ceramic specimens for tensile and modulus of rupture tests were ordered, and equipment for spray and roller-coat metallizing is being fabricated. The experimental sintering furnace and the Baldwin tester are being modified. A study of the variables which effect seal strength was planned. These variables include method of application; metallizing particle size, thickness, and composition; sintering temperature, time, and rate; impurities; furnace dew point and atmosphere; and type of plate. The study of leak paths was initiated using a dye-impregnation technique. (Author)

AD-266 158 4 N3

AD-266 246 Div. 8, 14, 26 U
(TIPTE/NTM) OTS price $2.60

Knights, James, Co., Sandwich, Ill.
GLASS HOLDER CRYSTAL UNITS.
Quarterly progress rept. nos. 5 and 6, Aug 60-July 61,
by C. W. Snyder. July 61, 25p. incl. illus.
(Contract DA 36-039-sc-81274)
Unclassified report

DESCRIPTORS: (*Glass seals, *Crystal holders, *Quartz crystals, Vacuum seals, Design, *Manufacturing methods.) (Seals, Industrial equipment, Industrial production.) Production.

Research was concerned with: a check-out of the production type glass-sealing machine; the determination of the effect on frequency and resistance of the crystal resulting from the glass-sealing operation; and attempts to produce a crystal design at 800 kc which when mounted in the HC-6/U glass holder would meet the performance specifications. Small batch runs of HC-6/U glass parts on the production sealing machine produced yields of approximately 75% free of cracks or excessive strains. It was determined that line voltage changes detuned the system sufficiently to interfere with the heat programming. The temperatures reached in sealing to provide seals sufficiently free of strains were found to be marginally high for silver plated 9 and 20 mc crystals but not too high if gold electrodes were used. (Author)

AD-266 246 10 N3

AD-266 351 Div. 7, 12 U
(TIPTM/EJC) OTS price $3.00

Delco-Remy Div., General Motors Crop., Anderson, Ind.
SILVER OXIDE-ZINC BATTERY PROGRAM.
Rept. 30 June-31 Oct 60,
by John Rhyne, Jr. May 61, 152p. incl. illus. tables, 223 refs.
(Contract AF 33(600)41600, Proj. NR 3145)
(WADD TR 61-36) Unclassified report

DESCRIPTORS: (*Storage batteries, *Electrolytic cells, *Wet cells, Design, Materials, Operation.) (Electrodes, Silver compounds, Oxides, Zinc.) (Electrolytes, Potassium compounds, Hydroxides.) (Electrochemistry, Gases, Pressure, Voltage, Electric currents.) (Containers, Packaging, Seals, Plastic seals, Membranes, Plastics.) (Spaceships, Satellite vehicles, Electric power production.) Bibliography.

A study was made of the Ag ion migration within the cell during cycling. Distribution of Ag in the various layers of separator was determined as a function of the number of cycles. A comprehensive study of the Zn electrode was partially completed. The effect of current density and certain addition materials was determined in terms of Zn particle size increase during cycling. Methods of controlling Zn penetration of the separator material were established. Other investigations included gas producing reactions at both the negative and positive plates, pressure control, effect of alloying elements on the voltage characteristics of the positive plate, production of a molded terminal seal resistant to KOH leakage. A sealed cell was constructed and cycled over a 20% capacity range for 590 cycles. The pressure change was from a 10 psi vacuum to a 9 psi positive pressure.

AD-266 351 11 N3

AD-266 502   Div. 8, 26   U
(TIPTM/DLM) OTS price $2.60

General Electric Co., Owensboro, Ky.
DEVELOPMENT OF AN IMPROVED JAN 6299.
Quarterly progress rept. no. 1, 1 May-31 July 61,
31 July 61, 21p.
(Contract DA 36-039-sc-85953)
Unclassified report

DESCRIPTORS: (*Electron tubes, Cathodes (Electron tubes). Ceramic materials, Seals, Silver solders, Aluminum compounds, Temperature, Coatings, Humidity, Reliability, Test methods, Processing, *Manufacturing methods, Production.)

Investigations are being made to obtain better performance and greater reliability from the JAN 6299 by evaluating and adapting the latest tube technological advancement. Seven projects are outlined which are designed to improve some specific part of the tube. Progress made on each of the projects is listed. Some expected problems are discussed with the steps planned for their solution. (Author)

AD-266 719   Div. 10, 26   U
(TIPTM/EJM)

Aerojet-General Corp., Sacramento, Calif.
DESIGN HANDBOOK FOR LIQUID FLUORINE GROUND HANDLING EQUIPMENT.
by L. E. White, P. S. Gakle and others. Dec 60,
1v. incl. illus. tables.
(Contract AF 33(616)6566, Proj. 8119)
(WADD TR 60-159)   Unclassified report

DESCRIPTORS: (*Liquefied gases, *Fluorine, Helium, Oxygen, Nitrogen, Liquid rocket propellants.) (Gases, Fluorine, Hydrogen compounds, Fluorides.) (Production, Handling, Transportation, Storage, Safety, Toxicity, Disposal.) (Chemical properties, Physical properties.) (Storage tanks, Fuel storage tanks, Pressure vessels, Containers, Pressure tanks, Propellant tanks, Pumps, Valves, Pipes, Seals, Instrumentation.) (Gas detectors, Chemical indicators.) (*Handbooks, Fluorine.) Materials, Cryogenics.

This handbook is a compilation of available design data and methods of operation representing present technical experience levels as it applies to F. It includes design and operational data for liquid He, liquid O, and liquid N systems in areas likely to be of use to a designer of a liquid F facility. It contains considerable design information to assist in scaling up current sizes of F and cryogenic systems to meet specific requirements. No optimum system is selected, though several F systems are discussed. Advantages and disadvantages of the components are discussed to enable designers to choose the best system for particular needs. Bases for comparison include cost, availability, length of operation, F loss and location. (Author)

AD-266 719   9

AD-266 768   Div. 20, 25   U
(TIPTF/MFA) OTS price $8.60

Foreign Tech. Div., Air Force Systems Command, Wright-Patterson Air Force Base, Ohio.
ATOMIC ENERGY (SELECTED ARTICLES).
Sep 61, 89p. incl. illus. tables (Trans. no. MCL-1208 of Atomnaya Energiya 10:pp. 343-346, 372-373, 377-392, 396, 403-404, 406-407, 412-415, 420-422, 1961)
Unclassified report

DESCRIPTORS: (*Atomic energy, Scientific reports.) (Translations, USSR.) (Fission, Nuclear energy to Electricity, Thermionic emission, Diodes, Cathodes, Plasma physics, *Research reactors.) (Isotopes, Uranium, Plutonium, *Alpha particles.) (*Statistical analysis, Distribution theory.) (*Monochromatic light, Measurement, Thermal radiation, *Neutron spectrum, Graphite moderated reactors.) (*Liquid metals, Heat transfer, Turbulent flow.) (*Shafts, Torque couplings, Power, Seals.) (*Gamma rays, Scattering, Aluminum.) *Radioactive isotopes, *Atomic energy engineering.

Contents:
Thermionic energy converters; Ternary fission of U-233, U-235, Pu-239, and Pu-241 Nuclei; Measurement of the kinetic characteristics of a reactor by a statistical p-Method; Distribution of the number of counts recorded by a neutron detector located in a reactor; Use of a monochromator to measure the thermal-neutron spectra of a nuclear reactor; Turbulent-flow heat transfer in liquid metals; Power losses and initial shaft torque in a seal of frozen sodium; Calculation of the passage of fast neutrons through cylindrical channels in biological shielding; Spectrum of scattered gamma-radiation; Monte-Carlo calculation of the passage of gamma-radiation from a plane Cs-137 source through aluminum under conditions of barrier geometry; Synthesis of uranium dodecaboride; A change in the activity of fission products of U-235 and Pu-239 with time; and News of science and technology.

AD-266 768   9   N3

AD-266 936   Div. 25, 14   U
(TIPTM/GEC)

Bendix Products Div., Bendix Corp., South Bend, Ind.
MATERIALS PROPERTY DATA.
Quarterly progress rept. no. 2 on Phase 1 and Monthly summary rept. no. 6 on Phase 2, 1 Apr-30 Sep 61,
by James W. Yates. Oct 61, 2 nos. in 1v. incl. tables.
(Contract AF 33(616)8086)
Unclassified report

DESCRIPTORS: (*Friction brakes, Friction, *Materials, High temperature research, Graphite, Silicon compounds, Dioxides, Tests, Tables.) (Friction, Materials for Bearings, Seals, Low temperature research, Magnesium compounds, Oxides, Flame spraying, Tests.) (Abrasive coatings, High temperature research, Flame spraying, Tests.) (Structures, Materials, High temperatur research, Tests, Tensile properties, Deformation, Low temperature research, Bearing, Seals.)

Identifiers: Composite materials.

There are three areas of coverage. The first area, High Temperature - High Friction, is the development in the aircraft brake field and is one-third complete in this report. The second area is that of Seals and Bearing, Low Temperature - Low Friction Materials. This data is complete. The third is Abradable Coatings, similar to seals and bearings, though quite different in mechanical properties. This data also is complete. (Author)

AD-266 936   4

AD-267 051   Div. 7   U
(TIPTM/RNM) OTS price $2.00

Inland Testing Labs., Dayton, Ohio.
ALKALINE BATTERY EVALUATION.
Rept. for 30 June 60-15 Apr 61,
by T. F. Luke, W. G. Ingling, and W. W. Clark.
Oct 61, 70p. incl. illus. tables.
(Contract AF 33(616)7529, Proj. 3145)
(ASD TR 61-236)   Unclassified report

DESCRIPTORS: (*Alkaline cells, Wet cells, Storage batteries for Spaceships, Satellite vehicles.) (Anodes (Electrolytic cells), Cadmium compounds, Hydroxides.) (Cathodes (Electrolytic cells), Nickel compounds, Hydroxides.) (Battery separators, Cellulose.) (Electric insulation, Ethylenes, Fluorides, Polymers, Nylon.) (Electrolytes, Potassium compounds, Hydroxides.) (Containers, Stainless steel, Magnesium.) (Seals, Rubber seals.) (Valves, Check valves.) (Design, Tests, Failure (Mechanics), Analysis.) (Cadmium, Nickel, Silver, Zinc.)

The report covers the first period of an alkaline battery applied research and failure analysis program to establish a broad base of battery test data for the design of electrical systems of future space vehicles. Another objective is to determine techniques and/or materials to prevent these failures while increasing the usable watt-hours-per-pound capability and cycle life of the battery. Cycle-life tests were conducted on 1 type of 12-amp-hr, sealed, Ni-Cd cell in 3 temperature environments and 4 depths of discharge. Initial results show that (1) cell cycle-life with shallow discharges is considerably longer than cycle-life at deep discharges, and (2) cycle-life is reduced by high ambient temperatures. The program will include as future work evaluation of Ag-Cd and Ag-Zn type cells. (Author)

AD-267 051   29   N3

AD-267 111   Div. 14   U
(TIPTM/TCG) OTS price $1.00

Physics Lab., Wright Air Development Div., Wright-Patterson Air Force Base, Ohio.
EVALUATION OF HUNA-N (NBR) GASKETS IN PROTOTYPE APPLICATIONS UNDER GAMMA FLUX.
Rept. for Sep 57-Jan 59, on The Chemistry and Physics of Materials,
by John A. Parker and Denver Hale. Feb 61, 31p. incl. illus. tables.
(Proj. no. 7360)
(WADD TR 59-661)   Unclassified report

DESCRIPTORS: (*Rubber gaskets, *Acrylonitriles, *Butadienes, Rubber, Chemical properties, Quantitative analysis, Seals, Sealing compounds under Gamma rays.) (Radiation effects on Physical properties and Bonding in Polymerization, Ferrocenes, Effectiveness in Reduction, *Radiation damage.) (Laboratory equipment, Measurement, Gas leaks, Gaskets.) Tensile properties, Flanges, Pressure.

Qualitative and quantitative sealability tests were evaluated to test the theoretical prediction of the stability of certain acrylonitrile-butadiene rubber stocks. Reasonable agreement between sealability coefficients was obtained from the seal tester and relative replacement times over a dose range from 2 x 10 to the 9th power to 2 x 10 to the 10th power ergs/g C (one roentgen = 87.7 ergs). Evaluation criteria for these compositions were established by application of the Bopp-Sisman relationship to observed changes in the elastic modulus previously reported. The effect of ferrocene as a potential anti-rad was investigated and found to improve the radiation resistance of these stocks. The apparatus described for determining sealability of gasket materials consist of a method of applying a load on the gasket, equipment for pressurizing one side of the gasket seal and a method for measuring gas leakage. (Author)

AD-267 111   4   N4

AD-267 121 Div. 26, 14, 31 U
(TISTM/EET) OTS price $1.10

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
DEVELOPMENT OF SEALS FOR HIGH PRESSURE AIR SYSTEMS IN SUBMARINES.
Progress rept. no. 5,
by R. D. Ford. 21 Sep 61, 10p. incl. illus.
(Rept. no. 28-8)
(Proj. no. S-F013-13-01)
Unclassified report

DESCRIPTORS: (*Rubber seals, *Pneumatic systems for Submarines, Effectiveness.) (Synthetic rubber, *Seals, Materials, Physical properties, Mechanical properties, High pressure research.) (Compressed air, High pressure compressors.)

The Laboratory was requested to develop a formulation for rubber seals to be used in the high pressure air systems of fleet-ballistic-missile submarines. The two best stocks reported previously were highly resistant to both the petroleum and phosphate-ester lubricants used in the air compressors. Seals made from both stocks had excellent resistance to swelling by high pressure air. O-rings prepared from the first stock, showed moderate backrinding when removed from hot molds. The second stock cured without backrinding. (Author)

AD-267 121 9 M4

AD-267 122 Div. 8 U
(TISTE/CDM)

Sperry Gyroscope Co., Great Neck, N. Y.
INVESTIGATION OF MICROWAVE WINDOW FAILURE MECHANISMS AND THEIR ELIMINATION.
Quarterly progress rept. no. 3, 1 Apr-1 July 61, by D. B. Churchill and E. W. Cheatham. July 61, 39p. incl. illus. (Rept. no. NA-8220-8235-3)
(Contract DA 36-039-sc-87309; continuation of Contract DA 36-039-sc-78314)
Unclassified report

Not releasable to Foreign Nationals except Great Britain and Canada.

DESCRIPTORS: (*Waveguide windows, *Microwave equipment, *Electron bombardment, Tests.) (Dielectrics, Materials, *Ceramic materials, Waveguides, Waveguide irises, Design.) (Sapphires, Aluminum compounds, Oxides, Seals, Radiation damage.)

High-energy electronic bombardment tests on sample windows of sintered alumina and crystalline synthetic sapphire were conducted. Both samples were bombarded for several hours with electron beams of energy levels up to 140 kv. The sapphire target showed no signs of damage; the alumina target was slightly eroded in the impact area, but no measurable crater was formed. In a second series of tests, each target was bombarded with an electron beam on one side and an unfocused ion beam on the other. Both targets were severely damaged on the electron beam side. Thin-disc S-band windows were tested at the 12-mega level using an SAS-199 tube as a driver. A 3-section window assembly was constructed, baked, and exhausted to a high vacuum. The window assembly performed satisfactorily in the first few hours of operation. Life-test data are being taken. The design criteria for thick-disc windows were studied. (Author)

AD-267 122 9

AD-267 145 Div. 25, 14 U
(TISTP/MFA)

Chance Vought Corp., Dallas, Tex.
COMPOSITE ELASTOMER-METAL O-RING SEALS.
Rept. for 6 Apr 60-31 May 61 on Exploratory Design and Prototype,
by L. E. Doughty. June 61, 36p. incl. illus. tables.
(Contract AF 33(616)7079, Proj. 7381)
(WADC TR 59-769, pt. 2)
Unclassified report

DESCRIPTORS: (Rubber seals, *Elastomers, Rubber gaskets, *Design.) (Tests, Materials, Physical properties, Shear stresses, Tensile properties, Hardness.) (Processing, Manufacturing methods.) (Experimental data, Photographic analysis, Tables.)

The objectives of this program were elastomer-metal composite O-ring seals with superior physical properties and evaluation of the performance of the prototype O-ring seals in adverse environments. A revised fabrication procedure, which improved the acceptability rate of composite O-ring seals, was developed to embed a metal spring concentrically in an elastomeric O-ring. Two adhesives to bond the elastomer to the spring core were found suitable. Hydraulic dynamic cycling tests were performed at 4000 psi and temperatures up to 500 F. Performance of composite O-rings installed on a test piston without back-up rings was not satisfactory for practical use at elevated temperatures and pressures. Performance of composite O-ring seals used with back-up rings was comparable to that of conventional O-ring seals. Because of the severe degradation of elastomeric properties at high temperatures, the dynamic sealing performance of elastomeric O-ring seals, conventional or composite, appears to be dependent on back-up ring performance. (Author)

AD-267 145 9

AD-267 501 Div. 14 U
(TISTM/TCG)

Boeing Co., Seattle, Wash.
DESIGN DATA FOR O-RINGS AND SIMILAR ELASTIC SEALS.
Rept. for 16 Feb-31 Dec 60 on Materials Applications,
by George E. Trepus, Robert S. Roper and others. May 61, 362p. incl. illus. tables, 17 refs.
(Contract AF 33(616)5722, Proj. 7381)
(WADC TR 56-272, pt. 6) Unclassified report

DESCRIPTORS: (*Seals, *Hydraulic seals, *Plastic seals, *Rubber seals, Elastomers, Silicones, Physical properties, Mechanical properties, Temperature, Tests, Cryogenics, Design.) (Thermal expansion, Tensile properties, Friction, Deformation, Stresses, Compression shock, Aging, Viscosity, Stability.) (High pressure research, Low temperature research.) Bibliography.

Identifiers: TEFLON.

A study was made to obtain and develop seal design information so that a handbook could be prepared concerning seals for extreme and/or unusual environments. A survey of current literature concerning seals and seal materials was conducted. Emphasis was placed on seals for extreme environments and on the aging of seals and its effects. These properties, including thermal expansion, compression modulus, compression set, frictional characteristics, and tensile strength, elongation, and modulus, tensile and compression stress relaxation were determined at temperatures from -425 to 600 F. Functional test data reported include information from static and dynamic pneumatic seal tests at temperatures from room temperature to 1000 F, hydraulic rod seal tests at temperatures from -65 to 800 F at pressures from 1000 to 4000 psig, and static and dynamic seal tests at cryogenic temperatures. (Author)

AD-267 501 9

1

AD-267 890 Div. 14, 20, 21 U
(TISTM/GEC) OTS price $21.?0

Radiation Effects Information Center, Columbus, Ohio.
THE EFFECT OF NUCLEAR RADIATION ON ELASTOMERIC AND PLASTIC COMPONENTS AND MATERIALS,
by R. W. King, N. J. Broadway and S. Palinchak. 1 Sep 61, 370p. incl. illus. tables, 205 refs.
(REIC rept. no. 21; Supersedes rept. nos. 3, AD-149 551; 3A, AD-210 759; 3-2A, AD-239 264; 9, AD-229 487; 13, AD-241 430 and memo. nos. 1, AD-207 043; 3, 8, 15, AD-225 904 and 17, AD-219 511)
(Contract AF 33(616)7375, Proj. 1448)
Unclassified report

DESCRIPTORS: (*Radiation effects, *Radiation damage, Organic materials, *Plastics, *Elastomers, *Polymers, Laminates, Nuclear propulsion.) (Seals, Gaskets, Hoses, Coatings, Electric insulation, Optical materials, Optical plastics, Structural shells.) (Theory, Tests, Controlled atmosphere, Mechanical properties, Physical properties, High temperature research.) *Bibliography.

Organic materials are susceptible to damage from all types of nuclear radiation. Consequently, plastics and particularly elastomers, present serious problems in connection with the development of components and systems for nuclear-powered vehicles. In addition to being susceptible to damage by radiation, many of these rubber and plastic materials are adversely affected by environmental conditions, such as extreme temperatures (-100 to +500 F), vacuum, oxidizing atmospheres (ozone), and various types of fuels, lubricants, and hydraulic fluids. Although there are a number of organic materials which have radiation resistance in the range required, they are not useful in the construction of many components and systems because they are lacking in some other needed property. For example, polystyrene has very good radiation resistance but low strength and heat resistance. Therefore, the major problem is to develop materials which will resist radiation and which are also satisfactory in the environments mentioned above. (Author)

AD-267 890 9 M4

AD-268 381 Div. 14 U
(TISTM/TCG)

Chance Vought Corp., Dallas, Tex.
PHYSICAL PROPERTIES OF SOME ENGINEERING MATERIALS. UNPUBLISHED DATA FROM COMPANY SPONSORED PROGRAMS.
Quarterly rept. no. 3, vol. 1 on Phase 1, 1 Sep-30 Nov 61,
by J. H. Pimm and J. J. Peterson. 15 Dec 61, 90p. incl. illus. tables (Rept. no. AER-E1R-754)
(Contract AF 33(616)7986)
Unclassified report

DESCRIPTORS: (Materials, Physical properties, Mechanical properties, Test methods.) (*Glass textiles, *Resins, Laminates.) (*Hydraulic fluids, Viscosity, Hydrolysis, Corrosion inhibition.) (*Magnesium alloys, Aluminum alloys, Corrosion.) (Rubber gaskets, Insulating materials, Stability.) (*Sealing compounds, Stresses, High temperature research.) *Rubber.

Contents:
Heat resistance of reduced resin laminated cones reinforced with E type fiberglass
OS-45-1 hydraulic fluid: lubricity tests
Evaluation of GE 81717 hydraulic fluid for system temperature ranges of 0 to 500 F and 0 to 700 F
Evaluation of Oronite 8515 hydraulic fluid per piloted aircraft spectrum A
Corrosion test on Mg fittings for fuel system applications
Evaluation of physical properties of O-ring compounds in OS-45-1 hydraulic fluid at 400 F and in dry air at 350 F
Evaluation of rubber compounds conforming to specification MIL-P-18017A
Evaluation of high-temperature sealants

AD-268 381 9

AD-268 416 Div. 8, 14, 26 U
(TISTE/NTN) OTS price $1.60

Chatham Electronics Div., Tung-Sol Electric, Inc., Livingston, N. J.
PRODUCTION ENGINEERING MEASURE FOR POLYOPTIC SEALING OF HYDROGEN THYRATRON TUBES.
Quarterly progress rept. no. 9, 1 July-30 Sep 61 by Ward W. Watrous. 27 Oct 61, 14p. incl. illus.
(Contract DA 36-039-sc-81289)
Unclassified report

DESCRIPTORS: (*Thyratrons, *Glass seals, Hydrogen, Sealing compounds, Production, *Manufacturing methods.) (*Miniature electron tubes, Electron tubes, Seals, Enamel coatings, Electrical properties, Mechanical properties, Temperature, Life expectancy, Tests.)

Research was continued on the polyoptic sealing of hydrogen thyratron tubes. Life tests were continued. All tubes required to have been made on the automatic exhaust machine have either completed life test or are being tested. The present results show no essential difference in survival rates between the 2 types of seals. No further work on variations in sealing techniques was done but consideration is given to an extension of the high temperature bake process investigation. Initial work was begun in determining the proper techniques for polyoptic sealing on trolley exhaust. (Author)

AD-268 416 9 N5

AD-268 419 Div. 26, 27, 12 U
(TISTA/VGM)

Component Evaluation Labs., South El Monte, Calif.
AIR FORCE EVALUATION TESTING OF ATLAS ''D'' COMPONENTS TWO PROPELLANT TANK PNEUMATIC CHECK VALVES GD/A PART NUMBER 27-08114-1 B. H. HADLEY PART NUMBER 10588-1 SERIAL NUMBERS 56-399. Oct 61, 28p. incl. illus. tables (Test rept. no. 2015)
(Contract AF 04(647)703)
(AFBSD TR 61-38)
Unclassified report

DESCRIPTORS: (*Check valves, Pneumatic devices, Plastic seals for Propellant tanks of Guided missiles, Surface to surface, Tests.) (Ethylenes, Fluorides, Polymers.)

Identifiers: Teflon, Atlas-D.

The performance characteristics of two propellant tank pneumatic check valves were evaluated. During the test program which is described in detail, each unit failed as described below: (a) After 17,000 cycles of a 20,000 cycle life test, the teflon seat in Unit 1 cold flowed to the extent that the poppet would no longer seal in the check flow direction. (b) After passing the 20,000 cycle life test, Unit 2 failed after 100 cycles of the contaminated endurence test. The contamination, introduced during testing, resulted in failure of the teflon seat to seal against the poppet in the check flow direction. (Author)

AD-268 419 2

AD-268 420 Div. 12 U
(TISTW/EET)

Component Evaluation Labs., South El Monte, Calif.
AIR FORCE EVALUATION TESTING OF ATLAS D COMPONENTS TWO HYDRAULIC RELIEF VALVES GD/A PART NUMBER 27-08588-1 VICKERS INC. MODEL EA 50094 SERIAL NUMBERS C-29064 AND C-14975. Nov 61, 23p. incl. illus. (Test rept. no. 2001-2)
(Contract AF 04(647)703)
(AFBSD TR 61-69) Unclassified report

DESCRIPTORS: (*Guided missiles, Surface to surface, *Hydraulic systems, *Hydraulic valves, Seals, Hydraulic sealing, Erosion, Reliability, Tests, *Safety valves.)

Identifiers: Atlas.

The performance characteristics of 2 hydraulic relief valves currently in operational use in D series Atlas missiles, are presented. Two problem areas were discovered during testing. First, due to improperly applied torque, the valve poppet housing assembly became loose during testing and, subsequently, excessive internal leakage occurred. Second, during the contaminated portion of the Life Test, undue erosion of the poppet sealing surface caused excessive internal leakage when the valve was in the closed condition. A detailed discussion of the test program along with the test equipment employed in the performance of the program, is included. (Author)

AD-268 420 2

AD-268 426 Div. 25, 30 U
(TISTM/GEC) OTS price $2.60

Engineering Supervision Co., New York.
INVESTIGATION OF STRESS DISTRIBUTION AND STRENGTH CHARACTERISTICS OF LOAD MAGNIFICATION ANVILS FOR ULTRA HIGH PRESSURE APPARATUS.
Quarterly scientific rept. no. 2, 1 Apr-30 June 61 on Photoelastic Investigation of Axially Compressed Cone Frustums,
by Herbert Becker. 30 June 61, 19p. incl. illus
(Technical rept. no. SM 61-10)
(Contract AF 19(604)7438; In cooperation with New York U. Coll. of Engineering)
(AFCRL-727) Unclassified report

DESCRIPTORS: (*High pressure research, Laboratory equipment, Installation, Gaskets, Load distribution, Photoelasticity, Stresses, Shear stresses, Pressure, Temperature, Fracture (Mechanics), Tests, Theory.)

Identifiers: Anvils.

Photoelastic studies were made of the stresses induced in conical frustums by uniform pressure applied to the small bases. One cone was investigated for uniform pressure applied to the small base and to a region of the conical surface adjacent to the small base. The data are reported in the form tau sub max/p where tau sub max is the principal shear stress at a point and p is the nominal uniform pressure applied to the cone base (axial force/base area). In addition, a room temperature test was performed on one cone to determine the fracture mode. The experimental method and results of these investigations are described and contain comparison with theory. (Author)

AD-268 426 9 N5

AD-268 507 Div. 8, 14 U
(TISTM/TCG) OTS price $3.60

Sperry Gyroscope Co., Great Neck, N. Y.
HIGH AVERAGE POWER R-F-WINDOW STUDY.
Quarterly memo. no. 3, 26 July-26 Oct 61, by D. B. Churchill and E. W. Cheatham. Nov 61, 16p. incl. illus. (Rept. no. NA-8220-8261-3)
(Contract AF 30(602)2428)
Unclassified report

DESCRIPTORS: (*Waveguide windows, Dielectrics, Ceramic materials, Single crystals, Synthetic gems, Sapphires, Aluminum compounds, Beryllium compounds, Oxides, Wave transmission, Electron bombardment, Seals, Design, Tests.) (Cavity resonators, Waveguide couplers, Design.) Microwave equipment, Broadband, Electron tubes, Waveguides.

The arc-detector and protective circuit was modified to achieve a faster response time and positive operation in the event of a waveguide arc; a special test system was developed to measure the response time. A system of automatic attenuation was developed to reduce the high-power drive signal in the presence of a severe load mismatch. A heterodyne arrangement was devised to permit the low-power c-w oscillator signal to be used as a fixed-frequency marker on the swept-frequency response pattern of the ring resonator. The first test window structure was tested at high levels of c-w power in the ring resonator. The structure consisted of a compression-sealed disc of sintered BeO ceramic in a cylindrical waveguide joined by conical mode transducers. The window transmitted 75 kw of power without evidence of breakdown or change of electrical characteristics. Internal arcing occurred in the ring resonator at the 75 kw level. A compression sealed double-disc window structure using thin sapphire discs was developed and cold tested. Some modifications were made to the BeO-disc water load to improve its bandwidth, and to increase the rate of water flow. (Author)

AD-268 507 9 N5

AD-269 034 Div. 7, 25 U
(TISTE/CHJ) OTS price $1.60

General Electric Co., Utica, N. Y.
RESEARCH AND DEVELOPMENT ON THERMOCOUPLE ENERGY CONVERTERS (PHASE II).
Quarterly progress rept. no. 2, 12 July-1 Oct 61, on phase 2,
by W. J. van der Grinten. 1 Nov 61, 13p. incl. illus.
(Contract DA 30-069-501-ORD-3274, Proj. TN2-8106)
Unclassified report

DESCRIPTORS: (*Thermocouples, *Thermoelectricity, *Power supplies, Lead compounds, Tellurides, Manufacturing methods, Design.) (Generators, Seals, Materials.)

Efforts are being made to develop techniques for fabricating, in quantity, good quality n- and p-type thermoelectric lead telluride cells and to assemble these cells to form 2 thermoelectric devices which will meet the following requirements: (a) each of the 2 thermocouple energy converters will consist of 9 couples and will have an output of 7.9 watts/cubic inch or better at a minimum load current of 20 amp when a temperature gradient of at most 600 K/cm exists across the thermoelectric material; (b) the converters will be capable of operation with the hot side of the thermoelectric material near its melting point (917 C) and the performance indicated should be attainable at a hot side temperature of at least 1000 K; and (c) the devices should be operable for a minimum of one hour. Current effort has resulted in perfecting the casting technique so that n- and p-type cells can be manufactured in quantity. Also, the design and fabrication of the complete cell assembly including a hermetic seal at the cold junction side, having a maximum temperature capability of 780 C, was realized. (Author)

AD-269 034 9 N6

AD-269 310 Div. 14, 26, 16 U
(TISTM/MJM)

Reaction Motors Div., Thiokol Chemical Corp., Denville, N. J.
ELASTOMERIC AND COMPLIANT MATERIALS FOR CONTACT WITH LIQUID ROCKET FUELS AND OXIDIZERS.
Rept. for May 60-May 61 on Nonmetallic and Composite Materials,
by Joseph Green and Nathan B. Levine. Sep 61, 59p. incl. illus. tables, 36 refs.
(Contract AF 33(616)-7227, Proj. 7340)
(ASD TR 61-76, pt. 1) Unclassified report

DESCRIPTORS: (*Liquid rocket propellants, Rocket oxidizers, *Hydrazines, Methylhydrazines, Nitrogen compounds, *Tetroxides, *Chlorine compounds, *Fluorides, Seals, *Rubber seals, Coatings.) (Polymers, *Elastomers, *Synthetic rubber, Vulcanizates, Silicones, Laminates, *Plastic coatings, Ethylenes, Fluorides, Resins, *Metal coatings, Films, Aluminum, Gold.) (Manufacturing methods, Test methods, Test equipment.)

Identifiers: Composite materials, Compatibility.

Various techniques were employed to provide elastomeric composites suitable for use in contact with hydrazine type fuels, N2O4, and ClF3. Through the use of compounding studies and investigations of resinous and metallic coating systems several elastomeric materials were recommended for field testing in contact with N2H4 type fuels and N2O4. A polyethylene coating was applied to an elastomeric O-ring. The elastomer base was protected from NO2 and N2O4 for 24 hr by this coating. Preliminary investigations were initiated for development of elastomeric composites which are compatible with ClF3. (Author)

AD-269 310 9

AD-269 404 Div. 8, 26 U
(TISTM/DLS) OTS price $3.60

General Electric Co., Owensboro, Ky.
DEVELOPMENT OF AN IMPROVED JAN 6299.
Quarterly progress rept. no. 2, 1 Aug-31 Oct 61, by Francis A. Norris. 31 Oct 61, 29p. illus.
(Contract DA 36-039-sc-85953)
Unclassified report

DESCRIPTORS: (*Electron tubes, Seals, Coatings, Design, Processing, *Manufacturing methods, Life expectancy, Humidity, Tests.)

The study to evaluate certain areas of advanced tube manufacturing techniques and to apply them where feasible to the JAN 6299 was continued. Tables are presented listing eccentricity data and humidity test results for present production tubes. A description of the proposed redesign of three seals from lead to hard solder is included. Life results of the cathode spray density evaluation are summarized. (Author)

AD-269 783 Div. 3, 1 U
(TISTM/RD)

Armament Research and Development Establishment (Gt. Brit.).
SEALING OF CHARGED FLUORESCEIN CONTAINERS USING RADIO FREQUENCY SEALING EQUIPMENT,
by T. I. J. Toler, W. N. Peacock, and N. N. English. Dec 61, 13p. illus. (ARDE memo. no. (MX) 65/61)
Unclassified report

DESCRIPTORS: (Sealing compounds, Organic compounds, *Containers, *Seals, *Pyrotechnics, Naval equipment.) (Containers, Polymers, Ethylenes.) (Test methods, Heating by Radio-frequency.) Great Britain.

A polyethylene container of fluorescein incorporated in a Naval pyrotechnic store was occasionally found to leak at the screwed filling plug. In order to remedy this fault an investigation was undertaken into the technique of sealing by radio frequency heating after liquid charging. A description is given of the equipment and components used to achieve a satisfactory seal. (Author)

AD-269 783

AD-270 222 Div. 8 U
(TISTE/NTN)

Power Information Center, U. of Pennsylvania, Philadelphia.
PROCEEDINGS OF THE ROUND TABLE DISCUSSION ON CATHODE EMISSION INVESTIGATIONS AND EXPERIMENTAL TECHNIQUES FOR FABRICATING AND OPERATING THERMIONIC CELLS, 1 AND 2 JUNE 1961.
30 June 61, 201p. incl. illus. table, 20 refs.
(PIC-ELE-TI-1/3)
Unclassified report

No automatic release to Foreign Nationals.

DESCRIPTORS: (*Thermionic emission, Atoms, Ions, Electrons, Metals, Seals, Insulating materials.) (Diodes, Cathodes, Crystals, Anodes, Alkali metals, Vapors.) (Refractory materials, Metals, Cesium, Vapors.)

Contents:
Seal and insulator problems in thermionic energy converters
Carbide cathode performance
Arc mode vapor thermionic converter
Thermionic emission of refractory metals in cesium vapor
Saturated emission or ion-limited
Measurements of thermionic emission from single crystal cathodes in alkali vapors
Basic experimental studies of the close-spaced thermionic diode
Experimental techniques of fabricating and operating in pile thermionic cells
Investigation of refractory compounds as thermionic emitters
A new approach to thermionic energy conversion
Techniques of fabricating a thermionic cell
Fabrication and operation of in pile cesium plasma cells
Can some standardization of thermionic terms and concepts be achieved now
Thermionic emission and surface equilibration
Presentation on cesium compatibility
Discussion of experimental I-V curves
Fabrication of the type E solar thermionic diode
A frangible all metal cesium capsule
Presentation of thermionic power conversion

AD-270 222 9

AD-270 582 Div. 8, 14, 26 U
(TISTE/CBJ) OTS price $2.00

Eitel-McCullough, Inc., San Bruno, Calif.
CERAMIC RECEIVING TUBE REFINEMENT PROGRAM.
Rept. for 16 June 58-31 Dec 60,
by Bruce E. Barnaby. Nov 61, 65p. incl. illus. tables, 10 refs.
(Contract AF 33(600)36699, Proj. 4156)
(ASD TR 61-268) Unclassified report

DESCRIPTORS: (*Electron tubes, Twin triodes, Triodes, Pentodes, *Ceramic materials, Processing, Production, *Manufacturing methods, High temperature research, Life expectancy, Reliability.) (Oxide cathodes, *Cathodes (Electron tubes), *Thermionic emission, Materials.) (Electron tubes, Degasification, Gases, Reduction, Materials.) (*Electrodes, Materials.) (Electron tubes, Metal seals, Ceramic materials, Vacuum seals.)

Identifiers: Getters.

Some of the materials and processing problems associated with the development and operation of ceramic vacuum tubes were investigated experimentally. The tubes were designed for operation where superior performance is required at ambient temperatures as high as 500 C. The following problems were investigated: cathode emission, vacuum seal, grid material, getter material and ceramic outgassing. Loss of emission from the oxide cathode is not a direct cause of tube failure as long as the cathode is processed carefully. A hot oxidizing atmosphere causes extensive oxidation of the seal elements. At ambient temperatures in excess of 400 C, seals begin to leak after a few hundred hours, but this failure cannot be attributed to oxidation alone. Gold grids inhibit adverse grid emission when exposed to hot oxide cathodes for grid temperatures up to 600 C. Cerālloy 400 operated at 500 C is an effective getter for the gases encountered in ceramic tubes. Heat treatment of ceramic under high vacuum causes the evolution of carbon monoxide, hydrogen, methane and water vapor. Heating to 1000 C effectively reduces the outgassing rate of the ceramic. (Author)

AD-270 582 29 N1

AD-270 746 Div. 26, 14, 30 U
(TISTM/BRW) OTS price $1.60

Rubber Lab., Mare Island Naval Shipyard, Vallejo Calif.
SUITABILITY OF VITON B O-RINGS FOR USE IN 3000 PSI HYDRAULIC SYSTEMS CONTAINING PETROLEUM BASE FLUID OR CELLULUBE 220.
Progress rept. no. 3,
by R. D. Ford. 29 Nov 61, 8p. illus. table.
(Rept. no. 92-11)
(Proj. S-F013-13-01)
Unclassified report

DESCRIPTORS: (*Hydraulic seals, Oil seals, *Rubber seals, Synthetic rubber, Phosphates, Organic compounds, Petroleum, Ethylenes, Fluorides, Polymers, Cyclohexanes, Effectiveness.) (Test methods, *Hydraulic systems, Hydraulic fluids.)

Identifiers: Viton, Teflon.

Efforts were made to develop a performance test for Viton O-rings to be used in 3000 psi hydraulic service. The service fluid may be either petroleum base or Cellulube 220. The test apparatus consisted of a piston with two O-rings which reciprocates 100 times/min in a vertical cylinder of 2.63-in. internal diameter. The piston travel is 3-1/2 in. The cylinder is held at 160 F. Teflon back-up rings are used on the downstream side of the O-rings. The pressure-on cycle lasts for 9 min; the pressure-off cycle lasts for 1 min. Leakage at both ends of the piston is collected. Difficulty was experienced in obtaining reproducible results with this equipment when the criterion was time required for a definite leakage (20 cc) to occur at 3000 psi fluid pressure. The test results show that Viton O-rings seal Cellulube 220 satisfactorily under these conditions and are not damaged by this treatment. The investigation is being continued with the criterion changed to the number of cycles before the leakage in a 24-hr period exceeds 150 cc for either O-ring. (Author)

AD-270 746 9 N1

AD-270 870   Div. 7   U
(TISTM/EJM) OTS price $1.50

Gulton Industries, Inc., Metuchen, N. J.
NICKEL-CADMIUM BATTERIES,
by Guy Rampel, R. Dagnall and others. Jan 61,
51p. incl. illus. tables.
(Contract AF 33(600)41670, Proj. 3145)
(WADD TR 61-34)   Unclassified report

DESCRIPTORS: (*Alkaline cells, Nickel electrodes, Cadmium, *Storage batteries, Electrolytic cells, Manufacturing methods, Electrodes Impregnation, Nickel compounds, Cadmium compounds, Hydroxides, Sealing compounds, Seals, Ceramic materials, Aluminum alloys, Design, Test methods.) (Electrochemistry, Recombination reactions, Oxygen.) (Satellite vehicles, Power supplies.)

Research and development leading to improved, long life, reliable, high watt-hour-per pound, sealed nickel-cadmium batteries are reported. Fundamental studies on electrode mechanisms were treated theoretically. Improved techniques were developed to impregnate active material into electrodes so as to gain increased output. Studies were made of the parameters involved in assembling sealed cells so as to achieve high overcharge capability without sacrificing capacity. Development progressed on techniques to produce ceramic-to-metal terminal seals; mechanical configurations and designs of prototypes were worked out. Five possible designs of sealed cells to meet the desired performance requirements are discussed. (Author)

AD-270 870   9   N1

AD-271 561   Div. 26, 8, 25   U
(TISTE/REB) OTS price $1.60

Knights, James, Co., Sandwich, Ill.
GLASS ENCLOSED MINIATURE CRYSTAL UNITS
CR(XN-29/U).
Rept. no. 2, Feb-Oct 61.
Oct 61, 16p. incl. illus.
(Contract DA 36-039-sc-81298)
Unclassified report

DESCRIPTORS: (*Crystal oscillators, Quartz crystals, Glass seals, Vacuum seals, *Seals, *Crystal holders, *Heating elements, Manufacturing methods.) Miniature electronic equipment.

Identifiers: Glober.

Three methods of sealing glass enclosures, type HC-26, were tried. A gas flame method was found to be best, since control of crystal unit heating is more satisfactory. A sealing machine was developed around the well-known vertical lathe but with an improved gas flow control and vacuum chuck. The operating procedure for this equipment is described. (Author)

AD-271 561   3   N2

AD-271 643   Div. 8, 26   U
(TISTM/DLM)

Georgia Inst. of Tech. Engineering Experiment Station, Atlanta.
AGING CHARACTERISTICS OF QUARTZ CRYSTAL RESONATORS.
Quarterly rept. no. 3, 15 Aug-15 Nov 61,
by R. B. Belser and W. H. Hicklin. 15 Nov 61,
23p. incl. illus.
(Contract DA 36-039-sc-87407)
Unclassified report

No automatic release to Foreign Nationals.

DESCRIPTORS: (*Quartz resonators, *Quartz crystals, Crystals, Aging, Frequency stabilizers, Bonding, Temperature, Seals, Processing, Quality control, Test methods.)

An apparatus for sealing the glass HC-6/U containers was developed. The container and base held in mated position in a jig in vacuo were preheated by a gas ring burner around the pyrex tube vacuum chamber. The Kovar ring in the base was heated by induction and the envelope and base compressed under the action of a spring. A dial gauge measured the relative motion between the parts. A relative movement of 0.020 in. was found to give consistently leakproof seals. Approximately 75 resonators previously fabricated remained on storage at a temperature of 85 C. However, frequency measurements were largely interrupted due to the extensive equipment modifications. (Author)

AD-271 643   9

AD-271 918   Div. 8, 26, 14   U
(TISTM/EJH) OTS price $1.60

Corning Glass Works, N. Y.
PILOT PRODUCTION OF GLASS CRYSTAL HOLDERS (HC-(XM-2)/U; HC-(XM-3)/U; HC-(XM-4)/U).
Quarterly progress rept. nos. 13-14, 1 June-30 Nov 61,
by R. K. Whitney. 30 Nov 61, 15p.
(Contract DA 36-039-sc-81255)
Unclassified report

DESCRIPTORS: (*Crystal holders, Quartz crystals, Glass, Seals, *Glass seals, Industrial production, Manufacturing methods, High temperature research.) (Gold plating, Bronze, Wire, Soldering, Soldering fluxes.)

A change in base suppliers for the HC-(XM-2)/U holder necessitated another preproduction sample for sealing and testing. This testing was completed. Preproduction sealing of the HC-(XM-3)/U holder has yet to be satisfactorily accomplished. The attainment of the specified temperature limit of 250 C. appears beyond attainment with the specified glass and metal and current sealing method. Pilot production of the HC-(XM-4)/U holder was completed. (Author)

AD-271 918   9   N3

AD-272 431L   Div. 14, 26, 30   U
(TISTM/TCT)

Rubber Lab., Mare Island Naval Shipyard, Vallejo, Calif.
SUITABILITY OF VITON B O-RINGS FOR USE IN 3000 PSI HYDRAULIC SYSTEMS CONTAINING PETROLEUM BASE FLUID OR CELLULUBE 220.
Progress rept. no. 4,
by R. D. Ford. 20 Feb 62, 19p. incl. illus. tables (Rept. no. 92-12)
(Proj. S-F013-13-01)
Unclassified report

Notice: Only Military Offices may request from ASTIA. Others request approval of Bureau of Ships, Navy Dept., Wash. 25, D. C.

DESCRIPTORS: (*Hydraulic seals, Oil seals, *Rubber seals, Synthetic rubber, Seals for Organic compounds, Phosphates, Esters or Petroleum, Fluids, Hydraulic fluids.) (Polymers, Eyhylenes, Fluorides.) (Test methods, *Hydraulic systems.) (Aging, Tensile properties, Deformation, Hardness, Corrosion.)

Identifiers: Vitron.

The objectives of this investigation are: (1) to develop a simulated service test for evaluating seals for use in 3,000 psi hydraulic service, (2) to develop a suitable Vitron stock for making such seals for use in both Cellulube 220 and petroleum-base fluid, and (3) to prepare a specification covering the procurement of such seals. The simulated-service test apparatus consists of a piston fitted with two O-rings which reciprocates 100 cycles/min in a heated (160 F) vertical cylinder. The space between the O-rings is filled with hydraulic fluid at 3,000 psig pressure for 9 min and at 0 psig for 1 min. Leakage past each O-ring is measured. Viton B O-rings of commercial origin were tested with Cellulube 220 and found to withstand as many as 5,000,000 without failure. Only slight wearing of the O-rings was observed. (Author)

AD-272 431L   9

AD-272 493   Div. IDEP   U
(TISTE/GEC)

Martin Co., Denver, Colo.
INTRODUCING FLUOROGREEN MATERIALS FOR CRYOGENIC GASKET APPLICATIONS.
Mar 61, 27p. incl. illus. (Rept. no. JDM T-1)
(IDEP rept. no. 345.10.13.85-F3-01)
Unclassified report

DESCRIPTORS: IDEP-Titan. (*Gaskets, Materials, Ethylenes, Fluorides, Polymers, Cryogenics, Liquefied gases, Oxygen.) (*Guided missiles, Surface to surface, Liquid rocket propellants.)

Identifiers: Titan, J. L. Dore' Co. E-600, Compatability.

Gaskets were tested under temperature and pressure conditions anticipated in the operational configuration of the SM 68, Titan Propellant Loading System. The sealing properties throughout a temperature range of 20 to -322 F and throughout a differential pressure range of 0 to 140 psi were tested. The gasket was subjected to extremes of thermal shock experienced in rapid cooldown of cryogenic components prior to flow and the effluent liquid from the line containing the gasket was tested for contamination level. It was subjected to high liquid and gaseous flow rates and checked for possible erosion of inner exposed surface. (Author)

AD-272 493   10

AD-273 046 Div. 17, 26, 14 U
(TISTM/BRW)

Royal Aircraft Establishment (Gt. Brit.).
THE PROTECTION OF MAGNESIUM ALLOYS BY THE M.E.L. FLUORIDE ANODISING AND SURFACE SEALING PROCESSES, by H. C. Davis and J. Neuseman. Nov 61, 12p. illus. tables, 2 refs. (Technical note no. Met Phys 342)
Unclassified report

DESCRIPTORS: (*Corrosion inhibition, Sheets, Castings, Metal plates, *Magnesium alloys, Magnesium castings.) (Effectiveness, Cleaning, Cleaning fluids, *Fluorides, Coatings, Chromates, Anodes (Electrolytic cell), Thin films.) (Seals, *Plastic coatings, Thermosetting resins, Epoxy resins, Paint primers, Paints.) Corrosion by Sea water, Tests, Test methods.

The results are given for sea-water spray tests to determine whether fluoride anodising and/or surface sealing with an epoxy resin could be used to replace or supplement chromate treatment as a protective for Mg-base alloys. Sheet and cast

alloys conformed to Specifications D.T.D. 626A, 742, 690 and 721A, and the processes were compared with and without subsequent painting. Surface sealing at 180 and 200 C after chromate treatment gave better protection than chromate treatment alone, there being little difference between the results with either of these temperatures. Fluoride anodising was detrimental if the anodic film was retained, but when the anodic film was removed, subsequent chromate treatment and surface sealing gives slightly better protection than when the anodic process is omitted. Tests on fluoride anodised castings surface sealed at 220 C indicate a slight advantage in using the higher temperature. (Author)

AD-273 046 6

AD-273 212 Div. 1 U
(TISTA/WAW) OTS price $13.50

Chance Vought Corp., Dallas, Tex.
MODULAR HYDRAULIC SYSTEM DEVELOPMENT. PROJECT HYDRATOY. PART IV.
Final engineering rept. on Modular Hydraulics, by G. Gilder. 31 Aug 61, 189p. incl. illus. tables, 72 refs. (Rept. no. AER-EIR-13120, pt. 4) (Contract NOa(s)59-6019-c)
Unclassified report

DESCRIPTORS: (*Jet planes, *Rocket planes, *Hypervelocity vehicles, Naval aircraft, *Hydraulic systems, Control systems, Servo systems, Hydraulic servomechanisms, Design, Packaging, High temperature research, Installation.) (Metal seals, Hydraulic seals, Hydraulic valves, Hydraulic conduits, Hydraulic actuators, Materials, Manufacturing methods, Tests, Reliability.) (Metals, Stainless steel, Physical properties.)

Identifiers: Hydratoy project.

This report is published in 4 separately bound

parts. Part IV is a report of materials and process development which occurred in conjunction with and as a result of development effort in metallic seals, packages, components, and system integration. (Author)

AD-273 212 7 N5

AD-273 284 Div. 14, 12 U
(TISTM/BRW)

Nonmetallic Materials Lab., Aeronautical Systems Div., Wright-Patterson Air Force Base, Ohio.
SELF-SEALING OF AEROSPACE SHIPS BY THE DOUBLE HOLLOW WALL CONCEPT.
Rept. for Mar-Oct 61 on Nonmetallic and Composite Materials,
by Philip A. House. Feb 62, 14p. incl. illus. (Proj. 7340)
(ASD TDR 62-248) Unclassified report

DESCRIPTORS: (Particles, Meteorites, Penetration, *Spaceships, Satellite vehicles.) (Seals, *Sealing compounds, Polymers, Catalysts, Propanes, Oxides, Toluenes, Cyanates, Ethylenes, Amines, Silicon compounds, Dioxides.) Test equipment.

The system devised is capable of automatically sealing aerospace vehicles in case of puncture by foreign objects such as micrometeoroids. This system repeatedly sealed punctures caused by 22 caliber bullets. The evaluation was aided by the use of an outer elastomeric rubber sheet which

elongated before being punctured, resulting in a hole approximately 0.075 in. in diameter. Such may not be the case in a space ship punctured by a micrometeroid traveling at such speed that the rubber sheet would not elongate and the puncture would probably be at least the size of the particle. The system consists of two fluids separated by a membrane. When the system is punctured, the two fluids mix, immediately solidify, and seal the puncture. The two fluids which demonstrated the greatest success were a polypropylene oxide-toluene diisocyanate adduct thickened with a colloidal silica and triethylene tetramine. (Author)

AD-273 284 29

AD-273 336 Div. 8 U
(TISTE/CDM) OTS price $2.60

Westinghouse Electric Corp., Baltimore, Md.
PHOTOGENERATOR POWER CONVERSION STUDY.
Interim rept. no. 1, Dec 61-Feb 62,
by L. Limansky, E. F. Wood, and A. S. Jensen. Feb 62, 27p. incl. illus. tables.
(Contract AF 33(657)7465, Continuation of Contract AF 33(616)7910)
Unclassified report

DESCRIPTORS: (*Phototubes, *Solar cells, Solar energy, *Power transformers, *Generators, Design, Production, Manufacturing methods.) (Electron tubes, Cathodes (Electron tubes), Photosensitivity, Photoemission, Anodes (Electron tubes), Glass, Insulating materials, Vacuum seals, Processing, Cleaning, Test methods.)

Ways and means of constructing developmental models of a close-spaced, thin, solar power converter using the photoemissive principle were investigated. The report discusses the merits of several designs; the equipment and techniques

available to fabricate, process and test these devices; and anticipated problems. In particular, the criticalness of the in-vacuum sealing is treated and a sealing fixture, wherein different sealing methods may be tested reproducibly, is described. A demountable process tube is described and compared with the WX-4220, an early process control tube. Chemical processing and vacuum processing techniques and problems are discussed. Using these processes, several component parts for the design were constructed. (Author)

AD-273 336 24 N5

AD-273 481 Div. 7, 14 U
(TISTM/GEC) OTS price $2.25

Armour Research Foundation, Chicago, Ill.
SEAL AND INSULATOR PROBLEMS IN THERMIONIC CONVERTERS.
Summary rept., 15 Feb 61-15 Feb 62.
12 Mar 62, 82p. incl. illus. tables, 44 refs.
(Rept. no. ARF 2215-6)
(Contract Nonr-344100, Proj. NR 099 354)
Unclassified report

DESCRIPTORS: (*Power supplies, *Thermionic emission, Seals, Electric insulation, Ceramic materials, Refractory materials, Metallic compounds, Oxides, Boron compounds, Nitrides, Zirconium compounds, Silicates.) (Physical properties, Density, Mechanical properties, Electrical properties, Resistance, Thermodynamics, Neutron cross sections, Neutron capture, Thermal conductivity, Stability.) Bibliography.

A discussion of the materials requirements for seals and insulators for thermionic energy converters is presented. Based on the requirements

and published physical property data, Al2O3 appeared to be the most desirable insulator material at temperatures below its melting point. The study includes a discussion of metal ceramic sealing technology and a bibliography of publications in the field covering 1959, 1960, and part of 1961. The insulator was required to possess a reasonable value of electrical resistivity at temperatures which may reach as high as 2000 K and the specific resistivity of candidate oxides was required to be as high above 100 ohm-cm as possible at any temperature at which it is to function. The thermal conductivity of the insulator was less important than the electrical conductivity but low thermal conductivity was desirable. The insulator-seal combination must be stable in the presence of fission products and have a low capture cross section for slow and fast neutrons. It must be a dense material with reliable physical and mechanical properties. (Author)

AD-273 481 9 N6

AD-273 880 Div. 14, 12, 1 U
(TISTM/BRW) OTS price $3.50

Southwest Research Inst., San Antonio, Tex.
HANDBOOK OF DESIGN DATA ON ELASTOMERIC MATERIALS USED IN AEROSPACE SYSTEMS.
Rept. for 5 Feb 60-30 June 61 of Design Data on Aircraft Elastomeric Materials and Products,
by A. G. Pickett and M. M. Lemcoe. Jan 62, 222p. incl. illus. tables, 233 refs.
(Contract AF 33(616)7093, Proj. 7381)
(ASD TR 61-234) Unclassified report

DESCRIPTORS: (*Elastomers, Design, Data, *Handbooks, Processing, Casting, Plastic coatings, Vulcanization.) (Mechanical properties, Physical properties, Thermal conductivity, Electrical properties, Aging, Heat, Oxidation, Ozone.) (Compatibility with Petroleum, Fuels, Lubricants, Hydraulic fluids, Liquid rocket propellants, Rocket oxidizers, Rocket fuels.) (Radiation effects, Low pressure research, Vibration isolators, Seals, Springs.) (Elasticity, Plasticity, Stresses, Deformation, Shock resistance.) Airframes, Spaceships, Satellite vehicles.

This handbook provides aerospace weapons system design engineers with useful data on the materials properties of elastomers. The data sources are Department of Defense research reports and the technical literature of engineering design and elastomer technology. The elastomeric materials considered are compounds of high polymers currently available in the U. S. A. The properties considered are original mechanical and physical properties and their changes that result from aging and exposure to environments of aerospace weapons systems. Elastomer compounding is only briefly treated because the handbook is intended for use by structural and mechanical engineers rather than by rubber chemists and technologists. A selected bibliography of the technical literature is included. (Author)

AD-273 880 19 N6

AD-274 085L Div. 14, 12 U
(TISTA/SEH)

Armour Research Foundation, Chicago, Ill.
RESEARCH AND DEVELOPMENT OF DESIGN CONCEPTS FOR SEALING APPLICATIONS IN AEROSPACE VEHICLE CABINS. Rept. for 1 Apr 60-15 Dec 61, on Design Technologies and Structural Configuration Concepts for Aerospace Vehicles, by Joseph S. Islinger. Feb 62, 67p. incl. illus. tables, 17 refs.
(Contract AF 33(616)7194, Proj. 1368)
(ASD TR 61-696) Unclassified report

Notice: Only military offices may request from ASTIA. Others request approval of Aeronautical Systems Div., Wright-Patterson AFB, Ohio.

DESCRIPTORS: (*Seals for Maintenance of Controlled atmospheres in *Spaceship cabins, Pressurized cabins.) (Materials for Vacuum seals, Rotary seals, Rubber seals, Plastic seals.) (*Sealing compounds, Rubber, Synthetic rubber, Nitrile rubber, Butadienes, Urethanes, Silicone resins.) (Acceleration, Shock, Vibration, Abrasion, Stability, Space environmental conditions, Pressure.) (Design of Sealing techniques for Spaceships, Satellite vehicles.)

Near absolute sealing techniques are described for small openings, such as for electrical conductors, and hatches in the walls of flight vehicle cabins. The need for absolute sealing stems from the requirement of a habitable environment in such vehicles for as long as one year under differential pressure and temperature extremes from -100 to +500 F. Considerations underlying selection of suitable seal materials for resistance to space environment are discussed. The mechanical design and material factors are also discussed. Investigations concerned with effect of elastomer permeability upon seal behavior and with techniques for sealing electrical wires and cables are described. Concepts for the four sealing applications considered are described. Recommendations are made regarding actual sealing concepts. (Author)

AD-274 085L 9

AD-274 112 Div. 8 U
(TISTE/CAM)

Microwave Associates, Inc., Burlington, Mass.
LONG LIFE K-BAND MAGNETRON.
Quarterly progress rept. no. 1, 1 Oct-31 Dec 61, by Norman Belmuth and David Hobbs. 31 Dec 61, 18p. illus. 2 refs.
(Contract DA 36-039-sc-87359)
Unclassified report

No automatic release to foreign nationals.

DESCRIPTORS: (*Magnetrons, *Miniature electron tubes, K band, Microwave frequency, Theory, Design, Model tests.) (Electron tubes, Cathodes (Electron tubes), Anodes (Electron tubes), Electron guns, Electron beams, Focusing, Magnetic fields, Heating, Life expectancy.) Metal seals, Tungsten.

Research concerns the design and development of a 5000 hr life, 50 kw, K band, fixed frequency pulsed magnetron, incorporating injection gun electron sources, for use with ASDE radar equipment. The initial magnetron design was completed; detailing was accomplished with the exception of the ion pump and some packaging parts. Assembly jigging has begun. The electron gun design has progressed through the selection and initial design processes. Efforts to obtain a sufficiently high fringing magnetic field within and at the mouth of the gun were successful, and gun parameters are being calculated. (Author)

AD-274 112 9

AD-275 443 Div. 8 U
(TISTE/CAM) OTS price $7.60

General Electric Co., Owensboro, Ky.
DEVELOPMENT OF AN IMPROVED JAN 6299.
Quarterly progress rept. no. 4, 1 Feb-30 Apr 62, by F. S. Sawicki. 30 Apr 62, 69p. incl. illus. tables.
(Contract DA 36-039-sc-85953)
Unclassified report

DESCRIPTORS: *Electron tubes, *Manufacturing methods, Anodes (Electron tubes), Cathodes (Electron tubes), Seals, Coatings, Design, Processing, Electrical properties, Temperature, Humidity, Life expectancy, Tests.

Effort continued towards the evaluation of certain areas of advanced tube manufacturing techniques and their application where feasible to the JAN 6299 to result in a more reliable and better performing tube. Tubes were fabricated with the one piece cathode support clip. Thirteen tubes from the lot of 250 tubes fabricated with hard solder anode seals were tested electrically and mechanically. The tubes and associated data have been submitted for Qualification Approval and the results are presented in tabular form. Life test data of two tubes fabricated with hard solder anode is also included. (Author)

AD-275 443 9 N4

AD-302 154 Div. 1/3, 1/11

Ballistic Research Labs., Aberdeen Proving Ground, Md.
PASSIVE DEFENSE FOR AIRCRAFT FUEL TANKS (Unclassified title) by John C. Harris and Roland G. Bernier. June 58, 44p. incl. illus. tables. (Proj. no. TB3-0238; BRL Memo. rept. no. 1150)
Confidential report

Two means of providing passive defense for the aircraft fuel system were tested: (1) a mechanical-seal cell for preventing fuel loss and (2) some inerting mats for preventing fires. Incendiary and high explosive projectiles (up to 40mm) were fired against mock-up and actual aircraft fuel targets. General principles of aircraft fuel vulnerability are discussed and pertinent firing test summaries are appended. (Author)
(Unclassified abstract)

AD-303 513 Div. 29/6, 3/4, 14/10

Forest Products Lab., Madison, Wis.
PROTECTIVE ENCLOSURE PROJECT (Unclassified title) by Alfred J. Stamm and Harold Tarkow. Progress rept. no. 3 on Thermal Degradation of Fiberboards and Protective Enclosure Boards. 1 May 55, 3p. illus. [CWL rept. no. ETF 634-10/16-6A] (In cooperation with Chemical [Warfare] Labs., Army Chemical Center, Md.)
([Contract] CP4-3943; [Continuation of Contract DA 18-108-cml-3487]) Confidential report

See also AD-87 646, AD-303 514

AD-303 514 Div. 29/6, 3/4, 14/10

Forest Products Lab., Madison, Wis.
PROTECTIVE ENCLOSURE PROJECT (Unclassified title) by Alfred J. Stamm and Harold Tarkow. Progress rept. no. 4 on Thermal Degradation of Fiberboards and Protective Enclosure Boards. 1 Aug 55 [11]p. incl. tables [CWL rept. no. ETF 634-10/18(8A)] (In cooperation with Chemical [Warfare] Labs., Army Chemical Center, Md.)
([Contract] CP4-3943) Confidential report

See also AD-87 646, AD-303 513

AD-304 961 Div. 22/1, 22/13

Naval Ordnance Lab., White Oak, Md.
METHODS OF ATTACHING ACCESSORIES TO PLASTICS MAGNETOMETER CANS, by A. Fisher. 9 Oct 58, 30p. illus. (NAVORD Rept. no. 6141)
Confidential report

Techniques have been devised for the attachment of "Penton", "Cycolac" and "Kel-F" plastics accessories to their respective plastics magnetometer can bodies by heat sealing. These accessories consist of a cover, cable housing, cable clamp and flanged permalloy slug supports. Sealing of oil filler hole bosses in the covers is also included. Sealing devices, surface preparation and heating cycles are described. The techniques presented have been shown to provide attachments of accessories to magnetometer cans capable of meeting current environmental test requirements for firing mechanisms Mk 25 Mod 0 and Mk 15 Mod 0 wherein the magnetometer cans are employed. (Author) (Unclassified abstract)

AD-310 520 Div. 3/4, 3/7
(15 Aug 61)
(Unannounced report)

Southwest Research Inst., San Antonio, Tex.
NEW PROTECTIVE METHODS FOR COMBAT TROOPS (U). Technical rept. no. 3, 1-30 Sep 54. 8 Oct 54, 42p. incl. illus. tables.
(Contract DA 18-108-cml-5526)
Confidential report

For reference only at each of the ASTIA Offices. This report cannot be satisfactorily reproduced; ASTIA does not furnish copies.

DESCRIPTORS: *Shelters, Military personnel, Combat, *Chemical warfare agents, Countermeasures, Vehicles, Fortifications, Buildings, Gas seals, Tents, Joints, Adhesive tapes.

AD-310 114 Div. 3/4, 29/6
(4 Jan 60)
(Unannounced report)

Institute of Research, Lehigh U., Bethlehem, Pa.
EQUIPMENT AND METHODS FOR COLLECTIVE PROTECTION. [U] Summary rept. no. 4 on Phase 3 a-b Classification System for Collective-Protective Shelters and Directions for Conversion. 30 June 54, 28p. incl. illus. tables, 11 refs.
(Contract DA 18-108-cml-4504)
(ETF 620-14/Summary rept. no. 4) Confidential report

Descriptors: Protective shelters*; Design; Ventilation; Seals; Chemical warfare agents.

See AD-310 130

AD-316 146L Div. 12/1, 12/3
(14 Apr 60)

Aberdeen Proving Ground, Md.
VIBRATION TESTS OF THE SECOND INDUSTRIAL PROTOTYPE HAWK MISSILE SYSTEM (U). by G. Hinb. Mar 60, 1v. incl. illus. tables (Engineering labs. rept. no. 103) Confidential report

Notice: Only Military Offices may request from ASTIA. Others request approval of Diamond Ordnance Fuze Labs., Wash. 25, D. C. Attn: ORDTL 06.33

Descriptors: Guided missiles*; Surface to air; Mobile; Transportation; Vibration; Tests; Reliability; Seals*; Effectiveness; Radar trailers*; Electronic equipment*; Cargo vehicles*; Shock resistance; Guided missile launchers*; Radar equipment*.

The purpose of test was to determine the reliability of the mounted electronic and mechanical equipment and to measure the magnitudes and frequencies of forces imparted to various predetermined locations and to determine the effectiveness of the sealing provided for water protection. Modifications are recommended a- - an endurance test as outlined in OPM 60-116 be conducted on the l battery to more (over)

accurately establish the equipment reliabili... (U) (Author) (See also AD-316 146L)

Open-ended Terms: Hawk.

AD-316 146L

AD-316 397 Div. 22/1
(5 May 60)

Bulova Research and Development Labs., Inc. [Woodside, N. Y.]
DESIGN AND DEVELOPMENT OF FUZE, BD, SD, T300 [U] by M. Brandon and I. McRae. Quarterly progress rept. no. 11, 1 Nov 59-31 Jan 60. 12p. incl. tables.
(Contract DA 30-069-501-ORD-2417) Confidential report

Descriptors: Base detonating fuzes*; Projectile fuzes*; Design; Temperature; Humidity; Tests; Quality control; Production*; Reliability; Antiaircraft ammunition; Projectiles; Seals.

See also AD-314 158, AD-314 521

Open-ended Terms: T300 fuze.

AD-316 580 Div. 3/5, 12/5
(6 May 60)

Vickers, Armstrong, Ltd. (Gt. Brit.).
NOTES ON VISIT TO A.R.D.E. LANGHURST ON 27th JULY, 1959 TO DISCUSS VIGILANT FLARE [U], comp. by S. A. Johnson. 5 Aug 59, 3p. (Rept. no. GW2/B/08/211) Confidential report

Descriptors: Guided missiles*; Surface to surface; Great Britain; Pyrotechnics*; Materials*; Powders; Colored flares*; Containers*; Design; Seals.

Open-ended Terms: Vigilant.

AD-321 271 Div. 12/1, 12/2, 12/4, 6/1, 6/3, 6/6, 8/6, 25/3 U
(24 Jan 61)

Institute of Science and Tech. U. of Michigan, Ann Arbor.
TRANSACTIONS OF THE BAMIRAC 1960 SUMMER STUDY (U).
Oct 60, 1v. incl. illus. tables, 267 refs. (Rept. no. 3768-6-X)
(Contract AF 19(604)7350)
Secret report

DESCRIPTORS: *Guided missiles, Launching, Guided missile trajectories, *Re-entry aerodynamics, Re-entry vehicles, Infrared radiation, *Electromagnetic waves, Measurement, Acoustics, Exhaust gases, Ionization, *Shock waves, Boundary layer, Microwaves, Scattering, Ionospheric disturbances, Mathematical analysis, Instrumentation, *Bibliography, *Symposia, *Antiaircraft defense systems, *Detection.

Open-ended Terms: Bamirac, Press Project, De- (over)

fender Project.

An unedited collection is presented of memoranda and papers prepared during the course of two-week study session on missile phenomenology. The study session undertook to review the state of the air of studies in missile phenomenology, to construct and evaluate mathematical models for studying ballistic missiles as objects in space from launch to impact, and to compare the fit of these models with experimental data. The group's working papers were divided into three aspects: launch, mid-course, and re-entry. Participants in the study session were drawn from a number of research and industrial organization and included specialists in problems of plume structure, plume radiation, radar cross-sections, radar measurements, ionospheric disturbances caused by missiles, re-entry radar and infrared measurements, nose-cone design, and atmospheric structure. (U) (Author)

AD-321 271

AD-321 948 Div. 27/3, 1/5, 22/1 U
(6 Mar 61)

Ogden Air Materiel Area, Hill Air Force Base, Utah.
SERVICEABILITY OF UNOPENED ROCKET MOTORS, 16-NS-1000, M15 AND M15A1 WITH IGNITERS. (U)
by Samuel H. Welch. Jan 61, 28p. incl. illus. tables (Rept. no. OOY TR 61-5)
Confidential report

DESCRIPTORS: *Rocket motors, *Igniters, Tests, Seals, *Jet bombers, Shipping, Statistical analysis, Reliability, *JATOS, Pellets, Ignition, Storage, Stability.

Open-ended Terms: 16-NS-1000 Rocket motor, B-47.

This test program was initiated as a result of a malfunction of a Rocket Motor, 16-NS-1000 at Dyess AFB, Texas causing the crash of a B-47 Aircraft. The motors and igniters were static fired. The ignition delay times, burning times, maximum chamber pressures and average chamber pressures recorded from the static firings were statistically analyzed and determined to be within the limits of the design specifications for the motors and igniters. The Air Force stocks of Rocket Motors, 16-NS-1000, M15 and M15A1, which have not had the hermetic seal broken by removal of the shipping plug for installation of the igniter and which have not exceeded the established shelf life or failed to meet the inspection criteria, are serviceable and would function satisfactorily. It is further concluded that the igniters that were manufactured using the B-73 formulation pellets were of doubtful reliability and should not be used for ignition of the rocket motors. (U) (Author)

AD-321 948

AD-322 684 Div. 12/3 U
(17 Apr 61)

Naval Ordnance Test Station, China Lake, Calif.
LAUNCH-VALVE DESIGN FOR 1/5 SCALE POLARIS MODEL TESTS IN VAT (U)
by O. N. Nelson. 14 Dec 60, 48p. incl. illus. tables. (Rept. no. NOTS TP 2593)
(NAVWEPS rept. no. 7602) Confidential report 4

DESCRIPTORS: Underwater, Underwater to surface, Surface to surface, Guided missiles, Model tests, Guided missile launchers, Launching, Pressure, Seals, Mathematical analysis, Theory, Tests, Thermodynamics, Equations, Test facilities, Calibration, Hydraulic systems, Hydraulic accumulators, Hydraulic valves, Hydraulic seals, Ejection, Air ejectors.

Open-ended Terms: Polaris, O-rings, VAT.

A detailed description of the development of the model launch-valve system for the 1/5 scale Polaris tests in the Variable-Atmosphere Tank is given, including design of the launch-valve contour, modifications of the basic launch-valve hardware and hydraulic system to improve launcher performance, and comparisons of desired and actual-launcher performance for the various phases of development. (U) (Author)

AD-322 684

AD-323 114 Div. 1/5, 1/9, 1/11 U
(15 May 61)

Ballistic Research Labs., Aberdeen Proving Ground, Md.
EFFECTIVENESS OF LIGHTWEIGHT SELF-SEALING FUEL CELL MATERIAL AGAINST CAL. 0.30 BULLETS (U)
by Walter S. Vikestad. Jan 61, 20p. incl. illus. tables. (BRL memo. rept. no. 1324)
(Proj. 503-05-010)
Confidential report

DESCRIPTORS: *Liaison planes, *Helicopters, *Fuel tanks, *Fuel seals, Test methods, *Armor piercing ammunition, Ammunition, Effectiveness, Antiaircraft ammunition, Vulnerability, Tests, Seals, USSR, Rubber seals, Nylon, Cordage.

Open-ended Terms: Cal 30.

Firing tests were conducted on a new type of lightweight self-sealing fuel cell material. The 7.62mm AP (Cal. 0.30) ammunition was fired at four types of self-sealing material under various striking conditions. These four types were approximately half the thickness and half the weight of conventional lightweight self-sealing materials for aircraft fuel cells. (U) (Author)

AD-323 114

AD-323 670 Div. 12/5, 22/10 U
(14 June 61)

Diamond Ordnance Fuze Labs., Washington, D. C.
PRODUCTION ENGINEERING OF THE T3019 BOMARC FUZE (U),
by J. J. Colliname and C. K. Eve. 10 Mar 61, 31p. incl. illus. (DOFL rept. no. TR-895)
(Proj. 77260)

Confidential report

DESCRIPTORS: *Guided missiles, *Surface-to-air, Engineering, Design, Arming devices, *Guided missile fuzes, Quality control, Reliability, Cooling, Costs, Fuzes, Seals, Production, Standardization, *Auxiliary detonating fuzes, *Proximity fuzes, Guided missile antennas, Radar.

Open-ended Terms: Bomarc, T3019 fuzes.

The activities of the T3019 Bomarc fuze production-engineering program are summarized. The objectives of this program were to modify fuze design to assure reproducibility and to prepare a technical data package for procurement of the fuze. To attain these objectives, it was necessary to perform cost, tolerance, and interchangeability studies; to improve the sealing and cooling capabilities of the fuze packages; and to perform reliability engineering. These activities resulted in a production-engineered fuze design and a technical data package that provided to the potential fuze manufacturer the information necessary to produce fuzes of consistent quality and predictable reliability. (U) (Author)

AD-323 670

AD-324 132 Div. 27/3, 14/2, 14/8 U
(18 July 61)

Naval Ordnance Test Station, China Lake, Calif.
SOME RESULTS OF APPLIED RESEARCH ON MATERIALS AND PROCESSES FOR PROVIDING HEAT-RESISTANT ROCKET COMPONENTS. SUPPLEMENT 2 (U).
May 61, 128p. incl. illus. tables, 12 refs.
(Rept. NOTS TP 2666)
(Suppl. no. 2 to NAVORD rept. no. 6370, AD-309 172)

Confidential report

DESCRIPTORS: *Rocket motors, *Rocket motor nozzles, Materials, Metalorganic compounds, Polymers, Aldehydes, Phenoxy radicals, Resins, Refractory materials, Pyrolysis, Binders, Phenolic resins, Solid rocket propellants, Hybrid rocket propellants, Synthesis, Metals, Oxides, Carbides, Guided missiles, Plastic seals, Servo mechanisms, Nozzles.

Open-ended Terms: Sidewinder, Vector control systems, Subroc.

Procedures for synthesis of refractory-metal phenoxyaldehyde high polymers are described, along with evaluation data and possible applications. The following developments are described: (1) a large 'SUNROC' nozzle evaluating secondary injection for vector control; (2) a special nozzle for the sustainer motor of the Sidewinder (SW) 1C Target Rocket; (3) a method of protecting the wiring of the SW 1C servo; and (4) a meaningful performance test for evaluating O-rings in a dynamic application. Data are presented on: (1) reinforced plastics; (2) materials for the SW 1C Mod 0 booster nozzle; (3) nozzle materials for the hybrid system; and (4) potential nozzle materials. (C) (Author)

AD-324 132

AD-326 200 Div. 10/6, 12/3, 12/4 U
(TIPSM/EJH)

Redstone Div., Thiokol Chemical Corp., Huntsville, Ala.
DEVELOPMENT OF PROPULSION SYSTEM FOR NIKE ZEUS MISSILE (U).
Monthly progress rept. for Sep 61.
26 Oct 61, 15p. incl. illus. tables (Rept. no. C-A-61-288A)
(Subcontract to Douglas Aircraft Co., Santa Monica, Calif.)

Confidential report

DESCRIPTORS: (*Guided missiles, Surface to air, *Rocket propulsion.) (*Rocket motors, *Solid rocket propellants, *Booster rockets, Boosters, Rocket cases, Loading, Storage, Low temperature research, Combustion, Ignition, Flight testing.) (*Rocket motor nozzles, Seals.) *Solid rocket propellants, Polymers, Sulfides, Propellant properties, Pyrogens, Butadienes, Acrylic resins, Ferrocenes.

Identifiers: Nike-Zeus.

First stage motor: 4 motors were flight tested, all of them operated successfully; 9 motors were shipped; 3 motors were loaded; and the pyrogen seal for the tactical nozzle closure was tested for leakage. Second stage motor: 2 motors were flight tested, both operated successfully; 5 motors were shipped; 7 motors were loaded; the first phase of a study to determine effects of loss of ferrocene from sustainer propellant at elevated temperatures was completed; 16 pyrogens were loaded. Third stage motor: 3 polysulfide motors were inspected and accepted; 8 pyrogens were manufactured; low temperature capabilities of the polysulfide motor were determined; a study showed that the polysulfide third stage propellant would not be materially affected by acceleration loads. (U)

AD-326 200

AD-326 483 Div. 22/1 U
(TIPTW/DLW)

Hamilton Watch Co., Lancaster, Pa.
SSS, MINE, APERS T-52 W/INTEGRAL FUZE (U).
Monthly progress rept. no. 17 for Sep 61, by R. C. Slever. Sep 61, 28p. incl. table.
(Contract DA 28-017-ORD-3911, Proj. 5807-06-011-01)

Confidential report

DESCRIPTORS: *Antipersonnel ammunition, *Mine fuzes, *Mines, Fuzes, Mine cases, Plastics, Seals, Moistureproofing, Design, Production, Temperature, Humidity, Vibration, Tests, Reliability, Epoxy resins.

Identifiers: T52 mines.

AD-327 982 Div. 27/3, 10/3, 12/4, 26/1, 30/4 U
(TISTA/VGW)

Rocketdyne, Canoga Park, Calif.
F-1 (U).
Quarterly progress rept. no. 12, 1 Oct-31 Dec 61, on NASA F-1 Engine Development Program (U).
12 Feb 62, 223p. incl. illus. tables (Rept. no. R-1555-12)
(Contract NASw-16)

Confidential report

DESCRIPTORS: (*Rocket motors, Heat resistant alloys, *Combustion chambers, Fuel injectors, Orifices, Rocket motor nozzles, Vane diffusers, Turbines, Fuel pumps, Thrust bearings, Seals, Fuel seals, Design, Tests, Effectiveness, Gas generating systems, Control valves, Control systems, Hydraulic couplings, Hydraulic actuators, Gimbals.) (Configuration, Manufacturing methods, Ignition systems, Vibration, Stability, Reliability, Fuel systems, Liquid rocket propellants, Combustion, Explosions.)

Identifiers: F-1 engines, Nova.

A description is presented of the progress made in the twelfth quarter of the NASA F-1 engine development program. Results of studies, tests, and fabrication procedures are discussed and graphically presented, and problems encountered are described. (U) (Author)

AD-327 982

SUBJECT INDEX

I. KINDS OF SEALS

Bushing - Sleeve fitted with close clearance over a shaft or rod.

140-613	201-739	225-213	235-052

Caulking and In Place Sealing - Sealing by injecting a fluid into a cavity.

Dynamic

239-180	241-830

Static

001-046	021-274	021-275	028-321
028-862	036-428	041-362	047-671
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Diaphragm - Dividing membrane which spans the gap between two mechanical components to prevent interchange of a fluid between two separate areas or chambers.

General

254-614

Dynamic

025-809	204-016	259-086	265-582

Diaphragm - (Cont'd)

Static

022-068	098-646	106-754	204-016
206-707	240-573		

Face or Mechanical Axial Seal - Two flat sealing faces contacting at right angles to the axis of rotation, with an axial force applied to cause the faces to contact each other.

Dynamic

011-080	025-808	025-809	043-686
048-377	060-141	094-807	094-809
119-437	121-112	123-771	134-918
143-970	201-024	202-438	202-439
205-179	207-505	225-213	232-073
258-535			

Static

047-412	060-141	232-073	258-535

Freeze Seal - Dynamic seal of a bushing type which also serves as a static seal by solidification of the fluid in the clearance in the absence of friction due to motion.

Dynamic

266-768

Gaskets - Piece of material which is clamped between static faces

General

029-951	123-839	134-197	136-780
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Dynamic

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103-928	116-938	147-856	153-356
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Gaskets (Cont'd)

Static

Plastic Gaskets

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Natural Rubber Gaskets

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Synthetic Rubber Gaskets

023-155	024-489	037-096	037-100
038-529	043-419	063-258	063-260
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072-267	072-268	072-269	074-322
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Gaskets made of other materials

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Hermetic Seal - A permanent seal by fusion or adhesion primarily for moisture and air-tightness.

Dynamic

054-206	055-132

Static

002-787	012-531	030-136	036-452
037-705	041-362	041-652	067-223
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Labyrinth - A device employing a tortuous leakage path, usually by fins on either the stationary or rotating member.

Dynamic

042-033	085-318

Lip or Radial Seal - Seal producing positive line contact circumferentially between the seal's inside diameter and a shaft or a rod.

Dynamic

094-444	121-112	122-294	222-593

Magnetic Seal - A seal produced by applying a magnetic field to concentrate magnetic particles suspended in the fluid.

056-036	156-901

Packing or Stuffing Box - Deformable material compressed into a cavity surrounding a rod or a shaft.

General

128-326	254-614

Dynamic

008-640	025-907	030-373	036-456
036-564	037-218	041-961	043-686
048-377	070-011	074-089	089-193
089-198	089-199	094-674	102-156
123-771	147-092	149-981	153-033
153-356	153-642	204-016	205-179
207-979	214-003	214-848	234-860
236-837	240-838	242-962	259-550
259-574			

Static

047-739	048-399	070-011	089-198
089-199	097-258	134-497	140-116
140-117	140-118	140-125	145-239
153-356	204-016	214-262	256-218
265-555			

Ring Seal - A positive contact seal whose primary component is an independent ring-proportioned element.

Felt, Quad, Delta, X, or other cross sections of ring seals.

Dynamic

048-808	240-838	259-550

Static

010-979	048-807	231-223	238-431
239-509	247-172		

"O" Ring - A ring seal with a circular cross section

General

027-728	029-114	078-234	089-193
089-198	123-839	127-094	131-094
134-197	145-846	146-748	146-749
150-973	155-541	162-472	163-020
205-587	213-568	232-977	241-589
246-574	247-750	250-028	250-742
254-607	254-806	256-590	260-920
261-561	261-562	265-443	267-121

Ring Seal (Cont'd)

Dynamic

008-640	025-907	070-011	074-089
077-374	089-199	094-809	103-928
118-213	129-444	145-307	147-856
151-181	207-979	209-179	229-115
237-833	240-838	248-647	249-387
252-996	260-465	262-398	265-443
267-145	267-501	270-746	272-431

Static

010-979	011-484	027-728	047-412
048-807	050-626	070-011	089-072
089-198	089-199	094-749	098-106
099-706	101-096	101-097	101-098
101-099	119-330	123-912	134-493
140-116	142-079	142-604	143-542
145-357	145-832	151-181	161-638
161-694	161-745	203-158	204-064
207-016	207-035	207-190	207-191
207-538	209-179	229-186	231-223
237-833	245-449	248-647	249-387
261-385	261-564	262-718	267-145
267-501	268-381	269-310	

Piston Ring - A one piece ring seal with a discontinuous circumference.

General

238-164

Dynamic

033-877	105-415	153-356	162-882
221-803	257-228		

Static

033-877	142-079	153-356

Segmental Ring - A ring seal composed of several butting or interlocking segments.

Dynamic

154-494	225-213	239-180	241-830

Static

239-180	241-830	247-172

Ring Seal (Cont'd)

"V" Ring - A ring seal having a "V" shaped cross section.

Dynamic

234-860

"U" Ring - A ring seal having a "U" shaped cross section.

Dynamic

077-374

Special Sealing Concepts (self sealing fuel tanks, injection seals, ceramic to metal seals, etc.)

Dynamic

002-400	048-377	077-374	097-258
118-213			

Static

In Aircraft (self sealing fuel tanks, canopies, etc.)

003-133	015-012	028-321	030-136
047-671	048-310	049-645	050-800
055-695	058-142	063-064	074-322
078-224	106-754	123-436	131-091
200-381	200-382	200-385	225-495
302-156	323-114		

In Munitions

001-046	021-274	021-275	094-922
107-713	107-802	112-953	130-661
140-572	149-799	158-661	159-156

In Launch Vehicles

083-716	147-837	150-474	240-357
242-285			

In Electrical and Electronic Equipment

030-136	088-545	109-660	110-287
122-254	226-195	256-214	260-114
274-112			

Special Sealing Concepts (Cont'd)

Static (Cont'd)

In other Applications

009-079	010-270	013-311	016-330
027-587	028-429	030-417	089-767
089-828	097-454	097-649	097-767
098-226	100-488	101-331	102-222
106-511	107-960	117-021	120-620
130-210	135-272	140-180	145-518
149-681	151-257	151-729	153-456
155-606	208-687	210-323	221-401
221-404	222-684	226-294	228-239
234-516	234-669	235-381	239-384
240-969	241-629	246-644	248-535
249-262	249-349	252-607	257-978
263-640	265-170	266-158	266-502
268-416	269-404	270-582	273-284
273-481	274-085		

Screw Seal - A device comprising a threaded shaft rotating in a close clearance plain bore (or plain shaft in threaded bore), sometimes called visco-seal or screw pump.

II. SPECIFIC INTEREST AREAS IN SEALING

Aging or Shelf Testing

009-494	013-714	023-155	027-728
037-100	055-695	067-223	067-333
067-600	090-148	098-106	107-274
113-822	113-830	119-330	125-427
134-405	145-239	145-307	145-357
145-361	146-748	150-973	151-257
155-606	161-951	200-230	200-385
201-085	202-445	221-461	229-186
232-977	236-067	246-574	248-647
249-262	250-028	256-214	260-114
265-443	267-501	273-880	

Back-up Rings - A washer type of reinforcement placed on the side of a packing, ring seal, or similar seal opposite the sealing interface.

007-347	025-907	029-114	070-011
074-089	089-198	089-199	105-671
142-079	151-181	207-979	237-833
241-589	252-996	260-465	265-443
267-145	270-746		

Balancing of Forces - The balancing of forces due to pressure on a face seal to reduce contact pressure at the sealing interface.

042-477	119-437

Bellows - Used alone or as a component of another seal in static or limited motion application.

201-085	234-860

Cavitation

213-956	262-196

Compatibility

Material-Fluid: the ability of a material to exist in intimate contact with a fluid without detrimental chemical, bacterial, radiation, or other effects.

022-068	024-489	037-096	037-705
038-529	063-258	064-335	067-333
072-267	125-427	134-181	134-405
145-399	145-518	147-990	151-257
153-356	155-606	157-911	157-912
158-530	161-971	200-862	201-085
201-202	201-203	201-204	202-445
203-011	203-833	208-687	212-661
218-038	220-832	226-521	228-846
229-700	229-707	234-721	238-020
238-482	241-831	243-966	247-750
249-387	253-753	254-607	256-590
261-561	261-562	261-563	261-564
267-111	268-381		

Material-Fluid-Material: the ability of relative motion to exist at the sealing interface in the presence of a seal material, fluid, and mating material.

003-133	011-484	013-714	028-321
037-218	037-705	055-695	078-224
080-889	120-620	128-326	130-210
135-272	140-572	142-079	145-846
147-856	163-020	201-024	203-011
204-016	204-065	205-478	206-707
208-838	212-634	225-495	240-573
242-285	242-347	246-574	249-542
251-130	252-992	252-996	254-496
255-145	256-678	257-228	260-465
265-443	267-121	267-890	269-310
270-746	272-431	273-880	274-085

Chemical Properties

028-862	037-705	072-231	118-597
134-405	150-973	201-835	220-860
245-424			

__Endurance Testing__

023-155	025-809	025-907	034-171
035-302	036-456	036-564	048-377
074-089	140-180	142-079	154-494
207-979	213-956	233-471	238-200
240-838	242-744	250-749	267-122
268-420	275-443	316-146	

__Faces__ - Effect of face surface contours or other face characteristics on the sealing capabilities.

123-771	154-494	258-535

__Flange Design__

009-494	116-938	157-761	222-849
232-073			

__Flange Serrations__

009-494	130-960

__Floating Types__ - Sealing without directly fastening the seal to any other component. (excluding ring seals)

105-415	153-033	214-003	214-848
234-860	236-837	259-550	

__Friction__ - A general discussion of friction and its effect upon sealing.

008-640	033-877	036-456	086-205
102-156	103-928	118-213	123-604
153-033	201-835	207-979	222-071
238-832	240-838	241-831	254-806
266-936	267-501		

__Friction, Torque or Power Loss Due to Friction__

094-807	134-918	158-661	161-932
206-773	247-172	252-992	256-649
258-865	266-768		

__Gas Sealing Applications__

002-991	006-687	018-810	023-155
036-319	041-362	042-477	048-377
059-125	067-223	068-873	075-568
082-916	085-318	086-205	105-415
106-777	107-274	107-960	110-636
113-178	116-389	119-437	128-365
140-613	147-856	157-761	162-726
200-028	202-438	204-937	207-505
207-979	208-687	210-822	213-956
214-848	219-109	234-860	234-940
235-052	237-928	238-020	238-164
238-431	239-509	240-357	241-565
242-285	242-962	245-377	246-034
246-574	248-037	248-647	249-144
252-786	259-086	259-550	260-368
260-526	260-920	261-561	261-562
261-563	261-564	261-886	262-398
262-439	263-029	267-111	267-121
268-416	272-493	310-520	

__Heat Transfer__ - The ability of a seal to conduct heat and/or withstand thermal cycling.

037-705	041-362	067-223	075-568
082-914	092-941	097-547	102-222
107-274	120-853	130-661	130-772
140-572	140-614	142-079	159-156
200-028	200-898	202-438	202-439
204-937	207-505	216-838	221-803
222-071	234-109	234-721	235-052
237-500	238-832	258-784	268-381
272-493			

__Installation__

036-428	121-294	121-502	162-481
209-179	231-223	247-172	254-804
268-426			

__Leakage__ - a general discussion of seal leakage, a specific value of leakage is given, or a comparison of leakage rates is made.

002-400	002-991	006-687	007-347
009-079	010-822	011-722	016-848
042-477	050-800	067-362	072-503
073-961	082-915	086-205	094-922
099-469	100-300	101-003	101-331
106-677	110-390	117-021	119-437
119-763	120-853	121-461	130-882
134-918	144-596	147-990	155-541
162-472	200-381	201-204	202-445
203-833	213-564	214-269	218-903
219-139	225-501	228-320	234-516
234-862	236-837	241-562	241-589
244-668	248-037	249-144	254-614
255-145	259-803	260-526	261-886
261-958	263-029	266-158	268-420
269-783	270-582	273-046	

Dynamic Seal Leakage

036-456	048-377	074-809	077-374
094-807	105-415	118-213	123-771
209-179	214-848	222-593	225-213
229-115	232-073	234-860	239-180
248-647	259-086	259-550	262-398
270-746	272-431		

Static Seal Leakage

009-494	011-484	036-452	041-362
041-652	048-399	050-626	078-224
082-914	082-916	089-914	097-258
098-646	107-274	108-720	116-389
121-339	123-604	130-661	140-116
140-572	145-239	146-232	149-799
153-347	157-761	159-156	209-179
228-846	232-073	238-431	239-180
248-647	250-231	251-669	262-566
266-351			

Leakage Test Techniques or Detection

002-011	002-991	006-687	010-822
018-810	041-362	046-159	050-800
058-978	068-873	082-914	082-915
082-916	092-941	101-003	105-415
106-511	110-390	113-178	120-853
154-814	162-726	206-622	209-169
214-101	218-730	234-140	241-564
241-565	245-377	246-034	248-037
249-144	251-453	257-411	258-784
260-235	260-385	260-526	

Leakage by Molecular Diffusion or Permeation

116-938	210-822	236-067	241-565
242-347	243-897	245-377	246-034
248-037	249-144	260-526	263-029
247-085	303-514		

Leakage Testing Results

002-787	002-991	006-687	013-311
016-848	036-452	041-362	046-159
048-807	048-808	051-200	068-873
072-503	074-089	075-568	082-914
082-915	092-941	106-754	107-713
121-112	206-773	212-661	218-730
219-139	220-293	233-471	237-928
241-564	241-565	242-285	245-377
246-034	248-037	249-144	250-742
257-411	254-020	260-526	265-582
267-111	272-493	326-200	

__Inverted Type__ - (seal rotates with the shaft)

048-808

__Loading and Sealing Characteristics__ - The installation or loading requirements and their influence upon seal life, wear, performance, etc. (degree of compression, tolerances, etc.)

036-319	042-033	047-739	050-626
077-374	089-193	098-205	102-156
107-960	116-389	118-213	157-761
162-329	236-837	242-962	

Lubrication

Fluid Film

094-809	097-649	118-047	273-046

Solid

086-205	094-807	118-047	155-689
201-024	266-936		

Lubricant Materials

008-640	013-714	025-809	025-907
034-171	037-705	056-708	094-809
121-502	153-642	163-020	214-848
216-630	241-589	241-831	247-172
247-750	249-542	252-996	254-804
255-145	256-590	258-535	259-550
260-465	267-890	268-381	

Magnetic - A method of affecting the contact pressure at the sealing interface by magnetic force.

Dynamic

077-374	156-901	200-028

Manufacturing Methods

002-991	010-270	082-914	094-922
097-454	108-720	120-620	144-596
147-837	151-729	151-747	153-347
200-859	201-321	214-082	218-730
225-501	234-862	235-681	236-067
236-184	243-685	244-668	245-318
250-399	252-786	254-496	256-649
263-722	265-170	266-246	267-145
268-416	269-034	269-310	269-404
270-222	271-918	275-443	323-670

Material Properties - (Mechanical and Physical)

of Elastomers (Buna-N, GR-S, Butyl, etc.)

119-330	121-339	123-912	128-326
150-973	151-181	151-257	155-606
163-020	204-016	204-065	225-495
229-186	238-020	240-357	245-424
247-750	248-647	249-572	251-130
254-496	256-218	261-561	261-562
261-563	261-564	265-443	267-145
267-501	267-890		

of Fluoro-Elastomers (Kel-F, Teflon, Viton)

070-011	145-307	145-846	150-973
151-257	155-606	163-020	225-495
237-833	242-285	246-574	256-590
260-920	261-563	261-564	265-443

of Natural Rubber

010-979	013-714	036-319	037-093
049-645	125-427	145-362	153-356
155-606	161-745	163-020	221-461
229-186	231-622	232-977	237-833
247-750	248-647	249-387	250-028
250-742	251-130	253-753	254-496
254-607	254-806	256-218	256-590
257-025	260-920	261-561	261-562
261-563	261-564	267-501	267-890
268-381			

of Plastics (Nylon, etc.)

037-705	049-645	055-695	067-333
067-600	097-767	098-226	140-572
159-156	163-020	202-445	204-065
205-810	205-951	206-374	214-262
220-048	225-495	234-721	237-833
238-020	249-542	251-130	258-784
259-086	265-443	268-381	

of Other Materials

021-275	024-489	028-862	035-302
036-456	037-096	067-223	074-089
107-274	113-822	113-830	118-213
119-763	120-853	145-360	145-361
145-572	145-833	149-681	153-456
155-689	161-932	200-230	200-385
201-273	203-011	204-064	208-838
212-634	220-832	220-860	225-501
229-700	231-501	234-669	235-024
236-067	237-500	237-928	240-969
241-629	243-897	244-668	246-644
247-172	248-535	250-163	260-114
266-158	266-936	267-111	267-121
268-417	273-481	273-881	275-443

Molding - The "potting" or packaging of electronic equipment or the molding of a seal itself.

002-400	036-319	097-767	098-226
107-802	113-822	113-830	123-436
140-125	145-832	161-951	228-239
260-920	266-351	267-121	

Pressure Above 5000 psi

145-361	237-928	261-886	262-398

Pressure Gradient

042-477	085-318	116-389

Pressure-Velocity Relationships - An optimum combination of the two parameters at which the seal can effectively perform.

094-807

Radiation Effects

151-181	161-971	200-862	220-832
225-495	231-622	237-833	238-482
241-589	245-424	245-449	252-607
253-753	257-025	267-122	273-880
267-111	267-890		

Reciprocating Motion

027-728	029-114	074-089	097-258
103-928	130-882	151-181	162-882
238-020	221-803	267-145	270-746
272-431			

Reliability and Service Life

022-068	072-503	086-205	089-828
121-112	140-613	145-307	145-846
146-228	146-229	146-231	146-232
151-181	200-028	200-385	201-321
202-439	214-003	214-848	219-109
222-593	236-067	254-614	259-086
267-051	270-582	272-431	316-146
321-948	323-670		

Rotary Motion

007-347	011-080	011-722	011-915
019-627	025-808	025-809	034-171
042-477	043-686	048-377	048-808
054-206	056-036	059-125	060-141
068-495	077-374	082-045	086-205
087-221	094-807	094-809	094-444
100-053	100-836	106-777	108-632
110-636	116-938	119-437	121-294
123-771	125-957	134-918	143-735
143-970	153-356	154-494	156-901
160-124	162-475	200-028	201-047
201-739	202-438	202-439	205-179
207-505	213-568	219-109	222-071
222-852	225-213	232-073	239-180
241-830	257-228	259-574	258-535
262-196			

Selection - A choice has been made from a field of parameters based on test results or analytical studies.

078-224	086-205	094-749	118-047
120-620	160-124	163-020	201-739
206-707	232-073	236-067	241-831
245-377	249-387	274-085	

Shaft Eccentricity, Misalignment, Whirl, Vibration, Whip

007-347	121-112		

Shaft Surface Finish and Hardness

007-347	048-808	068-495	118-213
121-502	129-444	157-761	254-804
254-806			

Speed, High - (Greater than 4000 fpm)

085-114	086-205	100-836	108-632
119-437	155-689	201-047	219-109

Spring Loaded Types - A spring used to increase the contact pressure at the sealing interface.

011-080	025-808	025-809	034-171
043-686	048-377	060-141	073-961
094-807	094-809	123-771	134-918
143-970	153-033	202-438	202-439
205-179	207-505	214-003	222-593
225-213	232-073	236-837	258-535

Standards in Sealing

012-531	014-468	067-223	130-210
147-837	153-356	203-011	206-707
212-634	231-223	242-347	254-496
270-222			

Stresses in Sealing Elements

021-275	042-776	120-853	153-347
157-761	161-745	200-898	218-730
220-048	234-669	235-024	235-381
239-509	241-831	243-897	245-449
246-644	248-535	263-990	268-426
273-880			

Suspended Solids and Abrasives in Fluids

008-640	056-036	156-901	242-347

Temperatures Below $-65^{o}F$

025-907	074-322	090-148	101-331
106-754	125-427	128-326	147-856
214-003	222-071	229-186	234-109
237-833	238-832	240-357	242-285
243-454	248-647	259-086	265-443
266-719	267-501	267-890	272-493
274-085			

Temperatures Above $+1000^{o}F$

122-254	155-689	204-937	234-516
243-897	269-034		

Testing, Specification, Evaluation, and Performance

of Aircraft Seals (Canopies, fuel tanks, etc.)

002-400	009-494	011-484	013-311
015-012	028-761	030-417	047-671
048-310	055-695	058-142	067-223
070-011	078-224	085-568	094-590
102-156	105-671	106-719	106-754
107-274	121-502	123-436	123-771
125-427	130-772	130-882	131-091
145-399	157-761	200-381	200-382
201-085	202-439	225-495	235-052
241-830	241-831	247-172	254-803
257-940	262-718	263-457	268-381
273-212	302-156	323-114	

of Bearing Seals

011-915	013-714	025-808	077-374
143-970	201-047	209-179	228-320

of Cryogenic Seals

147-856	234-109	243-454	259-006

of Electrical Cable Entry Gland

041-362	067-223	077-374	088-545
090-148	094-590	094-807	106-777
134-497	134-918	140-117	140-118
145-239	200-230	204-064	205-020
214-269	229-700	238-482	

of Exclusion Sealing

121-294	209-179	254-496

of Jet Engine Seals

110-636	142-604	219-109	264-762

of Launch Vehicle Sealing

036-452	041-652	050-626	070-011
102-156	147-856	151-747	205-587
225-213	229-115	234-109	234-860
235-024	240-573	241-830	241-831
247-172	254-803	259-086	261-958
268-419	268-420	269-310	273-[illegible]
321-948	324-132	326-200	327-[illegible]

of Munition Sealing

025-907	036-456	036-564	094-9[illegible]
098-205	107-713	119-763	125-427
140-613	143-970	149-981	153-033
153-642	202-439	225-501	234-860
235-052	238-431	250-231	254-496
259-550	261-886	262-398	304-961
316-397			

of Pneumatic Equipment

025-907	036-456	105-671	121-502
145-307	145-361	145-846	151-747
153-033	153-642	235-024	248-647
259-550	260-920	268-419	

of Vacuum Seals

041-362	094-590	120-620	201-321
210-323	236-067	241-906	273-336

of Valve Seals

034-750	147-856	241-830	241-831
242-962	261-958	268-419	268-420
322-684			

Theoretical or Analytical Evaluation

042-477	085-318	150-973	157-761
200-381	200-382	200-862	205-810
206-374	221-803	229-700	234-669
235-024	237-500	239-509	243-897
248-535	267-111	267-890	268-426

Thermal Effects - The effect of heat upon the parameters affecting the sealing capabilities of the seal.

010-270	085-114	094-807	094-922
106-777	118-047	147-856	153-347
218-730	229-186	240-357	243-897
244-668	249-144	254-614	267-501

Wear and Wear Particles - A discussion of wear other than that caused by rubbing.

011-484	022-068	025-808	073-961
086-205	110-636	119-437	121-112
130-882	153-033	155-541	214-848
219-109	239-180	241-830	247-172
254-614	259-550	260-465	265-582

Wear Due to Rubbing

025-809	034-171	077-374	085-114
118-047	153-642	155-689	162-475
207-979	238-832	243-897	254-804

III. SEALING MATERIALS.

Asbestos and Asbestos Compositions

028-761	036-452	037-218	097-258
116-389	153-356	238-431	

Carbon and Graphites

008-640	025-808	025-809	027-587
077-374	085-114	086-205	094-674
094-807	094-809	119-437	123-771
134-918	143-735	145-833	155-689
201-024	258-535	261-563	261-564
266-936			

Cements, Lutings, and Adhesives

010-822	028-862	036-319	041-362
055-695	061-987	067-333	078-224
089-828	107-802	127-073	130-210
131-091	135-272	140-572	145-399
145-518	147-837	149-799	151-257
154-972	155-606	157-912	158-661
161-932	200-082	200-230	200-898
203-143	206-374	206-446	210-323
225-495	231-622	242-347	245-449
249-262	252-992	258-865	262-015
267-145	268-381		

Ceramics and Cermets

010-270	028-429	060-141	088-545
089-767	097-454	109-660	110-287
117-021	120-620	122-254	130-882
153-456	206-446	210-323	214-269
226-294	233-471	234-669	235-381
236-067	237-500	238-020	238-832
240-969	241-629	243-897	246-644
248-535	249-349	252-607	254-020
256-214	258-784	260-114	260-629
262-580	263-391	265-170	266-158
266-502	267-122	268-507	270-582
270-870	273-481		

Cork and Cork Compositions

116-389	121-339	206-773	263-306

Felt

048-377	121-294	153-033	214-848

Fibers

036-319	067-600	147-092	201-204
229-662	238-020	243-966	248-037
249-144	251-669		

Glass and Glass Fibers

010-270	027-587	060-141	082-914
082-915	082-916	097-754	120-853
123-771	134-405	149-681	151-747
153-347	153-456	153-642	156-311
201-321	214-082	216-838	218-730
219-139	220-698	222-684	225-213
226-195	226-521	234-516	235-381
235-681	238-164	238-482	240-357
241-906	242-744	243-685	244-668
245-318	245-424	248-535	250-399
252-607	257-978	263-640	265-582
266-246	268-381	268-416	271-561
271-643	271-918		

Leathers

036-564	153-642	236-837

Metallic Seals

Aluminum

002-991	003-133	012-531	037-218
037-705	041-362	050-626	077-374
078-224	121-339	145-239	150-474
153-642	201-835	206-224	214-003
218-038	226-521	236-837	240-969
248-535	256-678	258-865	258-893
267-122			

Brass

002-991	041-362	077-374

Bronze

048-808	073-961	162-882	258-865

Metallic Seals (Cont'd)

Ferrous Alloys

025-808	025-809	033-877	034-171
037-218	041-362	050-626	073-961
085-114	094-809	106-511	134-918
140-117	140-118	150-474	151-257
153-642	162-882	218-038	242-285
256-678	258-865		

Metal Fiber Composites

097-258	157-761	238-020

Plated Metal Coatings

027-728	068-495	073-961	075-568
085-114	119-437	153-642	218-038
241-831	256-214	265-582	267-145

Other Metal and Alloys

Used in Caulking

001-046	021-274	021-275	061-987
075-568	083-716	088-545	092-941
094-590	097-547	107-713	127-073
145-518	154-972	158-661	200-898
225-495	239-384	242-347	254-803
254-804			

Used in Face Seals

025-808	060-141	201-024	225-213
232-073			

Used in Gaskets

036-319	050-626	060-141	061-987
070-011	074-322	149-799	153-356
157-761	201-085	226-521	232-073
238-020	245-424		

Used in Hermetic Seals

054-206	082-914	083-716	092-941
094-590	097-547	107-713	108-720
134-772	134-405	153-347	201-085
218-730	225-495	226-521	234-721
249-349	254-020	256-214	256-649
260-114	265-170	270-870	

Other Metal and Alloys (Cont'd)

Used in Special Sealing Concepts (self sealing fuel tanks, injection seals, ceramic to metal seals, etc.)

001-046	010-270	021-274	021-275
028-429	074-322	088-545	089-767
094-922	101-331	107-713	109-660
110-287	112-953	117-021	120-620
122-254	145-518	149-681	149-799
153-456	155-606	158-661	210-323
225-495	234-516	234-669	235-381
239-384	240-357	241-629	246-644
249-349	252-607	256-214	257-978
260-114	265-170	266-158	269-404
270-582	273-481	274-112	

Organic, Synthetic Materials

Chloroprene (Neoprene)

029-114	029-801	037-093	037-218
050-626	074-089	090-148	119-330
123-912	125-427	127-094	128-326
140-117	140-118	140-125	146-232
151-747	162-481	163-020	206-773
212-866	221-461	231-622	251-669
254-496	258-893		

Elastomers (Buna-N, GR-S, Butyl, etc.)

029-801	029-951	037-218	047-671
068-495	080-889	089-198	089-199
094-749	101-097	116-389	119-330
121-339	123-912	128-326	142-079
147-837	147-990	150-973	151-181
151-257	155-606	157-912	163-020
204-016	204-065	207-979	214-848
225-495	229-186	238-020	240-573
241-589	245-424	247-750	248-647
249-262	249-387	249-542	251-130
254-496	254-806	256-218	257-025
261-561	261-562	261-563	261-564
263-298	263-457	265-443	267-145
267-501	267-890	269-310	273-880
274-085			

Organic, Synthetic Materials (Cont'd)

Fluoro-Elastomers (Kel-F, Teflon, Viton)

003-133	023-155	025-809	033-877
034-171	034-750	036-564	037-218
050-626	070-011	105-671	123-771
130-960	142-079	145-307	145-846
149-981	150-973	151-257	153-033
153-642	155-606	163-020	201-835
206-707	214-003	222-593	225-213
225-495	236-837	237-833	241-589
242-285	242-744	243-966	246-574
249-262	252-996	254-804	256-590
259-550	260-465	260-920	261-563
261-564	261-958	265-443	268-419
269-310	270-746	272-431	304-961

Plastics (nylon etc.) Used in

Gaskets

023-155	034-750	034-100	145-284
213-956	231-956	238-020	251-130
265-443	272-493		

Hermetic

012-531	030-136	037-705	067-333
228-239	232-964	234-721	258-784
263-722	266-351		

"O" Rings

101-097	131-094	237-833	241-589
265-443	268-381	269-310	

Caulking

028-321	049-645	055-695	067-333
078-224	097-767	098-226	106-511
107-802	123-436	130-661	145-518
158-661	159-156	160-402	160-452
228-239	251-669	268-381	

Other Seals

067-600	085-568	121-112	134-497
140-125	140-572	144-596	145-239
157-356	157-357	157-912	161-951
163-020	201-204	201-835	202-445
204-065	205-810	205-951	206-224
206-374	210-323	214-262	214-848
220-048	220-293	222-684	225-213
225-495	248-037	249-144	250-163
253-227	259-086	261-958	267-493
268-419	269-783	273-284	273-880
274-085	304-961	324-132	

Silicones

037-218	059-550	072-503	080-814
106-754	119-330	136-762	145-239
145-307	145-846	147-837	147-990
151-257	151-747	155-606	200-230
205-810	228-239	236-184	237-833
238-431	242-347	248-647	254-496
258-535	261-564	263-029	265-443
267-501	269-310	273-284	274-085

Rubber, Natural Used in

Gaskets

029-951	036-319	037-093	037-218
060-141	116-387	116-938	123-839
125-427	136-781	142-604	145-284
146-228	146-229	153-356	162-329
201-085	206-707	221-461	229-186
232-977	250-028	250-231	250-742
251-130	253-753	256-218	259-160
265-555	267-890	268-381	

Hermetic Seals

098-106	201-085	267-051

"O" Rings

010-979	011-484	078-234	098-106
123-839	131-094	140-116	142-604
145-357	145-832	161-745	162-472
163-020	229-186	232-977	237-833
240-838	241-589	248-647	249-387
250-028	254-607	254-806	256-590
260-920	261-561	261-564	267-501
268-381			

Other Seals

005-181	008-669	013-714	016-848
033-308	033-877	036-564	047-739
048-377	049-645	083-653	107-960
135-272	140-180	145-239	145-362
146-231	146-232	150-474	151-747
155-606	156-416	206-106	210-822
215-094	218-038	231-622	234-148
236-837	247-750	251-669	252-992
254-496	257-025	259-550	261-562
261-563	274-085		

Rubber, Synthetic (a specific synthetic compound is not stated) Used in

Gaskets

023-155	024-489	037-096	037-100
038-529	043-419	063-258	063-260
064-334	064-335	065-209	072-231
072-267	072-268	072-269	074-322
080-814	091-555	116-938	121-339
125-429	134-181	134-197	134-462
134-463	145-360	145-361	145-833
146-748	146-749	204-016	204-064
206-773	208-838	220-831	220-832
221-461	229-186	240-573	253-753
256-218	263-306	265-443	265-555

"O" Rings

027-728	074-089	129-444	131-094
134-197	134-493	145-307	145-846
146-748	146-749	161-638	161-694
163-020	203-158	204-064	205-587
207-016	207-190	207-191	207-538
207-979	213-568	229-186	232-977
240-838	249-387	250-828	254-607
254-806	260-920	261-561	261-562
261-564	265-443	267-121	269-310

Other Seals

022-068	024-652	028-327	036-428
072-232	072-503	073-961	082-306
099-741	111-249	113-822	113-830
118-597	127-079	131-091	153-033
157-912	160-124	162-475	200-082
200-230	203-011	204-065	205-020
205-478	205-810	207-035	210-822
211-410	212-634	212-866	214-003
231-622	236-184	237-928	257-025
259-550	263-029	267-111	273-284
274-085			

IV. APPLICATIONS

Aircraft Jet Engines

110-636	119-437	125-957	142-604
219-109	264-762		

Aircraft Sealing - (fuel tanks, airframes, canopies, etc.)

002-400	003-133	009-097	009-494
011-484	013-311	015-012	028-321
028-761	030-136	030-417	036-319
047-111	047-412	047-671	048-310
048-377	049-645	050-800	055-695
058-142	061-987	063-064	067-223
067-362	070-011	074-322	078-224
080-814	085-568	094-590	097-258
099-469	101-652	102-156	105-235
105-671	106-719	106-754	107-274
111-249	121-502	123-436	123-771
125-427	125-957	130-772	130-882
131-091	145-399	154-494	155-541
157-761	162-882	200-028	200-381
200-382	200-385	201-085	202-438
202-439	207-505	216-630	222-071
225-495	231-223	231-501	235-052
239-180	241-830	241-831	245-449
247-172	254-803	254-804	257-940
262-718	263-457	263-485	266-936
268-381	273-212	273-880	302-156
323-114			

Automobile Sealing

011-915	034-750	090-148	094-444
102-222	121-112	125-427	140-180

Battery Sealing

137-420	143-420	144-596	151-729
153-553	157-356	157-357	161-951
220-293	242-744	249-349	252-077
254-020	254-493	259-160	259-653
260-629	263-722	265-170	266-351
267-051	270-870		

Bearing Sealing

011-915	013-714	025-808	034-171
077-374	143-970	201-047	209-179
228-320			

Chemical Industry Sealing

003-133	130-960	251-130	269-783

Compressor, Sealing

262-718	267-121

Cryogenic Fluid, Sealing

147-856	222-071	234-109	237-500
237-833	238-832	242-285	243-454
253-227	259-086	265-443	266-719
267-501	272-493		

Engines, Internal Combustion and Diesel

022-068	056-708	119-330	123-912
134-493	145-832	161-638	161-694
161-745	203-158	207-016	207-035
207-190	207-191	207-538	213-956
221-803			

Electric Machine, Sealing

099-495

Electrical Cable Entry Gland, Terminals, etc.

018-810	019-627	041-362	047-739
056-036	067-223	077-374	088-545
090-148	094-590	094-807	094-809
098-106	106-777	134-497	134-918
140-117	140-118	140-125	143-735
145-239	145-357	156-901	200-082
200-230	204-064	205-020	206-106
214-269	226-521	229-700	234-516
238-482	274-085		

Electronic Tube Sealing

010-270	027-587	028-429	035-302
097-454	117-021	153-456	206-446
210-323	226-294	233-471	235-381
236-067	240-969	241-629	241-906
246-644	248-535	252-607	253-211
260-114	263-640	266-158	266-502
268-416	269-404	270-582	274-112
275-443			

Equipment - (electrical, electronic, etc.)

006-687	009-494	030-136	036-319
036-452	041-362	041-652	055-132
067-223	075-568	082-914	082-915
082-916	088-545	090-148	092-941
094-590	097-547	107-274	108-720
109-660	110-287	120-853	122-254
128-365	130-772	134-405	153-347
160-124	200-082	201-085	201-321
214-082	216-838	218-730	219-139
220-698	226-195	232-964	234-940
235-681	236-006	236-184	238-164
238-200	243-685	244-668	245-318
245-449	250-399	250-738	250-749
252-786	256-214	258-784	260-114
266-246	271-561	271-643	271-918
274-112			

Exclusion Sealing

083-653	121-294	209-179	254-496

Ground Equipment - (Armed Forces)

083-653	121-294	140-180	143-970
145-362	209-179	228-320	251-145
259-160			

Hose and Pipe Connections and Quick Disconnects

005-181	008-669	011-484	023-155
047-411	050-626	074-322	098-646
105-671	107-960	130-960	145-361
146-228	146-231	206-707	208-838
214-262	221-404	231-223	231-501
232-073	235-024	239-180	240-573
241-830	241-831	247-172	254-803
262-718	263-029	266-719	268-381

Hydraulic Equipment, Sealing

002-400	005-181	007-347	008-669
011-722	029-114	030-373	033-877
036-456	037-096	038-529	048-399
060-141	070-011	074-089	074-322
080-889	086-205	089-198	089-199
097-258	102-156	105-415	105-671
121-294	130-882	140-117	142-079
145-307	145-360	145-361	145-846
147-092	149-981	150-973	153-033
153-642	155-541	162-882	203-011
204-016	204-065	205-478	212-634
216-630	222-852	229-115	231-223
231-501	234-860	235-024	237-833
238-164	239-180	241-589	241-830
241-831	247-172	247-750	252-996
254-607	254-803	254-804	256-590
257-228	257-940	259-550	260-465
261-564	261-958	265-443	267-501
267-890	268-420	270-746	272-431
273-212	322-684		

Instruments, Precision - (Optical & Electrical)

002-787	046-159	048-808	070-011
078-688	082-044	100-053	222-684

Launch Vehicle Sealing - (Rockets, Missiles)

036-452	037-218	041-652	050-626
070-011	078-688	082-306	083-716
089-072	098-646	099-706	102-156
125-957	147-837	147-856	150-474
151-747	163-020	205-587	206-707
222-593	225-213	229-115	231-223
231-501	234-109	234-860	235-024
237-500	238-832	239-180	239-509
240-573	241-830	241-831	242-285
247-172	249-342	254-803	254-804
256-145	256-161	256-678	259-086
261-958	262-196	262-566	268-419
268-420	269-310	272-493	273-212
273-880	316-146	316-580	321-271
321-948	323-670	324-132	326-200
327-982			

Liquid Metal Sealing

121-461	259-574	266-768

Microwave Guide Sealing

019-627	056-036	077-374	094-807
094-809	106-777	134-918	143-735
156-901	262-580	263-391	267-122
268-507			

Mixers and Agitators, Sealing

151-729

Munitions Sealing

001-046	008-640	010-822	021-274
021-275	025-907	030-373	036-456
036-564	048-399	051-200	052-775
068-873	094-922	098-205	099-469
101-334	105-415	107-713	107-802
112-953	119-763	125-427	130-661
140-572	140-613	143-970	149-799
149-981	153-033	153-642	154-494
157-699	158-661	159-156	162-329
200-028	200-859	202-438	202-439
207-505	207-979	214-003	214-848
218-903	225-501	234-860	234-862
235-052	237-928	238-431	239-509

Munitions Sealing - (Cont'd.)

240-838	250-231	252-992	254-496
256-649	259-550	261-886	262-398
304-961	316-397	316-580	323-670
326-483			

Naval Ships - (Submarines)

036-428	056-708	103-928	113-822
113-830	118-597	123-912	134-497
140-116	140-117	140-118	145-239
145-284	145-360	145-832	145-833
146-228	146-229	146-231	146-232
147-092	160-402	160-452	162-472
162-475	162-479	162-481	201-024
206-703	203-143	206-773	212-518
212-866	213-564	214-292	215-094
218-038	251-669	254-607	254-614
256-218	258-893	260-385	260-920
261-561	261-562	261-564	265-555
267-121	322-684		

Nuclear Plant, Sealing

121-461	232-073	241-101	245-449
257-025	259-803	260-235	265-582
266-768			

Packaging - (munition, electronics - "potting," etc.)

012-531	028-862	030-136	037-705
067-333	092-941	097-547	097-767
098-205	098-226	100-488	123-436
156-311	156-416	157-911	157-912
158-530	161-971	200-862	201-201
201-202	201-203	201-204	201-273
202-445	203-833	205-810	205-951
206-224	206-374	212-661	220-048
221-461	228-239	228-846	229-662
229-707	231-223	231-622	234-721
239-384	241-564	241-565	243-454
245-377	246-034	247-172	248-037
249-144	253-753	256-214	257-411
260-526	263-298	269-783	

Petroleum, Petrochemical Industries Sealing

009-079	011-484	013-311	015-012
016-330	027-728	028-321	028-761
030-417	033-308	037-096	038-529
043-686	047-411	047-671	048-310
049-645	050-800	055-695	056-708
058-142	063-064	078-224	086-205
089-828	091-555	106-719	111-249
123-771	128-326	134-462	140-180
142-604	143-970	151-257	200-381
200-382	200-385	201-739	205-179
221-401	221-404	225-495	250-742
251-130	265-443	302-156	323-114

Pneumatic Equipment Sealing

008-640	025-907	030-373	034-750
036-456	058-978	059-125	060-141
074-322	085-318	105-415	105-671
121-502	129-444	145-307	145-361
145-846	151-747	153-033	153-642
162-726	207-979	216-630	235-024
237-833	240-838	246-574	248-647
259-550	260-920	261-561	261-562
261-563	261-564	265-443	267-121
267-501	267-979	268-419	

Pressure Range - 0-100 psi

011-722	027-728	077-374	086-205
094-807	098-106	134-918	140-116
145-239	145-357	145-360	201-047
204-937	214-269	234-109	234-940
250-742	259-550	263-029	265-582
272-493			

Pressure Range - 0-5000 psi

023-155	029-114	070-011	074-089
074-322	089-198	089-199	097-258
105-671	116-389	130-882	238-020
241-589	246-574	252-996	260-465
260-920	261-561	261-562	261-563
261-564	267-145	267-501	270-746
272-431			

Pressure Range - 0-10,000 psi

145-361	237-928	261-886	262-398

Pressure Vessel Sealing

140-116

Propeller (Marine) Shaft Seals (see also Naval Ships)

213-568

Pump Sealing

007-347	011-722	043-686	123-771
155-541	201-739	205-179	222-071
222-852	232-073	234-109	241-589
259-574	262-196	266-719	

Railway Vehicle Sealing

107-802

Satellite and Space Vehicle Sealing

231-223	231-501	237-405	240-357
241-830	241-831	247-172	254-803
254-804	259-086	266-351	267-051
273-284	273-880	274-085	

Temperature Range - 0 to 2000°F

243-897	269-034	273-481

Temperature Range - 0 to 1000°F

072-503	073-961	085-114	086-205
097-258	097-434	121-461	122-254
131-091	147-990	151-747	155-689
216-630	219-109	204-937	226-521
228-239	233-471	234-516	238-020
240-357	245-424	249-262	257-940
263-640	267-501	268-381	270-582

Temperature Range - -70 to 300°F

003-133	025-907	028-862	037-705
070-011	074-089	074-322	090-148
097-258	098-205	101-331	106-754
116-389	121-502	140-572	142-079
156-416	214-003	240-357	241-589
242-285	250-742	257-025	259-550
265-443	267-501	267-890	274-085

Temperature Range - -300 to 0°F

050-626	147-856	222-071	234-109
237-500	237-833	238-832	242-285
243-454	253-227	259-086	265-443
266-719	272-492		

Thermo-energy Converter - (electrical, couples, thermionic emission, etc.)

239-384	260-368	265-360	266-768
269-034	270-222	273-336	273-481
274-085			

Turbine Sealing - (Steam & Gas)

042-477	059-125	082-045	254-614

Vacuum Sealing

016-848	041-362	060-141	094-590
100-488	116-930	120-620	153-456
154-814	156-311	162-726	201-321
210-323	210-822	236-067	241-629
241-906	253-227	265-443	266-246
266-351	267-122	267-890	270-582
271-643	273-336		

Valve Sealing

002-400	034-750	041-961	147-856
222-593	232-073	241-830	241-831
242-285	242-962	259-086	261-958
266-719	268-419	268-420	322-684

SUPPLEMENTAL BIBLIOGRAPHY OF

ASTIA LITERATURE ON SEALS

1963 - 1969

The following abstracts and subject index supplements the preceding bibliography originally published under NASA contract NAS 7-102

Supp. AD-110 598
Boeing Aircraft Co.
Design Data For O-Rings And Similar Seals, by F.W. Tipton.
WADC Tech. Report 56-272, Contract No. AF 33(616) - 2867, November 1956

In order to fully utilize materials for seal design, a knowledge of the relationship between the physical properties of the materials and sealing efficiency is required. This report includes a literature survey on O-rings and seal design and functional tests of O-rings prepared from rubber compounded to have various physical properties. No definite relationship between seal life and physical properties was found. The inability of the O-ring to maintain sufficient internal pressure and to withstand mechanical conditions are believed to be the two reasons for seal failure.

Supp. AD-230 639
Boeing Airplane Co., Seattle, Washington
Design Handbook For O-Rings And Similar Elastic Seals, by F.W. Tipton. 65p incl. illus. (Proj. 7340; Task 73405) WADC TR 59-428) (Contract No. AF 33 (616) - 5722). October 1959. Unclassified Report.

This is a summation of the work done under Air Force Contracts AF 33 (616) - 2867 and AF 33 (616) - 5722. It is presented in the form of a handbook covering the mechanism of O-ring sealing, the relation of physical properties to sealing, the effect of cavity configuration, back-up brings and adverse mechanical conditions on seal life and the design of seals for specific systems. It also includes test procedures and a bibliography of published articles pertinent to O-ring seals.

Supp. AD-230 658
Boeing Airplane Company
Design Data For O-Rings And Similar Elastic Seals, by George E. Trepus, R.S. Roper and W.R. Hickman.
July 1959, 141 pp. illus. tables. ref. (Proj. No. 734C, Task No. 73405), (WADC TR 59-272 Pt IV) (Contract No. AF 33 (616)-5722).
Unclassified Report

This is a continuation of a study initiated under AF Contract 33 (616)-2867. The purpose of this study was to determine design criteria for O-rings, back-up rings and other elastomeric seals. A survey of current literature concerning seals and seal materials was conducted with an emphasis placed on seals and seal materials for use in environmental extremes. Tensile properties of a number of silicone and Viton compounds are given. The effects of environment and groove configurations on the sealing force of an O-ring were determined. Thermal expansion and compression modulus tests were performed on elastomeric materials at cryogenic temperatures. The

Supp. AD-230 658 (Continued)
functional tests included pneumatic seal tests at 400°F, hydraulic rod seal tests at 400°F using O-rings and back-up rings made from a variety of materials and in several configurations, static annulus tests at elevated temperatures using various groove configurations, and static seal tests using helium at cryogenic temperatures. Certain physical properties were found to be of value to the life of a seal under specific environmental and/or mechanical conditions.

Supp. AD-268 592
Goodrich High Voltage Astronautics, Inc., Burlington, Mass.
Feasibility And Design Study For Electrostatic Generators, by A.S. Denholm, J.E. Lavelle, F.J. McCoy and C. Coenraads. 76 p. incl. illus. (Proj. 3145; Task 61081) (WADD Tech. Report 61-105) (Contract No. AF 33 (616) - 7230). July 1961.
This report contains the results of studies made during the period April 1960 to December 1960.

The development of compact electrostatic generators depends upon the insulation of high field strengths. A study of the several media which can support such fields in an electrostatic generator shows that high vacuum is the best, particularly for applications in space.

Several forms of the electrostatic generator are analyzed. The most promising, for use in space, is not that having the best power-to-mass ratio, since simplicity and reliability are also considered.

The generator is discussed with regard to the space environment, into which it fits well. Of particular significance (because of the ceramic and metal construction) is the potential of the generator for operation at high temperature.

The electrical design of the machine depends strongly on the electric fields which can be supported and on the capacitance parameters. These are discussed at some length, in particular with regard to a 1 kw preliminary design model. Consideration is also given the problems of shaft sealing and the operation of bearings in space.

The report concludes with the design specification and test procedure for the preliminary design model.

Supp. AD-283 365
General Electric Co., Cincinnati 15, Ohio
Dynamic Shaft Seals In Space.
Space power and propulsion section first quarterly project status report for period ending July 15, 1962. Contract No. AF 33 (657) - 8459. 76 p. incl. illus.

The objective of this program is to acquire the techniques for sealing high speed rotating shafts under the operating conditions of high temperature liquid metals and vapors, the near-vacuum environments of space, and to provide long seal life.

Supp. AD-283 459
Puget Sound Naval Shipyard, Bremerton, Washington.
Evaluation Of Nitrile Rubber Seals Fluorinated By Quantum Incorporated, by W. Sergeant. Material Laboratories Report R-552, 5 p plus illus., (Project SF-013-13-01), Task 0907, BuSHIPS Ident. No. 51-907-5. March 1962.

One minesweeper controllable pitch propeller blade seal and four O-rings were fluorinated by Quantum Incorporated. This treatment produced surface cracks on the sealing lips of the blade seal, but no visible damage was present in the O-rings. O-ring weight and dimensions increased from their initial values. Sealing friction was reduced for the fluorinated O-rings tested in an oscillatory test apparatus.

Supp. AD-284 239
Quantum, Inc., Wallingford, Conn.
Development Of Treatments Producing Low-Friction Surfaces On Elastomers, by R.M. Reihsmann. Technical progress report No. 5 (Feb. 15, 1962 to May 14, 1962) prepared for the Bureau of Ships. Contract NObs - 84503, Project No. R-007-03-03, Task No. 1003, BuShips Ident. No. 47-1003-2.

Work is currently aimed at process optimization. High speed wear testing is being conducted to determine treatments which will produce a coefficient of friction less than .45 for a sustained period. The side effects resulting from these treatments are being evaluated concurrently.

Present work is concerned with the use of the solid $SF_3 \cdot SF_4$ complex as a co-fluordinating agent with SF_4 and the development of a usable fluorinated polymer of acrylic acid. Application of the latter material will eliminate the necessity of exposing the base rubber stock to a fluorinating atmosphere.

Supp. AD-290 611
School of Mech. Engineering, Oklahoma State University, Stillwater, Oklahoma
A Study In The Field Of Fluid Seals For High Speed Rotating Equipment, by R.E. Chapel, M.E. Schlapbach and L.E. Hall. Fluid seals Project Report No. 1 on work performed between February 2, 1959 and August 31, 1959 under Contract No. AF 34 (601)-5470, Purchase Request Nr OCNBE - 59-7. Sept. 1959.

This report summarizes a literature survey covering the many devices that are used as seals between machine components having relative motion. The reference sources that were used in the survey include:

1. Engineering Index,
2. Industrial Arts Index,
3. NACA Technical Reports,
4. ASME Index,
5. Fluid Power Index,

Supp. AD-290 611 (Continued)
6. SAE Index,
7. Theses Index,
8. National Conference on Industrial Hydraulics,
9. Bibliographies at ends of articles,
10. Article summaries in Technical Literature.

Supp. AD-298 335

Office of Naval Research, Dept. of the Navy, Washington, D.C.
Seventh Joint Army-Navy-Air Force Conference on Elastomer Research And Development. Held at U.S. Naval Radiological Defense Laboratory - San Francisco, Calif. - October 22-24, 1962. Report No. ONR-13, Vol. 1.

The Joint Army-Navy-Air Force Conference has met under rotating sponsorship at approximately two-year intervals as a continuing program of coordination. Need for this conference is based on a periodic re-examination of the intervening scientific and technical progress in the field and an appraisal of future critical weapons and systems requirements.

The published Proceedings taken together over the years provide the most complete available reference to the widely scattered military research and development program in elastomers and related materials.

Since the last meeting at Boston in 1960 sponsored by the U.S. Army Quartermaster Corps, substantial progress in several areas has been achieved in many laboratories sponsored by military agencies. New emphasis has been placed on the difficult problems arising from the new missile systems, the efforts to master penetration of outer space, the demands of high performance airplanes, the deep undersea environment and the tropical and polar land environments.

Supp. AD-298 336
Office of Naval Research, Dept. of the Navy, Washington, D.C.
Seventh Joint Army-Navy-Air Force Conference on Elastomer Research And Development. Held at U.S. Naval Radiological Defense Laboratory - San Francisco, Calif. - October 22-24, 1962. Report No. ONR-13, Vol. 2

The Joint Army-Navy-Air Force Conference has met under rotating sponsorship at approximately two-year intervals as a continuing program of coordination. Need for this conference is based on a periodic re-examination of the intervening scientific and technical progress in the field and an appraisal of future critical weapons and systems requirements.

The published Proceedings taken together over the years provide the most complete available reference to the widely scattered military research and development program in elastomers and related materials.

Since the last meeting at Boston in 1960 sponsored by the U.S. Army

Supp. AD-298 336 (Continued)
Quartermaster Corps, substantial progress in several areas has been achieved in many laboratories sponsored by military agencies. New emphasis has been placed on the difficult problems arising from the new missile systems, the efforts to master penetration of outer space, the demands of high performance airplanes, the deep undersea environment and the tropical and polar land environments.

Supp. AD-299 586
Frankford Arsenal, Research and Development Group, Pitman-Dunn Laboratories, Philadelphia 37, Pa.
Development Of Pressure Weld Pad Cartridge Seal by J.B. McCormick
Report R-1666, Feb. 1963; 37 pp. incl. tables and illus. OMS Codes 4110.16.8500.1.33 and 4110.16.8500.1.14

This report is a presentation of major aspects involved in obtaining a satisfactory seal by the application of the pressure weld process to Propellant Actuated Devices (PAD) for advanced flight vehicles. The objective is to determine the feasibility of this process for replacement of O-ring sealing in those environments where it is known to be inadequate.

During the preliminary phase it was decided to concentrate on two basic pressure weld types - extrusion weld and lap weld. Two different cartridge designs were developed as an approach to the solution of the problem. Leak detection tests and limited ballistic testing were performed under conditions similar to those used for the standard M91 cartridges. Results were evaluated and compared to standard cartridge performance characteristics. These investigations determined that this pressure welding process in feasible and adaptable to both present and future PAD devices. Further investigation is recommended to determine the broadest possible application of this process.

N64-16870
Monsanto Research Corporation, Dayton, Ohio
Evaluation of Elastomers as O-Ring Seals For Liquid Rocket Fuel and Oxidizer Systems
Technical Report September, 1961 - March, 1963
Carmen L. Bellanca, et al
Wright-Patterson AFB, Ohio, AF Materials Lab, September, 1963
Contract AF 33(616)-8483
ASD-TDR-63-496-; AD 422466

N64-18946
General Dynamics/Fort Worth, Texas
Engineering Research - Sealants - Composite Fiber Metal O-Ring - Evaluation of
M. T. Carroll and D. J. Pritchard 15 April 1964; 17 p
Contract AF33(657)-11214
ERR-FW-174; AD 436017

The report describes the evaluation of a silver impregnated composite fiber-metal "O" ring seal fabricated for testing as a fuel seal in the temperature range -100° to 1200° F. The dimensions of the seal were 6-1/2 inches O.D. by 6-1/8 inches I.D. by 3/16 inches cross-section diameter.

N64-19021
Nippon Oil Seal Ind. Co. Ltd., Tokyo (Japan)
A Study of Oil Seals For Reciprocating Motion
H. Kambayashi and H. Isheivata 1964, 12 p
Presented at the British Hydromechanics Research Association 2nd International Conference on Fluid Sealing, Cranfield, England - April, 1964 (Paper B-3)

N64-19022
Dowty Seals Ltd (Great Britain)
High Temperature Elastomeric and Metallic Seals For Aircraft Hydraulic Systems
B. C. Smithers, 1964, 30 pages. Presented at the British Hydromechanics Research Association Second International Conference on Fluid Sealing. Cranfield (Great Britain), April, 1964 (Paper C-3)

N64-26823
Lockheed Missile and Space Company, Sunnyvale, California
Adhesives, Seals, Gaskets and Polymeric Materials For Aerospace Applications: An Annotated Bibliography
Helen M. Abbott, comp. November, 1963, 202 p
Contract AF33(657)-10107
(SB-63-77; Report 3-06-63-2; AD443256)

N64-31232
Hughes Aircraft Company, Culver City, California - Aerospace Group
Development of Self Sealants and Sealants for Aerospace Vehicles
Third Quarterly Report
15 November 1962 - 15 February 1963
A. J. Tuckerman and J. F. York
Contract AF 33(657)-9171
P-63-11; AD419762

N64-12941
Hughes Aircraft Company, Culver City, California - Aerospace Engineering Division
Development of Organic Sealants For Applications at Very Low Temperatures
I. M. Zelman, R. I. Akawie and H. Harvey
June 1963, 95 p
NASA Contract NAS 8-2428
NASA CR52256; P-63-57

The most promising candidate lox-safe sealant was found to be Kel-F 800 resin reinforced with glass fabric and using a Kel-F 800 based hot-melt primer.

N64-15768
British Hydromechanics Research Association Harlow Information Department, England
Some References On Different Aspects of Lip Seals
October, 1963

An annotated bibliography on different aspects of lip seals is presented.

N64-17297
Hanford Atomic Products Operation, Richland, Washington
Test on Reactor Helium Seals - PRTR Inlet
D. H. White 2 March 1964, 18 p
Contract AT (45-1)-1350
HW-76008

N64-19779
General Electric Company, Cincinnati, Ohio, M & SD
Dynamic Shaft Seals In Space - Quarterly Report
No. 6, Period Ending October 15, 1963
Contract AF 33(657)-8469
AD-429211

N64-21990
General Electric Company, Cincinnati, Ohio, M & SD
Dynamic Shaft Seals In Space, Quarterly Report No. 7
Period Ending January 15, 1964
Contract AF 33(657)8469
AD-432064

N64-25124
National Bureau of Standards, Boulder, Colorado
Cryogenic Engineering Laboratory
Review of Static Seals For Cryogenic Systems
R. F. Robbins and P. R. Ludtke. Report from Journal of Spacecraft and Rockets, V. 1, No. 3, May - June, 1964, -. 253-259 (Sponsored by NASA)

N64-27305
General Electric Company, Schenectady, New York, ATL
Design Criteria For Zero Leakage Connectors For Launch Vehicles. Fundamen Seal Interface Studies and Design and Testing of Tube and Duct Separable Connectors. Final Report March - December, 1963
F. O. Rathbun, June 1964
Contract NAS 8-4012
NASA CR 56571

N64-32651
Battelle Memorial Institute, Columbus, Ohio
Development of a Long Life Contact Seal For a High Speed Rotating Shaft in Liquid Metal Dynamic Power Systems
Quarterly Progress Report, 1 April - July 1, 1964
S. J. Bashan, C. M. Allen and W. A. Glaiser 15 July 1964
Contract AF 33(657)-10961
AD449609

N65-22354
NASA - Marshall Space Flight Center, Huntsville, Alabama
Accelerated Compression Set Properties of Fourteen Elastomers
C. D. Hooper and J. T. Schell, 2 April 1965, 49 p
(NASA-TM-X-53232) CFSTI:

N65-17270
Aerojet-General Corporation, Azusa, California, SNAP 8 DIV.
SNAP 8 - Seals-to-Space Development Test Program, Volume 1: Visco Pump
R. L. Lissley, May 1964, 118 p
Contract NAS 5-417
NASA CR-54234

N65-26638
NASA - Lewis Research Center, Cleveland, Ohio
A Technique for Joining and Sealing Dissimilar Materials - Technology Utilization Report
Washington, NASA, June 1965
NASA-SP-5016

AD-462510
Aeronautics Division, Philco Corp. Newport Beach, California
Seals and Adhesives for Telescoping Pressurized Structures
W. Bandaruk, N. E. Quackenbush and G. C. Herzl, October 1964, 35 p, Publication No. U-2852
Contract AF 33 616 7775, Prog. 1368, Task 1368008

Various synthetic elastomer impregnated fabrics were screened for the flexible seal. The developed seal was demonstrated in a telescoping structure approximately 19 inches diameter by 45 inches long.

AD-460774
National Bureau of Standards, Boulder, Colorado
Elastomeric Seals and Materials at Cryogenic Temperatures
D. H. Weitzel, R. F. Robbins, P. R. Ludtke and Y. Ohori, Report for December 1962 to December 1963; March 1964, 141 p
Contract AF 33 (616) 61-04, Project 7343, Task 734005, MLTDR-64-50

This research deals with low temperature investigations of elastomeric polymers, with particular emphasis on their use as seals at cryogenic temperatures. Appendices include a state of the art review on low temperature static seals and a paper on the design of elastomeric seals for cryogenic temperatures.(Author)

AD-458101
Lockheed Aircraft Corp., Marietta, Ga.
C-130E Ramp Actuator Rod Seal Evaluation
G. L. Ready, 15 January 1964, 13 p, Report No. ER 6717
Contract AF 33-600-4359
IDEP 345 50 65-00-FH-01

AD-457393
M-O Value Co. Ltd., London (England)
Metal-Ceramic Seal Technology
D. R. Wallis, Annual Report No. 1, 25 November 1964, 19 pages
Report No. 14703C, Project RP 354
ACSIL 64 4986

This report covers the investigation into the formation of hard inter-metallic layers in some metal-ceramic seals. Some of the physical and mechanical properties of sintered bar, and strip rolled from it, have been determined and some success in deep drawing of the strip has been achieved. Rolling has been shown to modify the expansion characteristics of the alloy. (Author)

AD-406307
Frankford Arsenal, Philadelphia, Pa.
Determination of the Mechanism Governing the Inflow of Moisture Past a Rotary Seal - Theoretical Model
Hoffman, Bernard - January 1963, 89 pages

Efforts were made to determine a method for achieving an effective moisture barrier for fire control instruments. A description is presented of the probable mechanism governing the inflow of water and its vapor through rotary seals. Utilizing the derived mechanism, calculations are made to obtain engineering estimates of probable leakages through actual rotary sealing systems (systems employing rotary "O"-rings). A design of experiment is proposed for proving out the mechanism.

AD-410389
Rubber Lab., Mari Island Naval Shipyard, Vallejo, California
Seals for 12500 PSIG Air Systems, Progress Report No. 1
Bourn, W. S.; Holloway, J. M.; and Barrett, A. E., 26 June 1963, 7 pages
Project SF013-13-01, Task 907
MINS 28-11, Unclassified

The ability of Veton B "O" rings to seal 12500 PSIG nitrogen pressure was evaluated under dynamic and static simulated service conditions using spiral Teflon back-up rings. The static O-ring seals in the three test devices used leaked badly after 15 1/2, 19 and 46 1/2 hours, respectively. Swell of O-rings due to dissolving of nitrogen gas in rubber was only 4.4% measured after 4 days exposure to 12500 PSI nitrogen pressure, and no blisters were observed in the O-rings after this test.

AD-406719
Sperry Gyroscope Co., Great Neck, N. Y.
Manual of Metal-to-Ceramic Sealing Techniques
Johnson, C. W. and Cheatham, E. W., May 1963, 55 p
NA27-0001
Contract AF 30 602 2371, Project 5573, Task 557303

AD-288558
General Dynamics, Fort Worth, Texas
Pipe Thread Sealant Qualification
Young, W. M., 30 October 1962, 3 pages
Report No. FCTR-4488
Contract AF 33(657)7248

Discusses the use of teflon tape and other sealants for pipe threads at 5000 PSI and 350°F.

A64-11509
Sealing Spacecraft Against Vacuum
J. S. Islinger (International Telephone and Telegraph Corp., Research/Development, Nutley, N. J.)
Research/Development, Vol. 14, October 1963

Brief discussion of sealing techniques used for Mercury, Gemini, Apollo and Dyna Soar spacecraft.

A64-14336
Elastomeric Seals in Hard Vacuum
Carrell, Ted R. and Blair, Harold (Parker Seal Co., Culver City, Calif.)
Journal of Spacecraft and Rockets, Vol. 1, Jan-Feb 1964, p 113-115

Study of data on elastomeric seals in vacuums below the level of 10^{-6} mm Hg, torr.

A64-17408
Protection of Labyrinth Seals From Flexural Vibration
Alford, J. S. (General Electric Co. LJED, Cincinnati, Ohio)
(ASME, Aviation and Space, Hydraulic, and Gas Turbine Conference and Products Show, Los Angeles, Calif., March 3-7, 1963
Paper 63-AHGT-9
ASME Transactions, Series A- Journal of Engineering for Power, Vol. 86, April 1964

Discussion of failure in labyrinth seals. See Supp 061 on page 200 of final report for period April 1, 1964 to October 1, 1964 - Contract NAS7-102.

A64-18465
Review of Static Seals for Cryogenic Systems
Robbins, R. F. and Ludtke, P. R. (National Bureau of Standards, Cryogenic Engineering Laboratory, Property of Solids Section, Boulder, Colorado)
Journal of Spacecraft and Rockets, Vol. 1, May-June 1964, p. 253-259

Survey and evaluation of various demountable, low temperature static seals. Plastic materials as well as metallic materials are considered for both pressure actuated and temperature actuated seals.

A64-22266
RTV Silicones as Sealants and Adhesives for Cryogenic Applications
Robinson, D. E. and Schreihans, F. A. (NAA Inc., Space and Information Div. Eng. Dev. Lab., Downey, California)
National SAMPE Symposium on Adhesives and Elastomers for Environmental Extremes, 7th, Los Angeles, California, May 20-22, 1964
Society of Aerospace Materials and Process Engineers, 1964

Experimental determination of sealing efficiency of methyl-phenyl room temperature vulcanizing elastomers at cryogenic and elevated temperatures.

A64-22281
Fuel Resistant Silicone Sealants and Adhesives for Supersonic Aircraft
Miller, A. A., and Lamont, Peter, Dow Corning Corp., Midland, Michigan
National SAMPE Symposium on Adhesives and Elastomers for Environmental Extremes, 7th, Los Angeles, California, May 20-22, 1964
Soc. of Aerospace Materials and Process Engineers, 1964

Description of a class of fuel resistant silicone elastomeric materials which utilize room temperature vulcanization systems.

A64-22283
Elastomeric Sealants for Space Environment
Tuckerman, A. J., and Crepeau, P. C. (Hughes Aircraft Co., Culver City, California
National SAMPE Symposium on Adhesives and Elastomers for Environmental Extremes, 7th, Los Angeles, California, May 20-22, 1964
Society of Aerospace Materials and Process Engineers, 1964
Contract No. AF 33(657)-9171

Chemical self-sealant systems and pressurization sealants for assembly and repair of space vehicles in micrometeoroid environment.

A64-22284
Recent Developments in Using Elastomers for Static Cryogenic Seals
Ludke, P. R. (National Bureau of Standards, Cryogenic Engineering Laboratory, Boulder, Colorado)
National SAMPE Symposium on Adhesives and Elastomers for Environmental Extremes, 7th, Los Angeles, California, May 20-22, 1964
Society of Aerospace Materials and Process Engineers, 1964

Optimum flange design of compressed elastomeric "O"-Ring used for static cryogenic seals evaluated from test results - USAF, supported research.

A64-22286
Metal Clad Elastomeric Core Seals for Liquid Rocket Propellants
Spain, R. G., Bazzarre, D. F. and Wethers, J. C. (USAF, Systems Command Air Force Materials Laboratory, Wright-Patterson, AFB, Ohio
National SAMPE Symposium on Adhesives and Elastomers for Environmental Extremes, 7th, Los Angeles, California, May 20-22, 1964
Society of Aerospace Materials and Process Engineers, 1964

Results of efforts to combine the chemical resistance of certain metals with the sealing ability of elastomeric O-rings are reported.

A64-22290
Seal Design for Space Station Usage
George, J. C. and Silverman, B. (Lockheed Aircraft Corp., Lockheed-California Co., Burbank, Calif.)
National SAMPE Symposium on Adhesives and Elastomers for Environmental Extremes, 7th, Los Angeles, California, May 20-22, 1964
Society of Aerospace Materials and Process Engineers, 1964
Contract No. NAS 9-1563

Summary of information obtained during a study of static and dynamic seals for use during extended periods of time in space environment. Emphasis is on the Manned Space Station.

A64-24812
The Development and Testing of Elastomeric Materials for Fluid Sealing Applications
Blow, C. M. (Precision Rubbers, Ltd., Bagworth, Leics., England)
(British Hydromechanics Research Association, International Conference on Fluid Sealing, 2nd, Cranfield, England, April, 1964) Aircraft Engineering, Vol. 36, July, 1964

Discussion of laboratory test methods for elastomeric materials used in seals which come in contact with various types of fluids. Some of the newer elastomeric materials are briefly reviewed.

N65-31572
Tennessee University, Knoxville
Analysis of the Visco Seal. Part I
The Concentric Laminar Case
William K. Stair, Washington, NASA, Aug. 1965, 33 p refs. (Grant NsG-587)
NASA-CR-285

This report presents a theoretical analysis of the visco-type shaft seal when operating concentrically in laminar flow.

N66-12199
Tennessee University, Knoxville, Department of Mechanical and Aerospace Eng.
Theoretical and Experimental Studies of Visco-Type Shaft Seals
William K. Stair, 21 May 1965, 27 p refs. NASA-CR-68220; ME-65-587-3

This report outlines the progress made on an analytical and experimental investigation of the visco-type shaft seal.

N66-12572 AD-621011
Air Force Systems Command, Wright-Patterson AFB, Ohio, Foreign Technology Div.
End Sealing of A Shaft
L. S. Arinushkin; A. Yu. Polinovskiy, Soviet Patent No. 134986

The object of this invention is the end sealing of a shaft in the form of a ring turning together with the shaft.

N66-15358
NASA, Marshall Space Flight Center, Huntsville, Alabama
Self-Sealing Shields for Micrometeorite Protection
Erwin D. Funk, 30 Dec. 1965, 26 p refs. NASA-TM-X-53376

Self-sealing shields for cryogenic propellant tanks are investigated. The self-sealing shields consist of a hexcel covering containing a porous media.

N66-15545
Republic Aviation Div., Fairchild Hiller Corp., Farmingdale, N.Y.
High Temperature Hydraulic System Actuator Seals for Use In Advanced Supersonic Aircraft
J. Lee, 14 Oct. 1965, 76 p refs. NASA-CR-54496; FHR-2702-1

This report covers the first six-month period of a program to investigate seal materials and to design seals for high temperature hydraulic actuator application.

N66-17285
Aerojet-General Corp., Sacramento, California, Liquid Rocket Operations
Development of Large Size Bellows Face Type Seals for Liquid Oxygen and Oxygen Hydrogen Hot Gas Service At Moderate to High Pressures
E. Roesch; T. Rasternak, 10 Feb. 1966, 85 p refs. NASA-CR-54818; AGC-8800-16

This report describes the development program wherein a four-element dynamic

face seal was evaluated for an application requiring positive separation of cryogenic bearing coolant and hot gas in a turbopump.

N66-17442
Crane Packing Co., Motor Grove, Illinois
Fluid Seals
John W. Abar, Defense Department Proc. of Aerospace Symp. 1965 p 5-8

Dyanmic and static sealing device for door activators of manned orbital space stations were evaluated.

N66-18434
Aerojet-General Corp., Sacramento, California
Valve Lipseals M-1 Sleeve-Type Thrust Chamber Valve
J. K. Hetrick, C. G. Linn, 1 March 1966, 31 p refs. NASA-CR-54808; AGC-8800-33

The design investigation reported was undertaken in an attempt to develop a zero leakage dynamic cryogenic seal.

N66-19598
NASA, Lewis Research Center, Cleveland, Ohio
Gas Ingestion and Sealing Capacity of Helical Groove Fluid Film Seal (Viscoseal) Using Sodium and Water As Sealed Fluids
Lawrence P. Ludwig, Thomas N. Strom, and Gordon P. Allen. Washington, NASA, March 1966, 26 p refs. NASA-TN-D-3348

An experimental investigation was conducted on gas ingestion and sealing capacity of viscoseals using water and liquid sodium as the sealed fluids.

N66-20581
Marine Engineering Lab., Annapolis, Maryland
Self-Lubricated Seals Development For High-Pressure, Oil-Free Compressors
John R. Ward, Jan. 1966, 23 p refs., MEL-399/65, AD-627230

Continued progress in the development of self-lubricated piston and rod seals for oil-free, high pressure air compressor is described.

N66-22347
Pratt and Whitney Aircraft, East Hartford, Connecticut
Development of Compressor End Seals, Stator Interstage Seals, and Stator Pivot Seals In Air Breathing Propulsion Systems
B. M. Hawkins, C. A. Knapp, 20 Jan. 1966, 213 p ref., NASA-CR-54625; PWA-2752

Feasibility analyses were conducted on different concepts of compressor end seals, stator interchange seals, and stator pivot seals for multistage axial flow compressions used in advanced air breathing propulsion systems.

N66-25161
Northrop Corp., Hawthorne, California
Evaluation of Self-Sealing Structures For Space Vehicle Application
Philip S. D'Anna, Roger M. Heitz; Washington NASA, May 1966, 102 p ref.
Contract NASA-102, NASA-CR-485

Investigations were conducted to determine the feasibility of using a self-sealing wall for ensuring the sealed integrity of pressurized space vehicle compartments in a micrometeorite environment.

N66-29168
Bendix Corporation, Mishawaka, Indiana
Development Of A Hermetic Sealed Nitrogen Storage System for the Talos Rim-8E Fuel Pressurization System
W. B. Claxton, 9 Feb. 1966, 67 p refs. Contract NOw-65-0298C, BxM-5930, AD-631443

This report describes a method used to provide a 2150 psi nitrogen storage system that retains operational pressure for a minimum period of three years without interim servicing.

N66-32803
Tennessee University, Knoxville, Department of Mechanical and Aerospace Eng.
Analysis of the Visco Seal, Part II
The Concentric Turbulent Case
William K. Stair, Robert H. Hale, 28 June 1966, 99 p refs. (Grant NsG-587)
NASA-CR-76866; ME-66-587-7

In a theoretical analysis of visco seal operation in turbulent flow, the general approach taken was to modify the Navier-Stokes equations.

N66-33673
NASA Lewis Research Center, Cleveland, Ohio
Lubricants, Bearings and Seals
Edmond E. Bisson, William J. Anderson, Robert L. Johnson, Erwin V. Zaretsky, and Lawrence P. Ludwig; Selected Technical for the Petrol Ind. 1966, p 103-123 refs.

The influence of extreme environments on the operation of bearings and seals is described.

N66-39975
NASA Lewis Research Center, Cleveland, Ohio
Experimental and Theoretical Study of the Viscoseal
John Zuk, L. P. Ludwig, and R. L. Johnson; Washington NASA 1966, 84 p refs.
NASA-TM-x-52245 - Presented at the meeting on Development of Turbulent-Flow Bearings and Seals for Process-Fluid Lubricated Turbomachinery, Latham, N.Y., 1 Nov. 1966

Investigated were pressure patterns, gas ingestion, and sealing capacity of viscoseals. Oil, water, and liquid sodium were used as the sealed fluids.

AD-416864
Aeronautical Systems Div., Air Force Systems Command, Wright-Patterson AFB, Ohio
Development of High Temperature Seal - Lubricant Deposit Evaluation Techniques
March 1962, 1 p

IDEP Report No. 484 15 10 00G7 01; ASD TDR 62 253 (Unclassified)

AD-429960
Saibel (Edward), Annapolis, Maryland
A Theoretical Investigation of Leakage Through Rotary Shaft Seals (Final Rpt.)
Edward Saikel and Frederic A. Lyman, 28 Nov., 61, 29 p

Contract N 161S1590, Proj. S F013 07 01, Task 3724, MEL A 850489 (Unclassified)

AD-432545
Boeing Company, Seattle, Washington
External Surface Seal Developments Tests
James H. Covey; 20 Dec. 63, 91 p; Document No. D2 80876; Contract AF 33 657 7132 (Unclassified)

AD-433277
Boeing Company, Seattle, Washington
Dual Seal Development Test Report
V. C. Dunham, 26 Dec. 1963, 49 p
Document No. D2 81190; Contract AF 33 657 7132 (Unclassified)

AD-438100
Aerojet-General Corp., Sacramento, California
Combustion Chamber Sealing Diaphragm
11 Apr. 1963, 5 p; Report No. 63 274; IDEP 501 78 00 00A6 02 (Unclassified)

AD-438119
Aerojet-General Corp., Azusa, California
Oxidizer Seal Failure Investigation
R. T. Dahl, 19 June 1963, 9 p; Report No. 63 391; IDEP 345 30 50 00A6 01 (Unclassified)

AD-443294
Aerojet-General Corp., Sacramento, California
Compatibility of EY105 With Propellants - Water and Lube Oil
D. A. Brown; 18 March 1964; 5 p; Report No. 64 215; IDEP 345 00 00 00A6 02 (Unclassified)

Specimens of EY105, a carbon-graphite material, were immersed in test fluids, and the weights checked periodically for 48 hours. The specimens were then placed in air-tight containers for 96 hours, followed by drying at 400F for hours. The largest weighted gains were obtained in specimens immersed in N204. N204 was also the most difficult to remove by baking..

AD-443964
Republic Aviation Corp., Farmingdale, N.Y.
Metallic Boss Seal Evaluation and Test Program
Final Report, 1 May 1960-1 Oct 1961; S. Cannizzarc and W. E. Mayhew, Oct 1961, 57 p.; Report No. 319 1 ARD 682 629; Contract AF 33 616 7297; Proj. 8128; Task 812807 61085 8; ASD TR61 486 (Unclassified)

This final report describes the results of a 17-month program devoted to the testing and evaluation of three types of metallic seals for use in AND10050 bosses.

AD-446765
Defense Documentation Center, Alexandria, Virginia
Seals Compatible with Liquid Rocket Propellant Including IRFNA - A DDC Report Bibliography
Bibliography for 1959 - May 1962, 7 p.; Report No. ARB 10133 (Unclassified)

AD-448155
Aerojet-General Corp., Sacramento, California
Materials for Lubricating and Sealing Propellant and Hot Gas Systems
E. F. Kerttula, Jr., 26 May 1964, IV Report No. 64298; IDEP 484 00 00 00A6 05 (Unclassified)

This report is intended to aid in the selection of lubricants and thread sealants for gasses and liquid O_2/H_2 service.

AD-449708
National Research Corp., Cambridge, Mass.
Study of Sealants for Space Environment
Summary Report, 4 Dec. 1959-20 Dec. 1960; Imre Farkass and Edward J. Barry, 20 Dec. 1960, 44 p.; Contract DA 19 020 5060RD5097, Proj. NRC R43 1 139 (Unclassified)

This report covers a one-year program of theory and experimental concerning the development of sealants capable of maintaining internal pressurization of a system subjected to space conditions.

AD-461695
Frankford Arsenal, Philadelphia, Pennsylvania
A Dynamic Seal Tester for Propellant Actuated Devices
Technical research report - Oct 1962-Jan 1964; Aynew E. Larsen, Apr 1965, 37 p
Report No. R-1753; SEG TR-65-11 (Unclassified)

This report describes a study of the conception, design, fabrication, and preliminary evaluations of a test fixture to measure the static and dynamic performance of seals applicable to propellant actuated devices.

IIT Research Inst., Chicago, Illinois.
Analytical Techniques for the Design of Seals for Use In Rocket Propulsion Systems, Volume I, Static Seals
Final report, 1 Feb 1962-15 Nov 1963; Paul Bauer, Myron Glickman, Frank Iwatsuki; May 1965, 270 p; Report No. IIT RI-K6017-Vol. 1; Contract AF04 611 8020; Proj. 6753; Task 675304; RPL TR-65-61-Vol. 1 (Unclassified)

This report documents investigations that were conducted to develop analytical design techniques for static, sliding, and rotating shaft seals as applied to rocket propulsion systems.

AD-464959
IIT Research Inst., Chicago, Illinois
Analytical Techniques for the Design of Seals for Use In Rocket Propulsion Systems, Volume II, Dynamic Seals
Final report, 1 Feb 1962-15 Nov 1963; Paul Bauer, Myron Glickman, Frank Iwatsuki; May 1965, 331 p; Report No. IIT RI-K6017-Vol. 2; Contract AF04 611 8020; Proj. 6753; Task 675304, RPL TR-65-61-Vol. 2 (Unclassified)

This report documents investigations that were conducted to develop analytical design techniques for static, sliding, and rotating shaft seals as applied to rocket propulsion systems.

AD-465-492
Douglas Aircraft Co., Inc., Santa Monica, California, Missile and Space Systems Div.
Cryogenic Seals Vibration and Shock Tests
R. E. Karleen, 18 Feb 1965, 9 p; Report No. TM-DSV4-ME-R4943; IDEP 345.00.50.00-D7-01 (Unclassified)

AD-601338
General Electric Company, Cincinnati, Ohio
Dynamic Shaft Seals In Space
Quarterly Report No. 8; 15 April 1964, 23 p; Contract AF33 657 8469 (AD 432064-No 1)

AD-603655
General Electric Company, Cincinnati, Ohio
Dynamic Seals In Space
Quarterly Report No. 9; 15 July 1964, 53 p; Contract AF33 657 8469 (AD-601338-No. 8)

AD-622958
Aerojet-General Corp., Sacramento, California
Hydrostatic Combustion Seal Demonstration Feasibility Program
Progress Report No. 1 for 7 Mar-15 Sept 1965; 15 Oct 1965, 86 p; Report No. 10784-Q-1, 298IT; Contract AF04 611 90784 (Unclassified)

The major objective of the program is the design of a seal for use in an oxidizer-rich atmosphere at 1200F and 3000 psi with a turbine shaft operating speed of 40000 rpm.

AD-609174
Air Force Aero Propulsion Lab., Research and Technology Div., Wright-Patterson AFB, Ohio
Research and Development of A Friction-Controlled Face Seal
Progress report - Robert J. Smith, Sept 1964, 30 p; Proj. 8128; Task 812807; APL TDR64 108

This investigation was undertaken to develop a dynamic seal for space power equipment.

N67-11171 AD 638972
Air Force Systems Command, Wright-Patterson AFB, Ohio, Foreign Tech. Div.
Operation of Cascade Contact Seals
A. M. Yevtushenko, 31 August 1966, 25 pp., Refs., (FTD-HT-66-281; 77-66-62243)

Translation of Russian research: operation of cascade contact seals.

N67-13998
Atomic Energy of Canada, Ltd., Ottawa (Ontario)
Performance Testing of Conical-Face High Pressure Rotary Shaft Seals
R. D. Watson, November 1965, 36 pp., Refs. (AECL-2551)

The performance characteristics of a special type rotary shaft seal have been determined at pressures up to 1000 lb ft/in^2 gage and speeds at 1900 rev/min. The seal is made up of two rings, with one flat face ring in contact with the very slightly cone-shaped face of the other ring. Leakage rates for the seal were studied.

N67-17506
NASA Lewis Research Center, Cleveland, Ohio
Screw-Seal Performance in Vacuum Environment
J. C. Hudelson and L. P. Ludwig, Washington, NASA, February 1967, 16 pp., Refs. (NASA-TN-D-3842)

Non-contact seals are being proposed for use in space power systems. The non-contact seals offer reliable operational lifetimes of over 10,000 hours. The leakage rate of an ether lubricant through a non-contact slinger pump and screw-seal assembly to a simulated space environment was measured.

N67-19126
Pratt and Whitney Aircraft, East Hartford, Conn.
Development of Main Shaft Seals for Advanced Air Breathing Propulsion Systems
Semiannual Report No. 2, 1 January-30 June, 1966
R. L. Thomas, A. J. Parks, and R. M. Slayton, 9 September 1966, 130 pp. (Contract NAS3-7609) (NASA-CR-82453; PWA-2879)

Detailed analytical studies of all seal configurations to be tested are continued. Approval was received for the following seal concepts: orifice compensating hydrostatic face seal and face contact seal with bellows secondary.

N67-22711
Royal Aircraft Establishment, Farnborough (England)
Temperature Limitations of Polymeric Materials. Part 6: Elastomers
G. Wood, April 1964, 40 pp., Refs. (RAE-TN-CPM-52)

Elastomers find many applications in aircraft, e.g., seals (static and dynamic), sealants, gaskets, O-rings, and different properties are significant for particular cases. This report reviews the properties of the more important commercial elastomers and sealants.

N67-23037
British Hydro Mechanics Research Association, Cranfield (England), Information Dept.
References on or Related to Screw-Type Rotary Shaft Seals
March 1966, 14 pp., Refs. (BIB-22)

A compilation of abstracts related to screw-type rotary shaft seals of interest to hydromechanics is presented. Areas covered include screw viscosity pumps, the sealing action of a viscous fluid, helical groove seals, high speed lubricating oil pumps, and screw seals in laminar and turbulent operation.

N67-25487
Pratt and Whitney Aircraft, East Hartford, Conn.
Development of Compressor End Seals, Stator Interstage Seals, and Stator Pivot Seals in Advanced Air Breathing Propulsion Systems
Semi-annual Report, 1 January-30 January 1966
R. M. Hawkins, 20 July 1966, 249 pp., Refs. (Contract NAS3-7605) (NASA-CR-83786; PWA-2875; SAR-2)

The design of compressor end seals, stator interstage seals and stator pivot seals is discussed in detail. One-side floated shoe seals, two-side floated shoe seals, and thin strip seal designs are considered.

N67-25842
NASA Lewis Research Center, Cleveland, Ohio
Improving Performance of Face Contact Seal in Liquid Sodium (400° to 1000°F) by Incorporation of Spiral-Groove Geometry
L. P. Ludwig, T. N. Strom, G. P. Allen and R. L. Johnson, Washington, NASA, May 1967, 39 pp., Refs. (NASA-TN-D-3942)

Conventional face contact seal performance was improved by incorporation of spiral-groove geometry. In comparison with conventional face contact seals, seals with spiral grooves had negligible leakage. The wear and contact patterns indicated that the spiral-groove seal operated with separation of the sealing surfaces, which is necessary for long life.

N67-26213
Atomic Weapons Research Establishment, Aldermaston (England)
Loads to Deform Elastomer O-Seals (Toroidal Sealing Rings) in Grooves and Between Plain Flanges

The loads necessary to deform O-seals in various types of grooves and between plain flanges have not been extensively investigated. This report describes and presents the results of 210 tests in the form of a nomogram.

N67-31128
Pratt and Whitney Aircraft, East Hartford, Conn.
Development of Main-Shaft Seals for Advanced Air-Breathing Propulsion Systems
Semiannual Report, 1 July-31 December 1967, 85 pp. (Contract NAS3-7609) (NASA-CR-85908; PWA-2996; SAR-3)

Four main-shaft seals for advanced gas turbine applications have been designed. The seals studied are an orifice-compensated hydrostatic face seal, an extremely pressurized hydrostatic face seal, a carbon-face-contact seal with a bellows secondary seal, and a carbon-face-contact seal with a piston-ring secondary seal.

N67-31151
Pratt and Whitney Aircraft, East Hartford, Conn.
Development of Compressor End Seals, Stator Interstage Seals, and Stator Pivot Seals in Advanced Air Breathing Propulsion Systems
Semiannual Report, 1 July-31 December 1966
R. M. Hawkins and A. H. McKibbin, 20 January 1967, 94 pp., Refs. (Contract NAS3-7605) (NASA-CR-85891; PWA-2995; SAR-3)

The basic design concept for the "OC" diaphragm thin strip seal is discussed. The final design of the one-side floated shoe seal is discussed in some detail. The final designs of the vane pivot seals and test rig are also discussed.

N67-31390
Tennessee Univ., Knoxville, Dept. of Mechanics and Aerospace Engineering
Theoretical and Experimental Studies of Visco Type and Buffered Shaft Seals
Semiannual Progress Report, 15 October 1966-15 April 1967
William K. Stair and C. F. Fisher, Jr., May 1967, 25 pp., Refs. (Grant NsG-587) (NASA-CR-85893; ME-67-587-9)

This report outlines progress made on the investigation of the viscoseal and the buffered seal. The experimental performances of fine grooved shaft viscoseal are presented.

N67-35568
NASA Marshall Space Flight Center, Huntsville, Alabama
Conference on Design of Leak-Tight Separable Fluid Connectors, Vol. I
25 March 1964, 340 pp., Refs. Proc. held 24-25, March 1964 (NASA-TM-X-60234)

Contains twelve papers given at the conference.

AD-379 053L Fld. 21/5
Pratt and Whitney Aircraft, East Hartford, Conn.
Component Propulsion Program for Future High-Performance Strategic Aircraft. Vol. XVI Seals
Semiannual Report No. 3, 1 August 66-31 January 67
R. C. Barningham, Contract AF33(657)-14903, Proj. AF-61139A

Confidential Report

N68-11025
Theoretical and Experimental Studies of Visco Type and Buffered Shaft Seals
W. K. Stair, C. F. Fisher, T. A. Arehart, Oct. 1967, Tennessee University

The experimental performance of three grooved shaft viscoseals is presented and the theory developed under this project is compared with independently obtained data. These results indicate that the theory advanced provides a good design estimate for grooved shaft viscoseals in both laminar and turbulent regions.

N68-12654
High Temperature Hydraulic System Actuator Seals for Use in Advanced Supersonic Aircraft
J. Lee, Oct. 1966, Fairchild Hiller Corp.

This report covers the third six month period of a program to investigate seal materials and to design seals for high temperature hydraulic actuator application. Various leakage and endurance tests were carried out.

N68-13547
Supersonic Transport Lubrication System Investigation (Semi-annual Report)
C. Hingley, H. Southerling, L. Sibley, May 1965, 215 p., SKF Industries

Ball bearings and bellows face seals for use in Mach 3 aircraft gas turbine engine mainshafts and suitable test rigs to simulate engine operating conditions, have been designed and manufactured. The initial phases of a development program are underway to determine the most suitable combination of presently available materials and lubricants to permit operation of these bearings and seals under typical engine load and speed conditions.

N68-14172
Development of Mainshaft Seals for Advanced Air Breathing Propulsion Systems,
Phase I, Final Report, A. Parks, R. McKibbin, C. C. W. Ng and R. Slayton, Aug. 1967, Pratt and Whitney Aircraft

Four mainshaft seals for advanced gas turbine applications were designed, tested, and analyzed. The seals studied were a rubbing-contact face seal with piston-ring secondary seal, a rubbing-contact face seal with bellows secondary seal, an orifice-compensated hydrostatic seal and an orifice compensated externally pressurized seal. Results are included.

N68-17610
Spiral Groove Face Seal Concepts; Comparison to Conventional Face Contact Seals in Sealing Liquid Sodium (400° - 1000° F)
T. N. Strom, L. P. Ludwig, G. P. Allen, and R. L. Johnson, Nov. 1967, NASA Lewis Res. Center

Conventional face contact seal performance was improved by incorporation of the spiral-groove geometry. Both conventional face contact seals and seals with spiral grooves were used to seal liquid sodium. In comparison with the conventional face contact seals, the seals with spiral grooves had negligible leakage.

N68-18124
Bearings & Seals for Cryogenic Fluids
H. W. Scibbe, Nov. 1967, NASA Lewis Res. Center

Bearings and seals in rocket engine turbo-pumps operate directly in the cryogenic propellant. Material compatibility and wear are the important factors for face contact seals in these cryogenic fluids.

N68-23409
Performance of Inerted Lubrication Systems for Turbine Engines
Johnson, Loomis and Ludwig, Mar. 1968, NASA Lewis Research Center

Inerted lubricating systems containing 125 mm ball bearings and 6.33 inch diameter face contact seals were operated in simulated turbine engine sumps at speeds to 14000 rpm and temperatures to 800° F. A persistent problem encountered was wear and leakage of the shaft seals. Additional experimental studies and analysis identified seal thermal deformation as a major factor in seal wear and leakage.

N68-24502
Window Seals for Ultrahigh Vacuum Use
R. W. Roberts, J. F. Harrod, H. A. Poran, May 1967
G. E. Company, R & DC, Schenectady, N.Y.

Sealing techniques are described for joining windows of lithium fluoride, magnesium oxide, and quartz to Pyrex glass, FN glass, quartz and silver. These seals will withstand repeated bakeouts at 300° to 350° C and can be used in ultrahigh vacuum systems.

N68-25416
Convertive Inertia and Gas Ingestion Effects on Flow Regimes of the Viscoseal Theory and Experiment
J. Zuk, T. N. Strom, L. P. Ludwig and R. L. Johnson; May 1967, NASA Lewis Res. Center

An experimental and visual investigation of the viscoseal operating regions was conducted. A mathematical model was formulated for the homogeneous fluid region which resulted in an evolution of a modified Reynold's number based on two characteristic lengths.

N68-25471
The Turbulent Visco-Seal: Theory and Experiment
W. K. Stair and R. H. Hale
Tennessee University

Experimental evaluation of the laminar visco seal for a number of seal geometrics has shown that the analysis of Boon and Tal is in better agreement with experiment than any other analytical approach. Experiment tests with 10 different seals confirmed the utility of both the theoretical analysis and the method of approximation of the experimental constants.

6843
Theoretical and Experimental Studies of Visco Type and Buffered Shaft Seals
W. K. Stair, C. F. Fischer, E. A. Picklesimer, May 1968
Tennessee University

Performance results of four grooved-shaft seals are presented. These results indicate that the theory advanced provides a good design estimate for grooved shaft visco seals in both laminar and turbulent regions.

N68-27529
Convertive Inertia and Gas Ingestion Effects on Flow Regimes of the Viscoseal - Theory and Experiment
Zuk, Strom, Ludwig, Johnson, May 1967
NASA Lewis Research Center

An experimental and visual investigation of the viscoseal operating regions was conducted. A mathematical model was formulated for the homogeneous fluid region which resulted in an evolution of a modified Reynold's number based on two characteristic lengths.

N68-29320
Development of Compressor End Seals, Stator Interstage Seals, and Stator Pivot Seals in Advanced Air-Breathing Propulsion Systems
R. H. Hawkins, A. H. McKibbin, and Charles C. W. Ng 20 Jan. 1968

Tracking analyses and thermal analyses of the O C diaphram thin-strip seal, the semirigid interstage seal, and the flexure-mounted shoe seal are discussed. The tracking analysis includes the effects of seal tilt angle, engine speed (take-off, cruise, and idle conditions), and natural frequencies of the seal.

N68-32955
Evaluation of Sliding Seals for Operation in Dry Helium
A. A. Abbatiello and F. H. Neill, June 1968
Oak Ridge Nat. Lab.

An evaluation was made of elastomer O-ring seals and chevron seals for piston ring and shaft seal service in dry helium. Conditions simulated were piston load, rubbing velocity, operating temperature (up to 350° F) and temperature cycles.

N68-33027
Supersonic Transport Lubrication System Investigation, Phase 2
W. L. Rhoads, May 22, 1968
SKF Industries

Ball bearings and bellows face seals for use on Mach 3 aircraft turbine engine mainshaft have been tested in screening tests using jet lubrication and inert gas blanketing. Bearing operating temperatures up to 700° F have been explored under typical engine load and speed conditions with the seals exposed to 1200° F air and to a pressure differential of 100 psi.

N68-33263
Pressures in Sealing Grooves of Hydraulic Assemblies
P. I. Blandov
News of Inst. of Higher Learning Aeron.
Eng. No. 1, 1966, Oct. 2, 1967 p. 10-20

This article provides calculations of pressures in the sealing groove of hydraulic assemblies and furnishes these and other conclusions: (1) high pressures in the groove promote accelerated wear of seals; seals are pulled into the clearances between fittings, impeding the flow of vapor; (2) unloading in groove causes cavitation.

N68-34280
Laminar Flow Analysis of Plane Disk Slinger Seal
R. L. Rosenthal and Eli Reshoto Aug. 1966
Case Institute of Technology, Cleveland, Ohio

The flow and temperature fields are considered in the region between a rotating and stationary disk. The governing equations are changed by a radial transformation into ordinary differential equations. Distinctions between regimes of large and small Reynolds numbers are clarified and the equations are solved accordingly.

N68-36254
Investigation of Positive - Type Shaft Seals - Interium Report
R. R. Hammond Sept. 25, 1968
Rocketdyne, Canoga Park, California

A series of new type seal concepts were generated and three of the most promising were detailed for fabrication and testing to evaluate the designs for future turbo-pump applications. Descriptions of the various concepts, basis for the final selections of the seals for evaluation, and results of testing are included.

N68-37608
Development of Compressor End Seals, Stator Interstage Seals, and Stator Pivot Seals in Advanced Air Breathing Propulsion Systems
R. M. Hawkins and A. A.H. McKibbin 19 July 1968
Pratt and Whitney Aircraft

The compressor end and interstage seals are described and procurement of the seals is discussed. There is also a brief discussion of the effects of heat shielding and heating on the one-side floated-shoe interstage seal. In addition, procurement of the stator vane pivot seals and modifications to their associated test apparatus is discussed.

N69-10055
Performance and Analysis of Seals for Inerted Lubrication Systems of Turbine Engines
R. L. Johnson, W. R. Loomis and L. P. Ludwig
NASA Lewis Research Center

An inerted lubricating system incorporating a 125-millimeter ball bearing and 6.33 inch (161-mm) diameter face contact seals was operated for short

duration screening runs in a simulated turbine engine sump at speeds to 14,000 rpm and bulk oil temperatures to 500° F. A persistent problem encountered was wear and leakage of the shaft seals.

N69-10749
Study of Dynamic & Static Seals for Liquid Rocket Engines. Selected Studies from the Technology of Mechanical Face Seals
G. E. Company, Research & Development Center
May 1968

Summarized are technical accomplishments achieved under a program designed to advance the technology of dynamic and static seals and to provide a basis for formulating design criteria for seals in future liquid rocket engines. Program tasks included non-Newtonian and Newtonian sealing concepts, a non-Newtonian correlative model, seal geometry effects and seal design theory.

N69-11317
A Simplified Constitutive Equation for Two-Dimensional Flow
Earl Blaine Adams 1967
Tennessee University

A simplified constitutive equation for viscoelastic fluids was analyzed for its ability to predict stress behavior in a 2-dimensional flow geometry. The stress equations for flow between converging or diverging flat plates were derived. Simple assumptions for velocity profiles and flow direction were made in order to study the character of the stress equations.

N69-12582
Bearing and Seal Technology
J. Anderson and P. Ludwig
N.A.S.A. Lewis Research Center

Improvements in seals and bearings used in aerospace technology are discussed and considered for possible application in electric power generating equipment. Face seals, visco seals, and spiral seals are briefly considered for reducing leakage and axial length of machines.

N69-14512
Design Criteria for Zero Leakage Connectors for Launch Vehicles. Tube Connector with Superfinished Seal Vol. 4 Final Report
B. Weichbrodt Jan. 1968
G. E. Co., Schenectady, N.Y.

The development of a new type of tube connector which is reusable and seals without a gasket is described. Instead of a gasket it uses a superfinished surface seal, and is designed for 6000 psi service at temperatures ranging from -320° to 700° F. Extensive tests were conducted and show that leakage is less than 3×10^{-6} atm-cc/sec for helium.

N69-14513
Design Criteria for Zero Leakage Connectors for Launch Vehicles. Advanced Leakage Tests, Vol. 3 Final Report
J. P. Laniewski July 20, 1967
G. E. Co. Schenectady, N.Y.

Sealing investigations were conducted in the temperature range of -321° to 400° F and at pressure differentials to 6000 psi. The some of the determinations which were made: (1) the amount of gasket stress required to affect a seal, (2) the gasket stress required to maintain a seal, (3) when rubber or plastic seals should be used instead of metal seals.

N69-14514
Design Criteria for Zero Leakage Connectors for Launch Vehicles. Mathematical Model of Interface Sealing Phenomenon, Vol. 2 Final Report
L. G. Gitzendanner, J. K. Hawley, H. W. Moore, F. O. Rathbun, Jr. and J. Wallach Nov. 9, 1967
G. E. Co. Schenectady, N.Y.

A procedure for the calculation of leakage flow through the seal interface of a fluid connector was developed and evaluated. The following procedures were followed in the calculation: (1) surface roughness measurements were made, (2) surface statistics were calculated, (3) the surfaces were generated mathematically and pressed together mathematically, (4) the interfacial gap map was generated, (5) a flow map was generated, (6) flow parameters were calculated, and (7) the flow was calculated.

N69-14515
Design Criteria for Zero Leakage Connectors for Launch Vehicles. Computer Programs for Flanged and Threaded Connector Design, Vol. 1 Final Report
R. E. Smith, E. J. Leclerc and M. C. Ray Oct. 5, 1967
G. E. Company, Schenectady, N.Y.

A computer program is described which will produce a connector design when given functional requirements such as operating pressures and temperatures, and material properties such as Young's modulus and allowable stress. The analyses and procedures in the program are based upon calculating static stresses in an assembled connector due to internal pressure, axial pipe load, etc.

N69-21002
Supersonic Transport Lubrication System Investigation
W. L. Rhoads 27 Dec. 1968
SKF Industries

Ball bearings and face seals for use on Mach 3 aircraft gas turbine engine mainshafts were evaluated in tests of up to 50 hours duration using a recirculating system with jet lubrication under typical engine load. Three lubricants were evaluated in these tests.

N69-21352
Development of Compressor End Seals, Stator Interstage Seals and Interior Pivot Seals in Advanced Air Breathing Propulsion Systems
R. M. Hawkins, A. H. McKibbin 20 Jan. 1969
Pratt and Whitney Aircraft

The compressor seal test rig was assembled and tested; modifications were made to correct rig discrepancies.

N69-22222
High Temperature Hydraulic System Actuator Seals for Use in Advanced Supersonic Aircraft
J. Lee 15 Sept. 1967
Fairchild Hiller Corp.

Ten potential seal materials selected from the general catagories of plastics, soft metals, and hard metals were investigated. Fluid compatibility and sliding wear tests were conducted on these materials to determine the most likely candidates for test use. Five designs were developed and evaluated in dynamic screening tests.

N69-23232
Shaft Face Seal with Self-Acting Lift Augmentation for Advanced Gas Turbine Engines
R. L. Johnson and L. P. Ludwig, Apr. 1969
NASA Lewis Research Center

The need in advanced engines for shaft seals with improved pressure, temperature, and speed capabilities is pointed out. In particular, the high speed requirement suggests that the seals must not have rubbing contact. It is shown that incorporation of self acting lift pads to the conventional face seal prevents rubbing contact even though the face deformation occurs.

SUBJECT INDEX

Ceramics	AD457393				
Fibers	N64-18946				
Metallic	AD43964	A6418465	N64-18946		
Organic, Synthetic Material					
General	AD298335	AD298336	A 6422266	N64-26833	N64-12941
Elastomers	A6424812	A6422286	AD230658	A6422284	N64-16870
	N64-19022	N67-22711			
Fluoro-Elastomers	AD410389	AD230658	AD284239		
Plastics	A6422281	AD230658	A6422266	A64-18465	
Rubber, natural		AD110598			
Rubber, Synthetic		AD283459			
IV. Applications					
Aircraft Jet Engines	AD622958	AD438100	AD438119	A6422281	N66-22347
	N67-19126	N67-22347	N67-25487	N67-31151	N6812654
	N68-29320	N68-37608	N6833027		
Aircraft Sealing					
General	AD299586	AD379053L	AD448155	A6411509	N64-26823
Hydraulic	N64-19022	N66-15545	N6922222		
Bearing	N6633673				
Chemical Industry	AD443294				
Compressor	N66-20581	N66-22347	N66-20581	N68-29320	N68-37608
	N6921352				
Cryogenic Fluid	AD446765	AD448155	A64-22266	A64-22284	A64-22286
	N64-12941	N64-25124	N66-15358	N6818124	
Design					
General	AD461695	AD464958	AD464959	AD622958	AD433277
	AD22290	N66-15545	N67-25487	N68-14514	N68-12654
	N68-36254	N6922222			
Back-up	AD230658				
Connector	N69-14512	N69-14513	N69-14515		
Diaphragm	N67-31151				
Face	N67-31128	N68-17610	N69-23232		
Ring	AD110598	AD230639	AD230658		

Viscoseal	N68-11025	N68-26843			
Equipment (Electric)	AD268592	N69-12582			
Hose & Pipe Connectors & Quick Discon.	N64-27305	N67-35568			
Hydraulic Equipment					
General	N66-15545	N6812654	N68-33263	N6922222	
Elast. & Met.		N64-19022			
"O" Ring	AD406307				
Launch Vehicle	AD446765	AD448155	AD46458	AD46459	N64-16870
	N64-27305	N6914512	N6914514		
Liquid Metal	AD283365	N64-32651	N66-19598	N66-39975	N67-25842
	N6817610				
Munitions	AD298335	AD298336			
Nuclear Plant	N64-17297				
Pneumatic Equipment	N66-20581	N66-22347	N67-25487		
Pressure Range 0-5000	AD288558	AD622958	N6629168	N67-17506	
Pressure Range >10000	AD410389				
Pump Sealing	N65-17270	N67-17506	N67-23037		
Space					
Systems Applications	AD449708	AD464958	AD464959	AD601338	AD603655
	AD609174	A6411509	A6422290	N64-26823	N64-19779
	N64-21990	N66-17442	N66-18434	N67-17506	N69-12582
Vehicles	A6422283	N64-25161	N64-31232	N69-14514	
Temperature Range 0-2000°	AD622958	N64-18946	N6813547	N6833027	
0-1000°	AD288558	N6725842	N6817610		
-100-300°	N64-18946				
-300°-0°F	AD460774	AD465492	N64-12941	N6914512	N6914513
Turbine	AD622958	N67-31128	N68-13547	N6814172	N68-23409
	N6910055	N6921002	N6923232		
Vacuum	AD283365	A6411509	A6414336	N67-17506	N6824502
Valve	N66-18434				

Bibliography of Open Literature on Seals

ACKNOWLEDGEMENT

Even more than in the case of the previous volume in this report, this document is the result of the efforts and cooperation of many people.

Most of the abstracts were prepared by the General Electric Company Main Library in Schenectady, New York, under the direction of Miss Esther S. Kalis. Direct supervision of the preparation of the abstracts was performed by William J. Leahy. He was assisted by Farrell D. Madden, Fay W. Secor, and Chester P. Kulesza.

As in the previous volume of ASTIA literature, Mr. John P. Laniewski categorized most of the references for their technical content, and his judgement served as the basis for construction of the Subject Index.

The important tasks of keypunching, machine searching, and listing of the abstracts for the Subject Index was accomplished by Mrs. Jessie Dockstader and Miss Doris Filkins.

Assistance in final preparation of this volume was given by the Schenectady General Electric Typing Bureau, Mr. Ray Ardizzone, Miss Kathleen Hay, Miss Carole Madden, Mr. Howard Robison, Mrs. Mildred Melideo, Mrs. Ismeine DuPont, Mr. Robert S. White, and Miss Lillian Giavonnangelo.

To all of the above, and the others who helped in still other ways, we would like to express our appreciation for the courtesy and cooperation we consistently received in the accomplishment of this project.

R. L. George
R. C. Elwell

INTRODUCTION

This is the second part of a bibliography on seals and sealing, prepared by the Advanced Technology Laboratories (formerly the General Engineering Laboratory) of the General Electric Company, under NASA Contract NAS 7-102.

The first part of the bibliography, which constitutes Volume 3A of this Final Report, was composed of 682 abstracts of literature available from the Armed Services Technical Information Agency (ASTIA). This part, Volume 3B, presents 1482 additional abstracts available from the open literature.

The literature search which culminated in the issuance of this volume was carried out by the General Electric Company Main Library in Schenectady, New York. Shortly after commencement of this search, two major bibliographies on the subject became available. These documents are as follows:

1. "Bibliography on Dynamic Shaft Seals, Preliminary Issue," W. K. Stair, University of Tennessee, Engineering Experiment Station, May 1962, 97 pages, 353 references, Prepared under Subcontract Number 2118 for Union Carbide Nuclear Company

2. "Bibliography on Fluid Sealing," BIB-1, A. L. King, British Hydromechanics Research Association, South Road, Temple Fields, Harlow, Essex, England, May, 1962, 38 pages, 373 references, Price: 20 shillings

Because of the excellent preparation of the abstracts in these two publications, we have repeated many of them in this volume with the approval of the issuing agencies of these documents. Abstracts repeated from the University of Tennessee Bibliography are identified by a letter "T" in the upper right corner, and items repeated from the BHRA bibliography are likewise identified by a letter "B". Examples are references A-142 and A-144 on page 11.

The majority of the abstracts contained herein, however, were prepared by the General Electric Main Library in the period of April-December, 1962. The sources searched were as follows:

Aeronautical Engineering Index	1947 - 1957
Aerospace Engineering Index	1958
Aerospace Engineering-Review & International Aeronautical Abstracts	1959 - 1960
Applied Science & Technology Index	1958 - Nov. 1962
ASM Review of Metal Literature	1944 - Feb. 1962
Battelle Technical Review Abstracts	1952 - Nov. 1962
British Hydromechanics Research Association Bibliography on Fluid Sealing, May 1962, 444 references, 38 pp.	
Chemical Abstracts	1907 - 1961
Engineering Index	1884 - Nov. 1962
Index of NACA Technical Publications	1915 - Sept. 1958
Index of NASA Technical Publications	Oct. 1958 - Dec. 1961
Industrial Arts Index	1914 - 1957
International Aerospace Abstracts	1961 - Aug. 1962
NASA Technical Publication Announcements	Jan. - Oct. 11, 1962
National Conference of Industrial Hydraulics	1947 - 1961
Nuclear Science Abstracts	July 1948 - July 31, 1962
Science Abstracts (A) (Physics)	1898 - Oct. 1962
Science Abstracts (B) (Electrical Engineering)	1898 - Oct. 1962
Technical Translations	1959 - Sept. 15, 1962
Translation Monthly	1955 - 1958
U. S. Government Research Reports	1946 - Sept. 20, 1962
University of Tennessee, Engineering Experimental Station, Bibliography on Dynamic Shaft Seals, W. K. Stair, May 1962, 353 references, 97 pp. Includes references through Dec. 1961.	
Vacuum Abstracts	1951 - July/Aug, 1962

Although it may not be apparent from the dates shown, each source was searched back to its beginning, in the interest of completeness.

The complete process in the production of this publication was as follows:

a. the sources above were searched for pertinent references
b. abstracts were prepared of articles which had potential value
c. each abstract was studied and its content coded in terms of a ten-digit code. For instance, the number 01-13-47-63-83 would describe a reference about installation of bronze labyrinth seals in steam or gas turbines. There were 2,414,720 possible code numbers of this type in the system
d. a serial number was assigned to each item, consisting of a letter followed by a number. The letter was the first letter in the primary author's name, and the number was the serial number within that alphabetical group. These identification numbers are used in the test and Subject Index.
e. an IBM card was punched for each code number for each reference, up to a maximum of 20. (Most abstracts had multiple codings because of multiple types of technical content). A total of approximately 9000 cards were required for the open literature abstracts.
f. While the text was being type-written (using IBM "Charter" type to save space), the IBM deck was searched for preparation of the Subject Index. For instance, to list all references on Bushings (see page 147), the deck was sent through a keysorting machine set for the number 1 in Column 8 of the card and the number 4 in Column 9. This task obviously could also be done manually, but because of the extensiveness of the Subject Index, a great deal more labor would have been required.
g. The final step was to duplicate the document.

We have intended in this search to abstract every English language document that it was feasible and worthwhile to obtain. This was mainly so that the task would never have to be done again. With a scope of this magnitude, errors and omissions are unavoidable. The authors would greatly appreciate being informed of these so that a potential second edition of this document would be of even greater value in the saving of future technical manpower.

A-001
Abbott, H.M.
Space Environmental Effects on Seals, Gaskets, Adhesives and Other Elastomeric and Polymeric Materials
Lockheed Aircraft Corp., Missiles and Space Div., Sunnyvale, Calif., NP-11194, 220 pp., Contract AF04(647)-673., Sept., 1961

This bibliography contains 558 selected references on seals, gaskets, adhesives, sealants, and other elastomeric and polymeric materials under space conditions. These references were compiled for the years 1951-1961. A material index is included as a possible aid in locating specific materials.

A-002
Adam, H.
Theoretical Principles of Compressive Glass Seals and Their Practical Results
Feinwerktechnik 56, No. 2, pp. 29-40, 1952
Spec. Lib. Assn. Trans. 57-2748 (53 pp.)
(Loan copy only)

A-003
Adam, H.
Compressed-glass Seals. Principles, Manufacture and Technical Applications
Glas-und Hochvaknum.-Tech. No. 7, pp. 123-34, 1952. Spec. Lib. Assn. Trans. 57-2757 (27 pp.)
(Loan copy only)

A-004
Adam, H.A., Kaufman, S.
Indium Seals for Dismountable Vacuum Systems
Journal Sci. Instrum., V. 341, pp. 123-124, March, 1957

The present investigation was required to produce a metal seal for applications where no external clamping was possible and compression of the seal was due solely to atmospheric pressure. Details of preparation, procedure, and use are described. Various sizes are checked. Leak rates against surfaces of various materials given.

A-005
Adilletta, J.G.
How to Apply O-Rings
Product Engineering, V. 28, pp. 176-182, Aug., 1957

Test program develops simplified principles for proportioning mechanical parts to achieve maximum sealing at minimum cost. Type of sealing required dictates seal choice.
Effect of factors such as fluid pressure, surface finish, and eccentricity of shaft and cylinder on initial "squeeze," cross-section rind diameter, and Durometer of rubber.

A-006
Adloff, K.
Creep of Flanged Joints
Warme, V. 58, pp. 352-355, June 1, and pp. 387-389, June 15, 1935

The calculations by different methods of the stresses in the tube and flange material and in the connecting bolts, both for solid and loose ring flanges is compared. In order to insure a tight joint under pressure, the initial load on the bolts should be at least equal to twice the load due to the steam pressure. It is an advantage to heat the bolts when the flange joint is being erected. The effect of the creep is to reduce the stress in the bolts. The determination of a time factor for estimating the life of a flanged joint under steady creep is considered.

A-007
Admire, B.W. and Mayor, F.S.
A Gas Shaft Seal For The HNPF Sodium Pump
Atomics International, Div. of North American Aviation, Inc., Canoga Park, Calif., NAA-SR-MEMO-2616, June 30, 1958, (6 pp.)

Lip type, oil labyrinth gas shaft seals were tested for use on Hallam power reactor sodium pumps with a 5' inch diameter rotating shaft. The seals were not recommended for use owing to excess helium leakage and short life.

A-008
Aitchison, T.C., and Bowers, B.N.
Nitrile Rubber Gaskets Make Better Seals
G-E Review, V. 55, pp. 46-50, May, 1952

Characteristics of nitrile rubber makes it an almost ideal gasket material for sealing the joints of liquid-filled electrical apparatus against oil and non-inflammable insulating liquids such as askarel, air, and moisture, for reasons listed. Radical changes were made in previous gasket practice to fully utilize the potential of the material.

A-009
Alexander, T.E.
How To Maintain Mechanical Seals
Oil and Gas Journal, V. 60, p. 123, 4+. Feb. 12, 1962

The six causes of trouble in mechanical seals are:
1) Alignment, 2) Faulty installation, 3) Wrong application, 4) Corrosion, 5) Temperature, 6) Solids.
Design s pressures up to 1,500 psi, temperature to 500°F, and rotating speeds to 40,000 rpm. Alignment, 0.002 in. max. for seal faces.
Drawings and tables.

A-010
Allen, C.M., Bell, J.C., and others
Rotating-shaft Helium Seal Investigation
Battelle Memorial Inst., Columbus, Ohio, NP-11243 Contract DA-44-009-eng-3375 (71 pp.)
Aug. 25, 1959

A research program was conducted to study positive rotating-shaft seals using helium for high speed turbomachinery.

A-011
Allen, D.S. and Whitley, S.
Improvements In Or Relating To Devices For The Sealing Of A Rotatable Shaft In Passage Through A Casing
British Patent 881,877, Nov. 8, 1961

A device for sealing a rotatable shaft in passage through a casing is designed which has a diametral clearance between the shaft and the sleeve so small that a small feed of sealing gas is sufficient and leakage rates are low.

A-012
Allen, R.E.
Hydraulic O-Ring Packing For Sealing Cylinders And Pistons
Prod. Engrg., V.18, pp. 162-166, July, 1947

Groove designs for non-moving and moving types of O-ring seals in hydraulic units and other applications. Arrangements shown for sealing cylinders, pistons, end caps, valve covers and adjustable threaded parts. Sealing action of O-rings and properties of compounds available for making O-rings are described.

A-013
Alpert, D.
Recent Advances In Ultra-high Vacuum Technology
Vacuum, V. 9, No. 2, pp. 89-96, May, 1959

The two major directions for advances in ultra-high vacuum technology are: 1) the attainment and measurement of pressures below 10^{-10} mm Hg, and 2) the development of large ultra-high vacuum systems. Included are comments and drawings of seals. Knife edge flange arrangements utilizing a flat copper gasket; the step type with copper [illegible] O-ring type; and gold gasket seal.

A-014
Akimus, M.E. Jr.
Trouble Shooting In Hydraulic Systems
Coal Age, V. 61, pp. 78-81, March, 1956

Methods of detecting leakage. Some features of hydraulic components often overlooked as sources of trouble shooting. Diagrams, photograph, labels.

A-015
Andrews, Julian N.
Hollow Metal O-Rings. Seals For Extreme Environments
Product Engineering, V. 32, pp. 47-57

Design guide for low and high temperature O-ring joints includes pressure limits, material selection, surface finishes, groove dimensions, and bolting forces. Temperatures from cryogenic temp. to over 2000°F. Plain hollow, pressurized hermetic, and self energized, 100 to 400 psi. Plating and coatings improve sealing.

Effects of venting O-rings, photographs, temp/stress curve, material use table, O-ring sealing force curve, O-ring vs. solid metal gasket table, pre-installation dimension table, drawings, design procedure.

A-016
Anon.
Packing As A Product Of The Rubber Factory
Ind. Rub. Wld, 3300 W, Sept. 10, 1897

Gives history of the use of packing with illustrated description of types.

A-017
Anon.
Packing Ring For Piston Valves
Mech. Eng., V. 4, p. 782, Nov. 12, 1899

A description of a packing ring for piston valves for a heavy freight locomotive built for the Illinois Central Railroad by the Brooks Locomotive Works. The packing ring is made in one solid piece, and turned to a larger diameter than the bore of the bushing. A saw cut is made on the bevel across the ring and a shim of the required thickness is inserted. The ends are then clamped together by a bolt, with lugs for this purpose being provided on the inside of the ring and accurately turned to the proper size. The design seems to have the advantage of a plain plug valve, and an additional one in that the valve may at any time be made tight with comparatively little expense.

A-018
Anon.
Labyrinth Packings
Engineering, V. 86, pp. 85-86, Jan. 10, 1908

A general description of labyrinth packings, pointing out that the essential idea on which the packing is based is not so much the causing of the steam to take a tortuous path as to wire draw it at a great number of points. In the latter part of the paper a formula for the discharge of steam through packings of this kind is established.

A-019
Anon.
Buhne Fibrous Babbitt Packing
Power, V. 42, p. 616, Nov. 2, 1915

Buhne fibrous babbitt packing is composed of long, fine fibers of composition metal for steam, water, oil, and ammonia. For each purpose a different alloy is used. Drawing and instructions for packing new and worn rods.

A-020
Anon.
Molding Metallic Packing For Saturated Steam Locomotives
Railway Mechanical Eng., V. 90, pp. 264-5, May, 1916

A-021
Anon.
Piston Rings
Mech. Wld., June 22, 1917

Types of rings used and points in removing and refitting.

A-022
Anon.
Sarco Metallic Gaskets
Power, V. 47, p. 472, April 2, 1918

A combination lead and copper gasket developed by the Sarco Company. For flange work, the lead ring forms the inner member, and just fitting over the outer edge is a copper ring of smaller cross section. This permits the lead ring to come under considerable pressure in tightening the joint before the flange begins to compress the copper ring. The copper prevents the blowout of the lead ring. Details, application and design drawings presented.

A-023
Anon.
Development Of John Crane Flexible Metallic Packing
Int. Marine Eng., V. 23, p. 318, May, 1918,
Ind. Management, V. 55, p. 422, May, 1918,
Power, V. 47, p. [illegible], May 14, 1918

A-024
Anon.
Mixtures For Metallic Packing
Foundry, V. 46, p. 271, June, 1918

A-025
Anon.
Cast Iron Piston Rod Packing
Railway Mechanical Engineering, V. 93, pp. 577-8, Oct., 1919

The London and North Western, and the North Staffordshire, having experienced difficulty with their white metal piston rod packing on superheater locomotives, have successfully developed a cast iron packing which has given particularly good service for some little time. L & N W RR uses a mixture of 40% selected scrap, 30% Kettering silicon material, and 30% old ingot molds (Hematite). The N.S. RR makes its packing from an ordinary commercial quality of cast iron with approximately 33% Barron Hematite added.
Detailed drawings of both packings are included.

A-026
Anon.
Steam, Hydraulic And Miscellaneous Packings
Belting & Transmission, V. 16, pp. 19-21, June 26, 1920

Notes on materials used in construction, how they are applied to engines, pumps, valves, etc., and conditions affecting their operation.

A-027
Anon.
Ring Packed Stuffing Box For Valve Stems
Railway Review, V. 67, pp. 482-3, Sept. 25, 1920

A-028
Anon.
Tiptop Plastic Metallic Packing
Power, V. 52, p. 864, Nov. 30, 1920

Tiptop packing fits in the stuffing box lengthwise. Being wedge-shaped, it fits the stuffing box and rod, making practically a true circle when the packing is forced against the rod or valve stem as the gland is forced into place.
It is made of a soft shredded anti-friction alloy which is permeated with graphite and mica and then compressed into bars, segments, or into rings which may be cut diagonally with an ordinary jack knife to fit the rod or box.

A-029
Anon.
Throttle Stuffing Box For Plastic Packing
Railway R., V. 70, p. 166, Feb. 4, 1922

A-030
Anon.
Failure Of Labyrinth Packing Shuts Down Turbine
Power, V. 55, p. 211, Feb. 7, 1922

This article describes an accident which occurred at a power company due to a packing failure and the measures taken to correct it.

A-031
Anon.
Use Of Non Metallic Packing In Refrigerating Field
Refrigerator W, V. 57, pp. 29-30, Nov., 1922

A-032
Anon.
All Metal Packing For Locomotive Air Pumps
Railway Review, V. 75, pp. 141-42, July 26, 1924
Railway Age, V. 76, p. 1410, June 11, 1924

The packing rings, two in number, are constructed with mechanical accuracy from four segments which break joints in the center of the rod bearing surface. The combination of two rings working in unison will prevent leakage of either steam or air. The wide bearing surface each ring has on the rod and the grooves of the case insures long life.
Drawings and description included.

A-033
Anon.
Alden Turbine-condenser Expansion Gasket
Power, V. 60, pp. 705-6, Oct. 28, 1924

To withstand temperatures that would prevail under atmospheric conditions, the expansion gasket is made of high-temperature gum rubber in the form of a circular tube provided with two small openings for the introduction and circulation of water under pressure as low as 10 lb./sq. in. Drawings and data curve indicating inflation pressures at which leakage starts.

A-034
Anon.
Steam Engine Packing
Power Engr., V. 20, pp. 175-7, May, 1925

Review of requirements of good packing and of methods adopted to fulfill these practices.

A-035
Anon.
Cooke Seal Ring Can Be Adapted To Any Rotating Shaft
Power Plant Engineering, V. 30, pp. 872-3, Aug. 1, 1926,
Power, V. 64, p. 382, Sept. 7, 1926

The seal is rotated with the shaft. This particular style uses double seal rings, one at either end of the stuffing box, with a spring between to exert the pressure necessary to effect the seal.
Detailed description and assembly drawing.

A-036
Anon.
Packing Glands For Large Steam Turbine Shafts
Power Plant Engng., V. 30, pp. 947-950, Sept. 1, 1926

This article discusses in brief the details of some of the packing glands used on large steam turbines. A few are: carbon ring, water seal and labyrinth gland. Illustrations.

A-037
Anon.
Artan High Pressure Cast Iron Metallic Packing
Railway Age, V. 82, p. 669, Mar. 5, 1927
Mechanical Eng., V. 49, p. 170, Feb., 1927

The packing consists of a series of ground cups which fit against each other forming a steam tight seal when pressed into the bottom of the stuffing box by the gland. Each packing ring set consists of two rings of rectangular section placed side by side with two cast iron spring rings surrounding the outer circumference of the two sealing rings.

A-038
Anon.
Britimp Metallic Packing
Engineering, V. 123, pp. 402-3, April 1, 1927

A type of packing ring which is self aligning, and when once correctly adjusted, it is impossible to increase the pressure on the rod or shaft. (7 types pictured). Applications are discussed.
A running life of 125,000 miles without adjustment has been recorded, compared with 22,000 with white metal packing.

A-039
Anon.
Packing For Hydraulic Machines
Power, V. 65, p. 969, June 21, 1927

A discussion of various packing materials. The limitations and advantages of braided flax packing, asbestos packing, hemp packing, granulated wood and metal packing, plastic-metallic packing, and braided cotton packing.

A-040
Anon.
Armor-clad Flexitallic Gasket
Power Plant Engineering, V. 33, p. 598, May 15, 1929
Blast Furnace & Steel Pl., V. 17, p. 727, May, 1929

A gasket designed to take care of the most severe modern service conditions - high temperature, high pressure, and acid conditions. The elasticity of the gasket was not exceeded at 100,000 pounds load (9000 lb/sq. in.).

A-041
Anon.
Spring Packing For Steam Turbine Stuffing Boxes
Marine Eng., V. 35, p. 83, Feb., 1930

V-shaped grooves on the outer circumference of rings of corresponding shape are held in position by a tension wire. Between the ends of the wire a corrugated spring is provided, which is under compression. Test data, tabulated and discussed are presented.

A-042
Anon.
Selection And Application Of Pump Packing
Nat. Pet. N., V. 22, p. 61, May 7, 1930

A-043
Anon.
High Pressure Gasket Pointers
Pulp & Paper Magazine of Canada, V. 30, p. 428, Oct. 9, 1930

A-044
Anon.
Carbon Packing Rings For Steam Turbines
Power Plant Engineering, V. 34, p. 1219, Nov. 1, 1930

A discussion of carbon packing rings, stressing the care of fit, spring pressure and examination. Some troubles similar to those caused by electrolytic action, and the correction of same.

A-045
Anon.
Typical Designs Of Stuffing Boxes
Product Eng., V. 3, pp. 25-6, Jan., 1932

Formulas for proportioning cast-iron stuffing boxes with brass glands; equations for calculating principal dimension when other materials are used or higher pressures are involved.

A-046
Anon.
Alco Plastic Metallic Packing
Railway Mech. Eng., V. 106, p. 42, Jan., 1932

The particular advantages of this new packing include ease of application and renewal, durability, reduced friction and the maintenance of steam and fluid-tight joints under most adverse conditions.
Alco is a plastic self-forming packing, composed of porous all-metallic kernels, saturated with non-drying lubricants.
Temperatures up to 600°F.

A-047
Anon.
Protecting Bearings With Unit Oil Seals
Machine Design, V. 4, pp. 30-2, March, 1932

Oil retaining rings and packings for plain and roller bearings; shafts from 1/2" to 2" dia.

A-048
Anon.
Self-contained Oil Seal For Bearings
Engineering, V. 133, p. 502, April 22, 1932

Single seal introduced by Charles Weston; known as "Gits" precision oil seal, and is standardized for shafts ranging in diameter from 5/16 in. to 6 in.; consists essentially of special form of self-contained unit.

A-049
Anon.
Garlock Packing Co. Expansion Gasket
Railway Age, V. 92, p. 1035, June 18, 1932

Photograph and paragraph describing a recently developed gasket known as the Guardian. Designed to meet any requirements of pressure or temperature, to resist the corrosive effects of gases and liquids, and to maintain tight joints under changing temperatures with continued resiliency.

A-050
Anon.
Granola Metallic Packing
Canadian Engineer, V. 63, p. 46, July 19, 1932

A-051
Anon.
Protecting Bearings With Unit Oil Seals
Machine Design, V. 4, No. 8, pp. 30-2, Aug., 1932

Describes the advantage of unit oil seals. The use of cork, leather, and asbestos as packing members is discussed, as is shaft whip and piston ring arrangement.

A-052
Anon.
Plastic Metallic Packing
Engineer, V. 154, p. 338, Sept. 30, 1932

New type of packing marketed under trade name of "Sealite" and made by R.C. Taylor and Co.; consists of 87% of finely divided non-fibrous lead-base alloy, to which has been added 10% of graphite and small amounts of long-strand asbestos.

A-053
Anon.
New Method Of Packing Expansion Joints; Another Piping Problem Solved (Yarnall-Waring Co.)
Power, V. 76, pp. 236-7, Nov., 1932

Expansion joint, force packed by "guns" on shell, forces semi-plastic packing through a drilled channel into the stuffing box. Packing ingredients are long-filler asbestos, graphite, and inorganic filler impregnated with high test oil.
Discussion of operational run. Drawings.

A-054
Anon
Stuffing Boxes Packed Under Pressure
Power Plant Eng., V. 36, pp. 814-5, Dec., 1932

Lubricated packings forced around slip joint cylinders with pressure maintained on joint.

A-055
Anon.
Spring Fitted Oil Sealing Ring
Engineering, V. 135, p. 30, Jan. 6, 1933

The device is known as the "Perfect" oil and grease retainer.
The leather ring with the tightening spring is enclosed in a light casing with a cover held in place by a beaded edge.
Manufacturing tolerances, operation, and application discussed. Drawing included.

A-056
Anon.
Haughtons Metallic Packing
Soc. Chem. Ind. Journal (Chem. & Ind.)
V. 53, pp. 187-8, Feb. 23, 1934,
Chem. Age (London), V. 30, p. 230, March 17, 1934

A-057
Anon.
Closures For Ball And Roller Bearings
Mech. World, V. 96, pp. 177-9, Aug. 24, 1934

Details of a number of different types of lubricant retaining closures which form simple and adequate protection under various conditions of service.

A-058
Anon.
Hoyt Metallic Piston Rod Packing
Engineering, V. 138, p. 484, Nov. 2, 1934,
Engineer, V. 158, p. 422, Oct. 26, 1934

A new packing ring for the glands of piston rods, given the name of "Silver Ring Packing." Differs from babbitt, but possesses high anti-frictional properties. The metal is made up in the form of conical washers with double bevelled edges which provide parallel faces to bear against the rod and the interior of a stuffing box. These packings have been given some very severe trials and are said to be effective from vacuum to 5000 lb/sq. in. Picture included.

A-059
Anon.
Flow Through Labyrinth Packing
Mech. Engrg., V. 56, pp. 678-680, Nov., 1934
(Abstract translation from Die Warme, Aug. 11, 1934, V. 57, pp. 513-517)

Application of the formula for the derivation of the labyrinth equation.

A-060
Anon.
Investigations Of Packings For Stuffing Boxes And Hydraulic Cylinders (In German)
Munich and Berlin, R. Oldenbourg (22 pp.), 1935
(Eng. Soc. Lib., N.Y.)

Tests made at laboratory of Berlin Institute of Technology; imperviousness, leakage, and frictional resistance of 9 popular packings was determined.

A-061
Anon.
Anti-friction Metallic Packings
Chemical Age (London), V. 33, p. 286, Sept. 28, 1935

A-062
Anon.
Seals Against Fluid Pressure, Pt. I; An Approach To The Problem Of Preventing Leaks In A Power Plant
Power Plant Engineering, V. 40, No. 5, pp. 280-2, May, 1936

Packing specially developed; procedure in preventing leaks.

A-063
Anon.
Glands And Seals In Turbines
Power, V. 80, p. 307, June, 1936

A brief general description of labyrinth, water, and carbon ring seals, with drawings of a few typical examples. No data.

A-064
Anon.
Seals Against Fluid Pressure, Part II; Materials Used As Principal Constituents Of Packing And Gasket
Power Plant Engineering, V. 40, No. 6, pp. 354-6, June, 1936

Material characteristics which suit them as packing and gasket constituents: aluminum, asbestos, asphalt, babbitt metal, brass, bronze, cardboard, cork, canvass, duck, felt, fiberboard, flax, graphite, hemp, iron, jute, lead, leather, linen, mica, Monel metal, nickel, oils, paper, papier-mache, paste and plastics, rubber, soapstone, steel, tallow, tar, wood, wool.

A-065
Anon.
Seals Against Fluid Pressure, Part III; Specifications For Sheet Packings Used In The Making Of Fixed Joints
Power Plant Engineering, V. 40, No. 7, pp. 416-9, July, 1936

Rubber packing and gaskets; molded, sheet, and strip. Rubber, cloth insertion sheet packing; rubber, wire insertion sheet packing; compressed asbestos sheet packing; asbestos, metal sheet packing.

A-066
Anon.
Seals Against Fluid Pressure, Part IV; Specifications For Gasket, Diaphragm Packing, And Fabric Packing For Condenser Tubes
Power Plant Engineering, V. 40, No. 8, pp. 482-4, Aug., 1936

Asbestos-copper corrugated gasket; asbestos metallic cloth gaskets; metallic-encased gasket, diaphragm packing; fabric condenser tube packing.

A-067
Anon.
Seals Against Fluid Pressure, Part V; Materials Used And Specifications For Making Seals Where Sliding Contacts Are Necessary
Power Plant Engineering, V. 40, No. 9, pp. 536-9, Sept., 1936

Braided asbestos rod packing; asbestos valve stem packing; spiral gland low pressure packing; tucks packing; leather hydraulic packing; flexible metallic packing; metallic and non-metallic packing; semi-metallic packing.

A-068
Anon.
Seal-lubricant In, Dirt Out
Electrical Mfg., V. 18, pp. 27-30, Nov., 1936

Designs of various types of ball bearing dust and dirt seals; a chart showing different type seals and how effective they are under various conditions; spring pressure seals for severe service; preventing oil leakage past a plain bearing.

A-069
Anon.
Seals Against Fluid Pressure, Part VI; Types Of Flange Faces Used On Power Plant Piping And Physical Requirements Of Gasket Materials
Power Plant Engineering, V. 40, No. 11, pp. 643-61, Nov., 1936

Flange faces with matched surfaces; purpose of gaskets; what squeese does to gaskets; relation of pressure to compression.

A-070
Anon.
Seals Against Fluid Pressures
Power Plant Engineering, V. 41, pp. 114-116, Feb., 1937

Part VII. Specifications issued by the United States Navy for packings suitable for gaskets, rods, shafts, and condenser tubes. Oil resistant gaskets. Cloth-insertation rubber packing. Unvulcanized rubber-graphite packing. Carbon packing. Shredded lead packing.

A-071
Anon.
Practical Problems In Packing
Power Plant Engineering, V. 41, pp. 128-9, Feb., 1937

A general commentary on good practices for monitoring packings during use and maintenance. No technical data.

A-072
Anon.
Steam Packing For Piston Rod
Engineer, V. 164, p. 357, Oct. 1, 1937

This packing is intended to resist high pressure with a minimum of friction on the rod. It consists of a black packing on the outside, and a cone packing within. Eight blocks are held in strong rings. Two blocks of each set are babbitt lined, the other two act as guides. The joints are staggered to close a through passage. Cone packing bears against a spherical seating. Drawing included.

A-073
Anon.
The Labyrinth Packing
The Engineer, V. 165, No. 4280, pp. 83-84, Jan. 21, 1938

An editorial review of the work of Parsons in developing labyrinth seals for his steam turbines. The article is chiefly historical in value.

A-074
Anon.
Report Of Special Section On Packing Classification
Proceedings Am. Soc. for Testing Materials, V. 39, pp. 476-7, 1939

A proposal for a classification of rubber-containing gaskets, valve disks, sliding contact packings, and diaphragms to be employed in the future proceedings of Subcommittee VI.

A-075
Anon.
Rodpak Mfg. Co. Develops Rod Packing For Severe Conditions
Steel, V. 104, p. 77, April 10, 1939
Refrig. Eng., V. 37, p. 117, February, 1939

This packing employs two metal rings (convex on one face and flat on the other) as the sealing medium on the shaft. For some applications these rings are made of a special babbitt metal, while for others a metal which will resist heat up to 700°F is employed. Between these sets of rings are rings of Neoprene, concave on both faces, to mate with convex surfaces or rings.

A-076
Anon.
Neoprene Being Successfully Used For Compressor And Pump Rod Packings
Mach., V. 45, pp. 582-3, April, 1939

A-077
Anon.
Current Types Of Gaskets And Closures
Heating-Piping and Air Conditioning, V. 11, pp. 500-1+, 565-6+, Aug./Sept., 1939

The usual bolted joint construction is being largely modified or superseded by more reliable, more easily maintained, and more economical forms. The author describes several of these current type gaskets and closures.
Manhole type joints, ring type joints, bellows type joint, formula for force calculation, and drawings included.

A-078
Anon.
Laminated Gaskets
Engineer, V. 168, p. 384, Oct. 13, 1939

The "Dixon" gasket consists entirely of a number of laminations. Materials can be varied. In marine and aircraft engines, only copper laminations may be used. Copper and terne plate was found advantageous. Copper-aluminum and aluminum-terne have also been used.
Drawings included.

A-079
Anon.
Dickson All-metal Laminated Gasket
Engrg., V. 148, p. 540, November 10, 1939

A static gasket composed of thin metallic laminations; used as head seal for internal combustion engines; can be used both hot and cold.

A-080
Anon.
Seals Rotate With Shaft
Neoprene Notebook, p. 118, July-Aug., 1940

A radial type seal is described in which neoprene is successfully used as the sealing material. Features of seal most attractive to design engineers are listed.

A-081
Anon.
New Material For Gasket; Layers Of Synthetic Rubber Buna And Steel Wire Netting
Automotive Industries, V. 83, pp. 265-6, Sept. 15, 1940

A-082
Anon.
Neoprene and Asbestos Used In The Production Of Gaskets
Machinery, V. 47, p. 169, Nov. 1940

Two types are available. One type has considerable compressibility and is adapted for joints where relatively light flange pressure is applied.
Advantages of new gasket are permanency of size and ability to resist the effects of relatively high temperatures.
Second type for heavier flange pressures.
Description of manufacture and advantages.

A-083
Anon.
Gland Design For R. W. Engines
British Motor Ship, V. 22, pp. 281-284, Dec., 1941

Article discusses an original piston rod gland design used on a Richardson Westgarth double-acting oil engine. Manner in which design was improved through detail developments is covered. Of equal importance to design was need for accurate workmanship and ascertainment of best working clearances to give best all-around results.

A-084
Anon.
Construction And Action Of Stuffing Boxes
Mech. World, V. 110, pp. 459-62, Dec. 26, 1941

Analysis of a variety of common and special type stuffing boxes.

A-085
Anon.
Lead Lubricant Combined In A Rubber Gasket; Swing Joint For Low Pressure Steam Line
Scientific American, V. 166, p. 66, February, 1942

The secret of the efficiency of the joint lies in a rubber gasket into which powdered lead is milled before the gasket is shaped. Because of the lubricating action of the lead, it is claimed that the new joint turns more freely under pressure than other types and that the lead lengthens the life of the gasket. The swing joint handles steam pressures up to 100 psi.

A-086
Anon.
Rayon Rope Seals In The Power Of Vital Hydraulic Machines
Rayon, V. 23, p. 672, Nov., 1942

A-087
Anon.
Formula For Packing Ring Length
Oil Weekly, V. 108, p. 30, Feb. 15, 1943

Formula requires knowledge of rod and plunger diameter, width of gland or box, and proper packing thickness. All of this information is available from factory specifications on pump, or may be experimentally determined at first shutdown.

A-088
Anon.
Gasket Material
Modern Plastics, V. 20, pp. 78-9 and 156, May, 1943

High tensile, fibrous sheet material impregnated with phenolic plastic developed by Detroit gasket manufacturer from which he cuts various types of gaskets for most exacting aircraft engine requirements; new material is reported to have passed severest tests under all conditions and has been accepted as alternate material by aviation industry; manufacturing process illustrated and described.

A-089
Anon.
You Can Pack Centrifugal Pumps To Hold High Pressures And Temperatures
Power, V. 88, pp. 8-10, 62a, Oct., 1943

Discussion of problems involved; classification of common types of packings and applications; general arrangements of sealing systems.

A-090
Anon
New Type Of Resilient Seal Simplifies High-pressure Valve Design
Product Eng., V. 15, pp. 158-9, March, 1944

Description of T-section seal developed by Maytag engineers for application to valves used in machine tool applications and in airplane hydraulic systems.

A-091
Anon.
O-Ring Packings Simplify Lightweight Hydraulic Equipment
Prod. Engrg., V. 15, pp. 581-585, Sept., 1944

Details of dimensional tolerances, materials, lubrication, finish of surfaces, frictional characteristics, and causes of failure are given. Reciprocating, static, and rotary seal applications are considered.

A-092
Anon.
Bellows-type Shaft Seal
Oil & Gas Journal, V. 43, p. 133, Oct. 21, 1944

A-093
Anon.
Silicone-rubber Gaskets Used In Navy Searchlights
Elect. Eng., V. 63, p. 455, Dec., 1944

Silicone rubber which retains its elastic properties at temperatures as low as 60 degrees below zero or as high as 575 degrees Fahrenheit, is now being used for gaskets for Navy searchlights. Its advantages, performance and other uses discussed.

A-094
Anon.
Packing, Engine And Pump
Mechanical Packing Manual, Graton and Knight Company, Worcester, Mass., 1945

A-095
Anon.
Packing - "U" Cup Hydraulic
P.B. 36832 - Off. Tec. Ser., Apr., 1945
(Army-Navy Aeronautical Standard Drawing AN6226)

This third revision of the ANA Standard for "U" cup hydraulic packing for pressures up to 600 psi gives the drawings showing the size and location of markings.

A-096
Anon.
Rotary Seal Is Extremely Compact
Machine Design, V. 17, p. 157, June, 1945

A rotary sealing unit of extremely compact design suited especially to use in fluid clutches, hydraulic transmissions or torque converters, and similar equipment, is covered by patent 2, 362, 341 recently assigned to General Motors Corporation. As illustrated in the accompanying exploded drawing, the design provides a concentric nonmetallic sealing nose with a separately pocketed spring loading arrangement. Short descriptive article. No data.

A-097
Anon.
New Packing Resists Corrosion And Heat
Iron Age, V. 155, p. 68, June 7, 1945

Flexible metallic packing capable of withstanding temperatures up to 2000°F and is highly resistant to corrosive gases, alkalies and most acids.

A-098
Anon.
Graphite Bearings And Seals
Heating & Ventilating, V. 42, p. 88, Sept., 1945

Carbon-graphite, which has self lubricating properties and a low coefficient of friction, is manufactured as Graphitar in various shapes and sizes to serve as air compressor and steam engine piston rings, cylinder liners, grease seals, turbo blower seals, etc. Resists chemical action and wear, is mechanically strong and light. Tensile strength 750 to 2500 lbs. per sq. in. Compressive strength 1850 to 37, 000 lbs. per sq. in. Breaking strength 3750 to 13, 000 lb. per sq. in.
Rebound (concrete floor) 70% of the height dropped.
Oxidizes slowly and loses weight at 700°F.

A-0[illegible]
[illegible]on.
[illegible]formation On Manufacture Of Oil Seal Rings
[illegible]immer rings)
Naval Tech Mission in Europe Tech. Rept. 375-45), Sept., 1945, Off. Tech. Serv. PB-22819

This report is a brief description of the manufacture of "Simmer Rings, " and a record of disposition made of drawings and samples removed from the plant of Walter, Hans, Otto and Richard Freudenk, Weinheim, Germany. These seals are used for preventing leakage of oil from bearings, for hydraulic packing and for other sealing purposes on rotating and reciprocating shafting of all types and sizes which operate within the temperature limitations of the rubber or leather used in the manufacture of the rings.

A-100
Anon.
Metallic Gasket Materials
Paper Ind., V. 27, p. 926, Sept., 1945

A-101
Anon.
Ajax Spiral Wound Gaskets
Marine Eng., V. 50, p. 220, Nov., 1945

Contrasted to the ordinary static seal as obtained with all metal gaskets or sheet packing. The Ajax spiral wound metal and asbestos gasket can be engineered for any specified service to give a predetermined compressibility and resiliency. The knifelike edges of the metal in Ajax gaskets seat themselves in a continuous spiral groove in the joint faces at a relatively low bolt stress. Description, picture, curves, and detail of sealing action.

A-102
Anon.
Sealing Motor Bearings
Elect. Rev. Lond., V. 137, p. 756, Nov. 23, 1945

A simple design of seal that prevents leakage of mineral oil and the ingress of foreign matter into the ball or roller bearings of an electric motor is described. The shaft is encircled by a stationary packing member of L-shaped section which is kept in light running contact with the shaft by a spring. The material used for the seal depends on the temperature and peripheral speed. When conditions require it, a dual type can be fitted.

A-103
Anon.
Pump Shaft Seal
Product Engineering, V. 16, p. 832, Dec., 1945

Heat resistant and noncorrosive are the characteristics of new seal; available in both bronze and stainless steel.

A-104 T
Anon.
Steam Turbines: Seals And Glands
Power, V. 89, pp. 800-1, Dec., 1945

A basic description of seals and glands used in steam turbines is given. Included are carbon rings, labyrinth seals, and water glands.

A-105
Anon.
Packings And Gaskets: Hydraulic
PB21549, Off. Tech. Ser. (22 pp.)(AN-1+H P-114b)
PB34918, Off. Tech. Ser. (36 pp.)(with amendment 1, Mar. '46)

A-106
Anon.
A Seal For A Rotating Shaft
Engr., V. 182, p. 218, Sept. 6, 1946

Brief description of an English shaft seal employing a "Hydroflex" metal bellows.
Design problems discussed.
Applicable to air-compressors, pumps, gear boxes, paper machinery, and chemical plant.

A-107
Anon.
Typical Methods Of Sealing Rotating Shafts
Prod. Engrg., V. 17, pp. 108-109, Oct., 1946

Shaft seals, used to prevent seepage of lubricating oils or other liquids, range from simple flange packings to elaborate bellows and lapped surface devices. Some of these, utilizing wave, coil, and garter springs, diaphragms, and locking rings, are described and illustrated.

A-108
Anon.
Life Stories Important In The Oil Fields
Inco., V-21, pp. 18-19, Winter, 1946-1947

Parts of new type pump for handling sour crude oil are made of monel, except for the graphite-impregnated plastic packing rings.

A-109
Anon.
Sealing Rotating Shaft
Soc. Chem. Industry (Chem. & Industry), p. 21, Jan. 11, 1947

Details of bellows shaft seal particularly suitable for shaft seal assembly in refrigeration compressors, pumps, gear boxes, paper machinery, and chemical plants.

A-110
Anon.
Press-fit Tolerances Maintained in Following Production Line Technique To Turn Out Millions Of Oil Seals
Steel, V. 121, pp. 86-87, 116, 118, 121, Oct. 13, 1947

Production line methods used in the manufacture of seals.

A-111
Anon.
Sealing Of Hydraulic Compartment Access Door, C54 Series, Douglas
U.S. Air Force
P.B. 91771 - Off. Tech. Ser., Mar., 1948

1. C54 (Airplane)
2. Airplanes, hydraulic compartment sealings
3. Fire prevention
4. AAF to 01-40NM-189

A-112
Anon.
Typical Ball Bearing Seals And Internal And External Slinger Rings
Product Engineering, V. 19, pp. 110-1, Oct., 1948, pp. 122-3, Nov., 1948

Design illustrations showing lubricant retention and dirt resistance functions of spring; flange and labyrinth seals; and plain, grease-groove and felt shields.

A-113
Anon.
Flange Type Seals For Ball And Roller Bearing Application
Product Eng., V. 19, pp. 122-3, Nov., 1948

Flange-type seals are so designated because they use a flange shaped sealing member with a wiping lip that lightly presses against the shaft.
62 examples (by cross sectional drawing), material identification and linear speed.
Felt; asbestos; cork; oak leather; 700 lineal f.p.m.
Chrome leather, 1000 lineal f.p.m.
Compositions, 1000-20000 f.p.m.

A-114
Anon.
Carbon Packing Rings
Mech. Eng., V. 71, pp. 159-61, Feb., 1949

A method of manufacturing carbon packing rings such as those used in turbines that results in a uniform precision interchangeable sectional carbon ring.
The description of the operation is laid out in twelve steps. Photographs included.

A-115
Anon.
Various Designs And Methods Of Sealing Available For Fluid Conveying Systems In Machines
Machine Design, V. 21, pp. 106-109, July, 1949

Various types of fittings commercially available and their basic design characteristics. Cross-section views of 20 different types and how they seal.

A-116
Anon.
These Five Gasket Types Do Most Of The High Pressure Work
Chem. Eng., V. 56, pp. 118-20, August, 1949

Description and drawings of the "Budgeman" type, the "delta" type, Wave ring, O-ring, and Lense-ring gasket. Brief discussion of each.

A-117
Anon.
Labyrinth Seals
Aircraft Production, V. 11, pp. 289-290, Sept., 1949

Used in gas-turbine engine of the British Armstrong Siddeley Mamba for pressure sealing between rotating and fixed members. Forming equipment and procedure for manufacture of the "fins" used in these seals.

A-118
Anon.
Modern Oil Leaks
Sci. Lubrication, V. 1, pp. 2-5, Nov.-Dec., 1949

Description of Gaco M.I. oil seal for rotating shafts, which is manufactured by George Angus Co. Ltd., New Castle on Tyne, England; advantages of Gaco synthetic rubber material when employed in conjunction with "hairline" contact of sealing lip; manufacture and testing; proper case and fitting.

A-119
Anon.
Electric Motor Lubrication
Lubrication, V. 36, pp. 1-12, Jan., 1950

An excellent general description of motor bearing types and lubrication. Factors effecting selection of equipment and methods.

A-120
Anon.
Mechanical Seal Design
The Chemical Age, V. 62, p. 328, Mar. 4, 1950

T

This abstract of a booklet published by a packing manufacturer deals with three principal considerations which influence the performance of a mechanical seal, namely shaft conditions, material and wear.

A-121
Anon.
Aluminum Foil Laminate Gasket
Journal of Metals, V. 188, p. 839, June, 1950

An interesting gasket material has been developed, based on an aluminum foil laminate bonded with special synthetic resin. In use, its capacity for plastic deformation is such that it will readily take up all surface forms against which it is bolted, including machining marks.
General, brief discussion.

A-122
Anon.
How To Be Happy With Teflon Packing
Chem. Eng., V. 57, pp. 229-239, July, 1950

Selection, fabrication, fit and run in, are steps discussed in detail. Compounded, not pure for rotating shafts; proportioning pumps regulator valves; centrifugal pumps; agitators; reactors; autovalves and solid teflon gasket.

A-123
Anon.
Seal Makes Pump Self-priming
Steel, V. 127, p. 106, Aug. 14, 1950

Explains that the purpose of the mechanical shaft seal on self priming centrifugal pump is to prevent leakage of liquid or entrance of air where the rotating pump shaft enters the stationary pump housing.

A-124
Anon.
Plastic Packing
Blast Furnace And Steel Plant, V. 38, p. 1350, Nov., 1950

A special new feature to "Super Seal" plastic packing. A securely vulcanized tape-back is now offered as an integral part of all "Super Seal" spiral packings. Increased flexibility.

A-125
Anon.
Chemlon V-Ring Packing Molded Of Teflon
Modern Plastic, V. 28, p. 148, Feb., 1951

Product announcement of long-life frictionless valve seal provided by tetrafluoroethylene packing.
Photograph.

A-126
Anon.
Stuffing Box For High Pressure Orifice Meter
Machine Design, V. 23, p. 204, Mar., 1951

The heart of the stuffing box is a combination packing and bearing machined from Teflon which is tough and chemically resistant. 5000 psi - 450°F. Shaft torque only 9.7 gram-inches.
Photograph and discussion.

A-127
Anon.
Mechanical Seal
Blast Furnace And Steel Plant, V. 39, pp. 718-19, June, 1951

The new shaft seal performs effectively under severe temperature and corrosive conditions. The seal incorporates a flexible ring molded from Teflon. The Teflon "wedge-ring" enables this seal to combine the chemically-inert properties of Teflon with the flexibility and positive sealing components essential to effective mechanical sealing.

A-128
Anon.
High Pressure Stuffing Box
Chemical Engineering, V. 58, p. 187, Sept., 1951

Staged packing, with controlled pressure drop across each stage, provides long packing life, minimum shaft wear, and easy maintenance. Range 3000 to 5000 psi and 300 to 600°F.
Rotary Mechanical Seal.
Details of pressure and oil - stage to stage.
(Blaw Knox engineers)

A-129
Anon.
Simple Ways To Seal Slow Speed Bearings
Mill & Factory, V. 49, p. 109, Nov., 1951

Simple and inexpensive methods of sealing; labyrinth seals; endwise movement, counterboring housing can save space; long thin housings; illustrative drawings.

A-130
Anon.
Pressure Filled Metallic O-Rings For Static Seals
Machinery, V. 58, p. 199, April, 1952

Hollow metal tubing rings filled with inert gas at a temperature of 600 pounds per square inch are providing a new answer to static sealing requirements in a wide range of applications. Positive metal to metal static seals, wherever problems of heat, pressure, corrosive liquids or gasses are involved. Stainless steel, mild steel, cadmium or nickel plated, 20,000 psi. Available from 11/16" dia. to 40" dia. in increments of 1/16 inch.

A-131
Anon.
Sealing Cabin Wiring; Molded Multi-cable Plastic Grommets For Pressure-bulkhead Mounting
Aircraft Production, V. 14, p. 115, April, 1952

A-132
Anon.
Beater Saturated Gasketing Materials
Prod. Engrg., V. 23, pp. 164-166, June, 1952

Cellulosic or asbestos fibers treated with water dispersions of rubber or resin prior to being formed into sheet, provide the basis for new classes of gasketing materials.

A-133
Anon.
Spiral Wound Gaskets
Mech. Eng., V. 74, pp. 585-6, July, 1952

Spiral wound gasket of 347 stainless steel, 10" dia., with thickness of 0.175" with asbestos filler, temp. to 1800°F, and a stainless steel retainer ring. Presentation of test data and discussion of results. Drawing included.

A-134
Anon.
High Compressibility Fiber Gasketing Material
Automotive Ind., V. 107, p. 59, Sept. 1, 1952, Product Eng., V. 23, p. 220, Sept., 1952

Features of these materials are that they will not dry out, harden, or shrink on aging or in service; they have high compressibility even under light flange pressures; they will not rupture or crush under flange pressure loads of 100,000 psi; they are homogeneous and impervious and will hold internal pressures from 1000 to 2000 psi materials: cellulose fibers, finely ground cork, and synthetic rubber, asbestos. Data discussed.

A-135
Anon.
Teflon, Components And Coatings
Product Engineering, V. 23, pp. 149-153, Sept., 1952

Complete description of teflon, capable of continuous service at temperatures -90 to 500°F. Properties of gaskets and packing. Physical properties of filled teflon compositions. Teflon packing case histories.

A-136
Anon.
Analytical Derivation And Experimental Evaluation Of Short-bearing Approximation For Full Journal Bearings
N.A.C.A. Report 1157, 1953. (Incorporates NACA TN 2808+TN2809)

An approximate analytical solution including the effect of end leakage from the oil film of short plain bearings is presented because of the importance of endwise flow in sleeve bearings of the short length commonly used. Analytical approximation is supported by experimental data. Approximating a maximum bearing temperature, and evaluation of effect of deflection in misalignment on eccentricity ratio at the ends of the bearing, are discussed.

A-137
Anon.
Treat Your Compressor Seals Right
Power, V. 97, p. 122, Dec., 1953

Discussion of shaft seals, coil spring, bellows, and balanced packings. Also discussion of leaks, choosing the seal, and maintenance.

A-138 T
Anon.
Rotary Shaft Seal
The Engineer, V. 197, pp. 293-294, Feb. 19, 1954

This article describes a cartridge type mechanical face seal for shafts rotating at high speeds.
Standard models: - Shaft size, 1/4-inch to 7 1/4" dia.
- pressure to 50 lbs. /sq in.
- temp. to 130°C

A-139
Anon.
The O-Ring Spiral Failure Problem
Applied Hydraulics, V. 7, pp. 90-94, March, 1954

Discussion of O-ring failure:
Types of failures; probable cause; conditions which effect the normal load; conditions which effect static sliding and rolling coefficients; conditions which effect the contact area; other conditions that can accelerate or induce spiral failure and weaken the ring structurally; corrective action.

A-140 T
Anon.
Keeps Seal Tight At High Temperatures
Chemical Engineering, V. 61, p. 264, Oct., 1954

Stuffing box cooling system used to increase operating temperatures in pumps with mechanical face seals.

A-141
Anon.
Design Manual On Nonmetallic Gaskets
Machine Design, V. 26, pp. 157-188, Nov., 1954

Subjects covered are: 1) Gasket requirements, 2) Materials and uses, 3) Joint and gasket design, 4) Specifications and testing. Description, discussion, data tables, etc.
a) Asbestos products, b) Cork and rubber, c) Cork composition, d) Rubber and plastics, e) Paper, f) Leather, g) Composite and miscellaneous gaskets.
Applications - test - detailed

A-142 T
Anon.
Stopfbuchslose Zentrifugalpumpen (In German)
Konstruktion, V. 7, p. 204, 1955

This sort note describes a centrifugal pump which is sealed by the dynamic head created by a small impeller situated behind the main pump impeller.

A-143
Anon.
Performance Cycling - Test Of High Temperature Hydraulic O-Ring Packings
WADC-TN-WCLS-54-71, WADC, Wright Patterson A.F.B., Jan., 1955

This report describes the methods used to determine the mechanical performance of synthetic rubber O-ring packings in hydraulic fluids at temperatures up to +550°F.

A-144 B
Anon.
Oil And Water Seals In Kaplan Runners
Water Power, V. 7, p. 38, Jan., 1955

(Abstract of article in Technische Rundschau. 46, 26 July, 1954.)

A-145
Anon.
O-Ring Design Factors
Mech. World, V. 135, pp. 64-7, Feb., 1955

O-rings supply simple versatile seals for systems of all kinds containing fluids under pressure; generalized data in tabular form for rapid static or dynamic applications to flange, shaft or piston seals.

A-146
Anon.
Low Friction Fluid Seals
Engineering, V. 179, p. 153, Feb. 4, 1955

A near perfect seal under a pressure range of 0 - 4500 psi consisting basically of a "U" ring. Trade name "Twinset."

A-147
Anon.
Fluid Seal For Reciprocating Shafts: Compound U-Ring And Header
Engrg., V. 179, p. 769, June 17, 1955

A seal which provides a convenient and effective gland suitable for applications involving the reciprocating motion of a shaft is described.
A test is described which operated from 0 to 50,000 psi for 7,000 operations.

A-148
Anon.
Quad-rings Replace Regular O-Rings In Piston Mechanism
Rubber Age, V. 77, p. 550, July, 1955

A-149
Anon.
High Temperature Hydraulic Seal Research And Development Program, Status Report No. 1
Douglas Aircraft Company, Report Dev. 2030, September 1, 1955.

A systematic approach to high-temperature hydraulics is described which will minimize the possibility of overlooking a promising technique or material, make full use of test facilities, and extract maximum information from test data. Evaluation at high temperature (550°F). Material will be based on the results of the following tests: life cycling, comparative screening, friction, physical properties after hot oil aging or cycling.

A-150
Anon.
High Temperature Hydraulic Seal Research And Development Program, Status Report No. 2
Douglas Aircraft Company, Report Dev. 2030, Nov. 1, 1955

A progress report on the overall project. The program has been divided into two phases, one phase covering 400°F packings for systems utilizing OS-45 fluid, and the other phase covering 550°F packings for use with MIL-0-8200 fluid. An attempt will be made to develop a seal applicable for the glands described in MIL-P-55144. If early indications are such that this approach appears impractical, then standard and non-standard seal configurations will be investigated.

A-151
Anon.
Better Static Sealing
Automation, V. 2, No. 11, pp. 75-76, Nov., 1955

Methods applicable in design of fluid power or control system for aircraft or other installations; in automation equipment, as in aircraft industry, more designs are incorporating O-ring type of sealing where possible; efforts to obtain O-ring quality seals without attendant fitting problems.

A-152
Anon.
High Temperature Fluid Packing And Operating Problems
Applied Hydraulics, V. 8, pp. 74-78, Dec., 1955

General comments on papers presented at fall meeting of SAE-A-6-Committee, Aircraft and Hydraulics and Pneumatic Equipment.

A-153
Anon.
High Temperature Hydraulic Seal Research And Development Program, Status Report No. 3
Douglas Aircraft Company, Report Dev. 2030, Jan. 1, 1956

The following categories of seals are being investigated:
1. Standard AN 6227 configuration O-rings designed for standard glands, 2. Modified O-rings configuration designed for decreased tension in standard glands, 3. Non-standard configuration seals, 4. O-ring back-up configurations and design, and packing and gland configuration which will take full advantage of elastomer materials.

A-154
Anon.
High Temperature Hydraulic Seal Research And Development Program, Status Report No. 4
Douglas Aircraft Company, Report Dev. 2030, March 1, 1956

Compounding studies with acrylic rubbers have been continued during the two-month period. Some of the work has been completed and is reported in full. Stress/strain test data are presented on acrylic stocks stressed at 400°F. Stress/Strain values of acrylic rubbers, aged 168 hours in OS-45 at 400°F and stressed at that temperature are low.

A-155
Anon.
Frozen Slugs Seal Shafts
Chem. Engg, V. 63, pp. 124 and 126, March, 1956

How it works, practical considerations, practical experience, limitations and designs.

A-156 T
Anon.
Nuclear Reactors Pose New Problems In Mechanical Design
Product Engineering, V. 27, pp. 197-8, April, 1956

Static seals, and frozen seals for liquid metal pumps are described. Friction heating maintains a liquid film, but high conduction coefficient of liquid metal does not allow the film to become big enough to leak.

A-157
Anon.
Can Liquid Gasketing Cut Your Costs?
Iron Age, V. 177, p. 103, April 19, 1956

Liquid gasketing and insulating material may offer economies - particularly if the present method entails cutting material to size, fitting, and securing by suitable adhesive agents. New material is called Vynafoam Plasticol. Offered by Inter-Chemical Corporation.

A-158
Anon.
High Temperature Hydraulic Seal Research And Development Program. Status Report No. 5
Douglas Aircraft Co., Report No. Dev. 2030, May 1, 1956

Physical properties of stocks previously reported are presented. Thermal activation techniques, such as employed with butyl rubbers, may have some merit, although improvement in properties of these formulations is not large, use of silicone polymers as plasticizers and mill release agents is of doubtful value since physical properties are poorer than stocks formulated without them. Microcel A does not offer the reinforcement of carbon black in Hycar 4021. However, volume increase (swelling) is not one-half that of stocks containing carbon black. Varying the type of amine produces differences in vulcanizate properties after OS-45 fluid immersion at 400°F.

A-159
Anon.
Tubular Rubber Gas Seal
Engineer, V. 201, p. 648, June 8, 1956

"Tubeseal" system developed for sealing floating roof tanks for large petroleum storage tanks. Drawing shows seal at flotation level so there is no vapor space. Description, in details, of ribbed scuff band and tubular seal. 80 ft. dia., 20-200°F. (Tank 80 ft. dia., 36 ft. high butadiene acrylonitrile synthetic rubber and nylon.)

A-160
Anon.
A Summation Of The Pneumatic High Temperature Seal Evaluation Program
Walter Kidde & Co. Inc., Belleville, N.J., June 12, 1956

Describes a procedure for testing O-rings at 3000 psi pressure and test cycles of 4 hours, as a preliminary selective step, aiding in sample selection for further test.

A-161
Anon.
High Temperature Hydraulic Research And Development Program Status. Report 6
Douglas Aircraft Co., Report No. Dev. 2030, July 1, 1956

Work with Hycar 4021, Acrylons EA5, and BA12, neoprene and nitrile type rubbers has been discontinued. Basic polymer deficiencies precluding their use in high temperature applications are discussed. Compounding studies with Fluororubber IF and Kel-F Elastomer has been continued. Philprene VP-15, Philprene VP-25 and Philprene VPA are of interest, but all of them contain an unsaturated Diene backbone.

A-162
Anon.
Metal Boss Seals
SAE Journal, V. 64, p. 85, July, 1956

Metal boss seals give good service in 3000 psi - 400°F hydraulic systems if the parts they seal are dimensionally accurate. Inaccurate fittings harm seals. Description and discussion. Drawings.

A-163
Anon.
O-Rings May Answer Your Gland Leakage Problems For Valve And Pump Maintenance
Oil and Gas Journal, V. 54, pp. 263-4, July 30, 1956

A-164
Anon.
Elgin A.F.B., Climatic Laboratory Cold Weather Test of Teflon Back-up Rings As Installed In The Hydraulic And Pneumatic Systems Of F7U-3 Airplane Report #1, Final Report
NATC PTR AC-1303.13.1, Naval Air Test Center, Patuxent River, Md., Aug. 1, 1956

This report covers the cold weather evaluation of the teflon back-up ring installed in lieu of the AN 6291 leather back-up ring in the universal fittings of the pneumatic and hydraulic systems down through -65°F.

A-165
Anon.
High Temperature Hydraulic Research And Development Program. Status Report No. 7
Douglas Aircraft Company, Report No. 2030, Sept. 1, 1956

An apparatus for tensile testing of ring specimens while immersed in hot fluids has been designed. Screening test results at 400°F. In addition, a teflon O-ring was run with the same test conditions. These tests were conducted after certain modifications of the screen machine; modifications described in status report No. 1

A-166
Anon.
Seals For Aircraft Gas Turbines
SAE Jour., V. 64, pp. 46-50, Sept., 1956, based on SAE papers 685, 686, 687 and 688, by J.B. Stevevens, J. Palsulich, J. W. Pennington, R.L. Johnson and others in a symposium on High Temperature Dynamic Seals at the Jan. 12, 1956, meeting.

Positive clearance or labyrinth, shaft, bore, and face seals.
Lists problems in main shaft bearing lubrication systems, containing or controlling air flow within proper channels, and sealing accessory drives. Also included is the problem of selecting and designing fluid flow barriers against rotating shafts in gas turbines.

A-167
Anon.
Improved Gasketing Material
Materials and Methods, V. 44, p. 149, Sept., 1956

Discussion of properties of teflon impregnated felt used as a gasketing material.

A-168
Anon.
High Temperature Hydraulic Research and Development Program. Status Report No. 8
Douglas Aircraft Company, Report Dev. 2030, Nov. 1, 1956

Work with Fluororubber SF-4 and Kel-F polybends reported in report No. 6, was continued. Formulas and physical properties for twenty-two additional stocks are presented. Seventeen stocks were prepared by varying the compounding ingredients and modifying the amounts added to the blend of Kel F and SF4. Five compounds are modifications of the time and temperature of cure of three of basic seventeen.

A-169
Anon.
High Temperature Hydraulic Research And Development Program. Status Report No. 9
Douglas Aircraft Company, Report No. DEV 2030, Jan. 1, 1957

O-rings made from five rubber stocks based on Hycar 4021 and 1072, and four polybends of Kel-F-3700 and IF-4 elastomers were turned over to the Propulsion and Mechanical Section for O-ring cycling and chew testing. Rings made from Hycar base compounds are being recycled to see if variations in milling and molding techniques would enhance mechanical performance.

A-170
Anon.
Non-corrosive Seal For Shafts
Engineering, V. 183, p. 90, Jan. 18, 1957

"Fluoseal," a mechanical seal for rotating shafts, made entirely of "Fluon." Its use is for sealing acids, strong oxidizing or reducing agents, and organic compounds. Normal range 20 inches vacuum to 50 lbs/sq. in., - 40°C to + 80° C.

A-171
Anon.
Improved High Pressure Packing
Tech. News Bulletin N.B.S. 41, 22, Feb., 1957

N.B.S. scientists have successfully used polytetrafluoro-ethylene impregnated with 5% by weight of molybdenum disulfide as a packing material in the high pressure morringseal of differential area pressure intensifiers. Sealed effectively at pressures approaching 200,000 psi. Description and test observations are discussed.

A-172
Anon.
What A Gasket Is And How It Works
Petroleum Processing, V. 12, No. 3, pp. 96-110, March, 1957

Features of confined, unconfined, partially confined, self-confining, and self energizing joints; temperature and nature of fluid or gas as factors in choice of gasket material; materials available; construction of metal gaskets; yield stress; gasket design for special purposes.

A-173
Anon.
Sealing Drives And Anti-friction Bearings
Design News, V. 12, No. 7, pp. 156-7, April 1, 1957

A-174
Anon.
Superfinished Seal Face Cuts Axial Force
Design News, V. 12, No. 8, pp. 30-1, April 15, 1957

By using Alnico-5 for the stationary housing ring and magnetic stainless steel with a lapped carbon sealing face for the shaft ring, a mechanical seal design which eliminates springs has been produced. The stationary ring must be insulated from the magnetic housings, but all inherent disadvantages of springs are claimed to be elminated.

A-175
Anon.
Development Of Coupling Seal Assemblies For Corporal Missile Propulsion System
T & R/GMD ERS 505-19, Firestone Tire and Rubber Co., May, 1957, Dayton, Ohio

A-176
Anon.
How To I all A Packing On Air And Oil Powdered Equipment
Applied Hydraulics, V. 10, pp. 104-109, May, 1957

Installing packings presents certain problems to both the designer and maintenance man. Tese problems pertinent to O-rings, V-rings, U-rings and cup packings are discussed.

A-177
Anon.
Improved Ring Gasket For API Flanges
Petroleum Engr., V. 29, No. 5, pp. B111-112, May, 1957

New seal ring for API flanges is now higher than conventional octagonal or oval rings in crosssection, and is provided with sealing flats on both inner and outer surfaces; outside diameter of ring is slightly larger than ring groove.

A-178
Anon.
Metal Seals Solve Missile Leakage Problems
Aviation Age, pp. 49-51, May, 1957
(Skinner Seal Co.)

Use of Skinner Inconel "X" flexible metal seal.

A-179
Anon.
Testing Packings On A Plane-built Machine
Applied Hydraulics, V. 10, p. 86, May, 1957

A description of a machine designed by the Warner and Swasey Co. of New Philadelphia (an earth-moving equipment co.) for testing piston head and piston packings.

A-180
Anon.
Parker O-Ring Handbook
Parker-Hannifin Corporation, Cleveland, Ohio, Nov., 1957

A-181
Anon.
Sealing Means For Receptacles Containing Metal In Liquid Or Fused State. (Swedish)
British Patent 780,151, and Nuclear Eng., V. 2, p. 538, Dec., 1957

A ring of soft wire laid between flanges of the receptacle is separated from the liquid metal by a cushion of inert gas introduced into a chamber between the sealing means and the interior of the receptacle.

A-182
Anon.
Handbook Of O-Ring And Dyna-seal Packings
Rubber Products Corporation, Dayton, Ohio, 1958

A-183
Anon.
High Pressure Well Head Equipment
Mechanical Engineering, V. 80, pp. 62-68, March, 1958

Design of pressure - energized gaskets for use at very high pressures, ambient temperatures (15,000 psi working flanges).

A-184
Anon.
Elastomer Seals Hot Fluids
Iron Age, V. 181, p. 103, April 10, 1958

Rubbers compatible with high temperature and fluids are hard to find.
Viton, a development of E.I. DuPont de Nemours and Co. is good from -60° to 350°F and is compatible with solvents, fuels, lubricants, hydraulic fluids, acids and bases.

A-185
Anon.
Viton: Your New Design Material
Power, V. 102, No. 6, pp. 117-119, June, 1958

New DuPont synthetic rubber for service in oils, fuels and solvents at over 400°F, is linear copolymer of vinylidene fluorine by weight; physical, mechanical and electrical properties; most important applications include aircraft seals, pump seals, fuel cells, etc.

A-186
Anon.
Cooled Mechanical Seal
Engineering, V. 186, p. 196, Aug. 15, 1958

A mechanical seal that will operate at temperatures in excess of 800°F and pressures up to 800 lbs/sq. in. developed by Flexibox Limited, Nash Road, Trafford Park, Manchester. Normal water cooled jacket.

A-187
Anon.
Proceedings (of the) Joint Army-Navy Air Force Conference On Elastomer Research And Development (No. 5) V. One.
Materials Lab., Wright Air Development (W.P.A.F.B.)
PB182031, Off. Tech. Ser. 283 p. 42, ref.
Oct., 1958

Testing of fluid seals for extreme environments. Evaluation of elastomers in H.E.F. Methods for for studying behavior of elastomers in aircraft fuels at high temperature. O-rings for cylinder liners of diesel engines.

A-188
Anon.
Technical Memorandum No. 8; The Effects Of Nuclear Radiation On Seals, Gaskets And Sealants
Radiation Effects Information Center - Battelle Memorial Inst., Nov. 30, 1958

A-189
Anon.
Universal Ball Bearing Seal
Engineering, V. 187, p. 294, March 6, 1959

A synthetic rubber seal with a steel reinforcement which can be fitted very easily to any standard sized bearing race that has the necessary groove. A distinguishing feature of the new seal is that the lip is "hinged", in effect, by an annular thinning or "waisting" of the synthetic rubber a short distance back from the lip itself.

A-190
Anon.
X-15 - Hydraulic Seals
Aircraft and Missiles Mfg., p. 32, June, 1959

Discussion covering the application of teflon as extrusion barrier for O-rings in the hydraulic system.

A-191
Anon.
High Temperature And Pressure Pipe Connections
The Engineer, V. 207, p. 930, June 12, 1959

Commercially available, two-bolt coupling with "bayonet" seal. Description, photograph. -175°F to +1000°F. Unions for 1/4" to 1 1/4" pipe dia. Two-bolt connections for 1 1/2" to 30" pipe dia.

A-192
Anon.
Magnetic Shaft Seals Gaining Favor
Chemical Engineering, V. 66, p. 69, Sept. 21, 1959

Favorable commentary on magnetic shaft seals; small size, light weight, elimination of adjustment problem of conventional seals. Application of heavy rhodium plate to alnico magnet eliminates corrosion.

A-193 T
Anon.
Shaft Seals For Compressors And Turbines For Gas Cooled Reactor Application
Proceedings of meeting at Oak Ridge National Laboratory, published as USAEC report TID-7604, Dec. 16-17, 1959.

This publication is a record of the papers presented at this conference and the reproduction of recorded discussions.
There are twenty papers in all.

A-194 T
Anon.
Army Gas-cooled Reactor Systems Program - Semi Annual Report
Aerojet-General Nucleonics Report, Jan. 1 - June 30, 1960

Gas-buffered labyrinth type seals employed on the lubrication system of two-turbine compressor unit designs.

A-195 T
Anon.
Investigations Of Types Of Seals For Main Coolant Pumps For Large Pressurized Water Reactor Nuclear Plants
Ebasco Services Inc., Report NYO-9321, June, 1960, U.S. Atomic Energy Commission, Contr. No. AT(30-1)-2547, 80 pages

Report on study results of the comparative reliability and installed and operating cost of canned-motor and controlled leakage pump designs for application in a pressurized water reactor installation today. The scope and adequacy of research and development programs on seals leaves much to be desired.

A-196
Anon.
Sealing Against Pressure And Vacuum
Chem. Eng. V. 5, pp. 652, 654, 656, 658, Sept. 1960

Reviews are given of several papers concerning sealing application in pressure and vacuum systems.

A-197
Anon.
Teflon O-Rings Find New Uses As Both Dynamic And Static Seals
Power, V. 104, p. 249, Dec., 1960

A short article on the advantages, use, properties, and characteristics of teflon O-rings

A-198 T
Anon.
Compressed Air And Gas Handbook, Third Edition
Published by Compressed Air and Gas Institute, New York, 1961

This reference book is a practical handbook on all phases of industrial gas compressors and compressed air powered portable tools and rock drills used in industry. Pages 3-46 through 3-50 contain a description of the commonly used compressor shaft seals. Included are soft packings, labyrinths, restrictive rings, seals, liquid film seals, and mechanical film seals.

A-199 T
Anon.
Evaluation Of Helium Seal Selected For Nuclear Coolant Compressors In Gas Cooled Power Reactor Res. & Dev. Program
Allis Chalmers Manufacturing Co., Jan., 1961

Final report covering development and test work on the floating-bushing buffered shaft seal desi[illegible] for the Experimental Gas Cooled Reactor proj[illegible]

A-200
Anon.
Rubber Pump Wall Is Seal And Spring Too
Product Engineering, V. 32, No. 2, pp. 60-61, Jan. 9, 1961

This multi-purpose ring eliminates costly machi[illegible] in fuel injection pump. Speed of the engine and manifold pressure set correct fuel-flow rate for a[illegible] load.
Synthetic rubber ring combines three functions: 1) seals pump cylinder with no need of expensive machining; 2) serves as a spring to keep plunger roller follower in contact with cam; 3) pumps fuel when compressed by plunger.

A-201
Anon.
The Seals Book
Machine Design (Spec. Issue), pp. 1-136, Jan. 19, 1961

Handbook to aid design engineers in selecting and applying seals, packings, and gaskets.
Chpt. 1 - Introduction
2 - Felt Radial Seals
3 - Radial Positive-Contact Seals
4 - Exclusion Devices
5 - Clearance Seals
6 - Split Ring Seals
7 - Axial Mechanical Seals
8 - Circumferential Seals
9 - Simple Compression Packings
10 - Molded Packings
11 - Diaphragms
12 - Nonmetallic Packings
13 - Static O-Ring Seals
14 - Metallic Gaskets

A-202
Anon.
Composite Metals Form Static Seals
Chemical and Engineering News, V. 39, p. 56, Feb. 20, 1961

Armour Research Foundation is swinging into final development of a static seal that will take pressure cycling from 0 to 5000 psi, at temperatures to 1500°F. The seal will probably be a composite of two metals; one hard, one soft - since work so far at ARF in developing seals that will take these pressures up to 1200°F has gone in this direction. Description: Flame & filler technique. Molybdenum, stainless steel, silver, indium/silver alloy.

A-203
Anon.
Precision Upgrades Oil Seal Performance
Steel, V. 148, No. 9, pp. 132-133, Feb. 27, 1961

Some recommendations resulting from a three year study program of seals and packings. Particularly application of lip seals.

A-204
Anon.
Fiber Metal Passes Tests In Seals And Gaskets
Steel, V. 148, p. 136, March 13, 1961

Composite materials that combine the desirable characteristics of their components are said to be suitable for static seals at temperatures up to 1200°F, and pressures to 5000 psi. The materials are also being tested in dynamic seal applications. Resilience of skeleton combines with filler conformability, providing an effective seal for most static, high temperature, high pressure applications.

A-205
Anon
Fiber Metal Seals Take 5000 Psi, 1200°F
Materials in Design Engineering, V. 53, p. 17, April, 1961

Experimental static seals have been made by impregnating strong fiber metal skeletons with soft metal fillers; Armour Research Foundation for W.A.D.C. Short descriptive article.

A-206
Anon.
Petrol Pump Has Viton Seals
Engineering, V. 191, p. 475, April 7, 1961

First-class solvent resistance and dimensional stability are two reasons for choice of Viton rubber in new British-made petrol dispensing pump, says Du Pont.
Operational results are described.

A-207
Anon.
Metal Fiber Seals
Engineering, V. 191, p. 476, April 7, 1961

Short description of high-temperature high-pressure static seals, developed for the Air Force, made by impregnating a porous body of fibre metal skeleton with a soft metal (Armour Research Foundation). Molybdenum fibre and silver. Pulsating pressures 0 - 5000 lb/sq. in. at 650°C. Brief test, result discussion. By Republic Aviation.

A-208
Anon.
Wire Vacuum Seals
New Scientist, V. 10 (230), p. 37, April 13, 1961

A high vacuum system that makes use of aluminum wire gaskets in place of conventional rubber seals makes it possible, it is claimed, to achieve a ten-thousand fold reduction in pressure. The seal is made secure by placing the wire gasket between steel flanges and heating the joint to about 360°C. Frictional rubbing then occurs owing to the expansion of the gasket.

A-209
Anon.
Research Attacks Seal Leakage: Three Step Approach Unveils Failure Causes
Iron Age, V. 187, pp. 72-74, June 1, 1961

Considerations in the research. Discussion and data.
1) Lip Seals 4) Joules effect
2) Shaft surface 5) Trim interference factor
3) Torque factor 6) Raw materials
Description of "Sealactor" testing machine

A-210
Anon.
Performance Of Some Modern Gasket Materials
Mechanical World and Engineering Record, V. 141, pp. 208-210, June, 1961

The range of modern gasket materials available provides ample scope for choice of compressibility sealing pressure, physical performance and chemical inertness. The basic factors governing the performance and selection of resilient seal materials and the properties of some typical modern compositions are given.

A-211
Anon.
Atomic Reactor Uses Silicone Rubber Seals
Rubber World, V. 144, p. 106, June, 1961

A-212
Anon.
Mechanical Seal For High Speed Shafts
Engineer, V. 211, p. 916, June 2, 1961

Face-type balanced multi-spring design. For applications with restricted axial space. For peripheral speeds to 18,000 fpm, pressures to 1000 psi and temperatures to 500°F.

A-213
Anon.
Pumps For Water-cooled Power Reactors
Nucleonics, V. 19, No. 7, pp. 55-63, July, 1961

Describes reactor system pumps in use or planned. Also brief summary of AEC seal development programs.

A-214
Anon.
Eight Instrument Seals
Product Engineering, V. 32, pp. 84-85, Sept. 18, 1961

Sketch, and three-line description of each of eight seals for instrument cases. Besides keeping water and dirt out, they must also strengthen the rim of the instrument case.

1) Offset 2) Extrusions 3) Flange 4) O-Ring
5) Bead 6) Space Saver 7) Tubular Extrusions
8) Chevron

A-215
Anon.
Speciality Rubbers For Special Uses
Chemical Engrg., V. 68, pp. 204, 206, 208, Sept. 18, 1961, and pp. 132, 134, 136, Oct. 2, 1961

Design information for gaskets and seals made from the new heat and chemical-resistant elastomers. These include Viton, Hypalon, Fluorel 2141, Kel-F, and various silicone rubbers. A table of their resistance to various solvents and corrosive chemicals is added.

A-216
Anon.
Tube Connections, A State Of The Art Survey: Flanges, Seals, And Joints
Report No. 1068, Chrysler Corporation Missile Division, Huntsville Operation, Oct. 20, 1961

IG Pneumatic Test of soft material sealed flared fittings. Flanges, seals, and joints (pp. 131-189). Seal materials: seals for V band coupling flanges; helium seals.
Temperature - energized seals for liquid hydrogen; pressure energized seals; high temperature hydraulic seals; seals for hot gas servo systems.
Appendix II. Misc. reports on O-rings and sealants.

A-217
Anon.
Pumps
Lubrication, V. 48, pp. 17-32, Feb., 1962

Liquids which constitute a large portion of this planet are conveniently moved by means of pumps.
Many general purpose and special pumps are described. Among the latter are jet pumps, gas lift pumps, hydraulic ram, electromagnetic and canned pumps. Stuffing boxes or mechanical seals are used on many of these pumps and they are discussed. Electromagnetic or canned pumps have no seals. The pumping action is obtained by electromagnetic means, either direct or indirect, depending upon whether or not magnetic or non-magnetic fluids are being pumped.

A-218
Anon.
Magnetic Shaft Seals
Engineering, V. 193, P. 260, Feb. 23, 1962

The seals consist of a magnetized ring with a sealing surface fixed in a stationary manner to the housing, and a second ring also with a sealing surface coupled to the shaft by means of an O-ring. This second ring moves axially along the shaft in response to strong magnetic attraction, firmly engaging the first ring. Sealing surfaces are held in contact by magnetic forces when no fluid pressure exists and eliminates a troublesome spring.

A-219
Anon.
Sticky Vacuum
Time, p. 51, April 13, 1962 (National Res. Corp.)

Seals, materials

A-220
Anon.
Magnetic Particles Seals
Mech. World, V. 142, pp. 172-3, June, 1962

Flexible membrane consisting of finely divided iron particles suspended in oil and constrained in air gap by magnetic field can be used as efficient shaft seal for stationary, rotating, or reciprocating actions.

A-221
Anon.
User's Comments On Rubber/Metal Gaskets (Hyclad, Made by Fireproof Tanks)
Engineering, V. 194, p. 50, July 13, 1962

A gasket material of metal sandwiched between layers of rubber has proved itself able to withstand abuse and to seal difficult joints. Consists of 26 swg aluminum sheet (L 72) sandwiched between two layers of nitrile rubber. Aluminum can be varied from 12 to 30 swg.
Sealing electronic equipment; integral fuel tanks; washers for "all metal" couplings; flameless pipe couplings. Discussion of application and drawings included.

A-222
Anon.
Rubber O-Rings For Automotive Seal And Packing Applications (Reprint from SAE Handbook)
SAE paper, H.S. 212, September, 1962.

A-223
Anon. (French Patent 1,260,206), Mar. 27, 1961
Removable Fluid Tight Closure Means For A Hole Provided In A Wall Separating Two Fluids, In Particular Inside A Nuclear Reactor
Nuc. Sci. Abst. 28500 (V. 16, No. 20, p. 3726, Oct. 31, 1962).

Closure piece inserted into the channel forming the hold, so that the force presses sealing surfaces of the closure pieces against sealing shoulders provided in the wall and in the closure piece.

A-224
Anon.
A Gasketless Joint For Application To High Temperature, High Pressure Water Systems
Naval Research Lab., Nov. 4336, AD 27384

Describes development of a joint with a metal gasket. Single profile type.

A-225
Anon.
Design Data For O-Rings And Similar Elastic Seals Part 2
WADC-TR 56-272, Boeing Airplane Co., Seattle, Washington

This study is to gain knowledge of the relationship between the physical properties of seal materials and sealing efficiency so that materials may be fully utilized in seal design. Commercially available polymers have been compounded with various physical properties. Static annulus, rotating shaft, and reciprocating shaft functional test jigs have been designed and manufactured. Seals tests have been run in these jigs.

A-226
Anon.
Dynamic Sealing: Theory And Practice
Koppers Company, Inc., Metal Products Division, Baltimore, Md.

This booklet is a collection of six papers on dynamic seals, showing the application of the basic equations of fluid mechanics to the problems of sealing; discussion of labyrinth seals, clearance seals and split ring seals. Typical applications and performance data are presented.

A-227
Anon.
Radial Sealing Rings: Shaft Packings
(Trans. of Verein Deutscher Ingenieure, Zeitschrift, West Germany, V. 97, No. 7, pp. 226-230) S.L.A. Translation 59-20843

A-228
Anon
Rubber Packing, Or Joints, For The Conning Tower Hatchway Cover
PB 46381, Off. Tech. Ser., 2 pages

A cross sectional drawing of a water-tight rubber seal and fitting for a conning tower hatchway cover is shown.

A-229
Anon.
Standard Flange Seal
Raco Engineering

Standard Flange Seal

A-230
Anon, D.
A Mechanical Seal For Very High Pressures
J. Sci. Instr., V. 32, p. 446, Nov., 1955

A piston seal is described and illustrated for use up to 15,000 Atm. of pressure with alc. as the working fluid.

A-231
Apkarian, H.
Gas Leakage Tests On Shaft Seals
R49GL74, GEL Tech. Data Center, G.E. Co., Sch'dy, N.Y., 14 pp., Oct. 26, 1949

Progress report on two types of rotary shaft seals. The seals tested were a face type seal and one was an original design. Tests run on 5/8" shaft over a range of speeds and duty cycles. Evaluation based on gas leakage rates while varying time, speed, and duty cycles.
Several types tested. Best double face type seal with water circulated through it. Leakage rate less than 1CC per year.

A-232
Apple, F. C.
Leakage Of Water From Valve Stuffing Boxes
U.S. A.E.C. Res. & Dev. Report D.P 538, 12 pp., 6 Fig, Jan., 1961

A-233 T
Asanuma, T.
Study On The Sealing Action By Viscous Fluid (In Japanese)
Trans. Japan. Soc. Mech. Engrs., V. 17, No. 60, pp. 119-125, 1951

For the purpose of obtaining a sealing method of rotary shaft penetrated into a vacuum, or high pressure vessel, the author proposes to prevent the leakage by viscous fluid which has a very high pressure due to the pumping action of the rotary shaft with a helical screw groove. In this paper he studies theoretically the pump performance under the laminar theory and it is clear that to obtain the maximum pump efficiency, some dimensions of the screw groove are to be selected as follows:
a) the width of the thread to be as narrow as possible
b) the ratio of width to depth of the groove should be equal 5-10:1
c) the ratio of a gap to the depth of the groove should be 0.1-0.2:1
d) the helical angle of the screw thread to be about 10-20 degrees.

A-234 T
Asanuma, T.
On The Flow Of Liquid Between Parallel Walls In Relative Motion (In Japanese)
Trans. Japan. Soc. Mech. Engrs., V. 17, No. 60, pp. 140-146, 1951

The author discusses some theoretical formulae for the turbulent flow through a fine clearance space (δ) between two parallel walls, one of them being in relative motion to the outer at arbitrary angle to the direction of the pressure gradient. Therefore he applies this formula to the flow through the annular space formed by two concentric pipes in relative rotation, and then obtains an empirical formula for the turbulent flow in such a case using the experimental data of R. J. Cornish.

A-235 T
Asanuma, T.
Studies On The Sealing Action Of Viscous Fluids
Int. Conf. on Fluid Sealing, paper A3, 21 pages, British Hydromechanics Research Assn., April, 1961

An extensive report on the theoretical and experimental study of the screw viscosity pump when considered as a seal.
In Part I - the pumping and sealing performance of screw pump having various thread or groove forms are treated theoretically in effort to ascertain the most desirable design geometry. Also, the confirming experimental data and conclusions, using mineral ore and grease.

A-236
Ashmead, R. R.
Static Seals For Missile Applications
Jet Propulsion, pp. 331-340, July, 1955

Excellent general discussion of seals and sealing problems. Seals are classified as: (1) plastic, (2) rubber O-ring, (3) expanding seal compounds, (4) soft and flexible metal.

A-237
Asp, H.
How Teflon Packings Behave Where Temperatures Vary
Applied Hydraulics, V. 10, pp. 127-132, May, 1957

Teflon has a low coefficient of friction and excellent properties over a wide range of temperatures. However, its thermal expansion and shrinkage, where temperature variations are 100°F or more, can cause leakage. A production technique is described which controls this.
The result of the application of this technique are "memorized" packings.

A-238 T
Aspelin, L.L.
Shaft Seal For Alternate Wet And Dry Operation
U.S. Patent 2,957,711, Oct. 25, 1960

A shaft seal for use where it is desirable to maintain continuous shaft rotation under the intermittent presence and absence of a sealed fluid. The pressure of the sealed fluid maintains sealing contact while the fluid acts as a lubricant. When the fluid pressure is removed, sealing contact is no longer obtained and wear does not occur.

A-239 T
Atkins, B.R., Bradshaw, J.R., Mitchell, P.J.
New Carbon Materials For Mechanical Applications
Int. Conf. On Fluid Sealing, Paper F3 (8 pages)
British Hydromechanics Research Association, Harlon, Essex, England, April 1961

Description of a series of molded carbon products containing various additives such as polytetrafluroethylene, and molybdenum disulphide. These products are designed for applications in which normal lubricants fail or are absent. Some physical properties are listed, i.e., thrust ring, experimental apparatus seal, medium of air. Thrust 12 lb/sq in, stator cast iron, 600 hour run, alternate material 60 hour run.

A-240 T
Atkinson, R. P.
Piston Ring Turbine Shaft Seal
U.S. Patent 2,812,196, Nov. 5, 1957

A split piston ring, contained in a groove on a rotating shaft sleeve, made to spring against a stationary sleeve. The split in the ring is inclined outward with rotation of the shaft, so that oil attempting to flow inward will be picked up in the ring gap and forced outward. A visco seal of short length.

A-241
Atkinson, W.B.
Gasket Materials For Electrical Equipment
Prod. Engrg., V. 20, pp. 124-127, Dec., 1949

Eleven types of gasketing material suitable for electrical equipment. General advantages and disadvantages of each type. Joint design using cork-neoprene, rubber bonded asbestos, and others.

B-001
Babb, S.E.
Combined Bridgman and O-Ring Static Pressure Seal
Rev. Sci. Intr., V. 31, p. 219, 1960

A seal adopted for use in high-pressure applications up to 12,000 atm. The seal assembly is illustrated and consists of an extrn. ring, O-ring, Teflon back up ring, and an unsupported area seal ring.

B-002
Bailey, E.C., Schroeder, H.C., Carlson, D.H.
Operating Experience with Mechanical Joints in High-Pressure High Temperature Steam Piping
A.S.M.E. Transactions, V. 75, pp. 97-101, Jan. 1953

Two types of mechanical transition joints were installed at the Ridgeland Station of the Commonwealth Edison, Chicago, Ill. The joints, pressure seal and bellows type, were designed for operation on steam at 1800 psig and 1050°F.
The pressure seal joints have been in service one year and five months, the bellows type joints one year.
Pressure joint seal - gasket soft steel, silver plated.
Bellows type seal - gasket cobalt base alloy L-605.
Photos, drawings, test curves data included.

B-003 T
Bailey, J.F.
Analysis and Design of the Frozen Seal
Oak Ridge National Laboratory, Report No. ORNL 2110, Feb. 12, 1957

Theoretical and applied aspects of a seal in which the liquid whose leakage is to be prevented is frozen to form its own seal. A frozen sodium pump seal is used as an example.

B-004
Bailey, J.F.
Analysis and Design of the Frozen Seal
Oak Ridge National Lab., Tenn. ORNL-2110 Contract NO W-7405-eng-26, 11 pp., March 8, 1957

B-005
Bailey, J.M., Swikert, M.A., and others
Wear of Typical Carbon-base Sliding Seal Materials at Temperatures to 700°F.
NACA TN 3595, Feb. 1956, 22 pp.

Wear and friction studies were made to show the effects on performance of temperature, type of mating material, and minor composition changes in typical carbon seal materials. Most data were obtained at a surface speed of 10,000 feet per minute, a load of 1000 grams on a 3/16 inch-radius specimen, and temperatures to 700°F.

B-006
Baker, D.
An All Metal High Vacuum Valve
Vacuum, V. 12, pp. 99-100, March/April 1962

Valves which employ a knife edge seal have been developed which offer significant advantages compared with other types of all metal valves. A low initial closure torque of 2.5 lb-ft. for a valve of 2.5 cm operation has remained unchanged after 3000 operations without measurable increase in the closed conductance of $< 10^{-14}$ l/sec. Since closure torque is directly proportional to the valve aperture, the design may be used for large apertures. (Copper gaskets and knife edge seal)

B-007
Baker, J.R.
Steam Turbine Shaft Gland
Power, V. 53, pp. 881-3, May 31, 1921

Why packing is necessary
Carbon ring packing seals in the shaft glands
Labyrinth packing
Drawings and pressure loss data curves.

B-008
Ball, W.P., Cheyney, L.E., and Clark, F.E.
Molding of O-Ring Packings
Rubber Age (N.Y.), V. 62, pp. 652-4, 1948

A newly developed method for testing the phys. properties of O-ring hydraulic packing is described and illustrated.

B-009
Ballif, J.L.
A High Vacuum Application of Inflatable Seals for the Processing Refabrication Experiment
Atomics International Div., North American Aviation, Inc., Canoga Park, Calif., NAA-SR-2544, Contract AT-11-1-GEN-8, 47 pp., Jan. 15, 1959

A method was developed for sealing the massive shielded doors of the vacuum access lock in the PRE cell complex with inflatable rubber seals of the type used by the aircraft industry. Leak rates of less than 1 ft^3/day were maintained.

B-010
Bannock, R.R.
Molecular Sieve Pumping
Vacuum, V. 12, pp. 101-106, March/April 1962

Details of demountable flange seal. In principle a compressible double knife-edge ring is clamped between the two flanges with 10μ copper washers between the knife-edges and the polished flange faces. Five screws are used to clamp the flanges together. The elasticity of the knife-edge ring maintains the vacuum-tight seal over a wide range of temperatures (-196°C to 500°C).

B-011 T
Barbouv, R.J., Isenbarger, R.O.
Fluid Seals
SAE Paper No. 467, March 1-3, 1955, 4 pp., also - Machine Design - V. 27, No. 12, pp. 214-6, 218, Dec. 1955

Conditions which a lip seal must withstand not only affect the design, but also determine the choice of material from which the seal is made. Effects of pressure and speed on lip-type shaft seals, comparison of various synthetic rubbers with leather, and causes of failure are three areas of discussion. Sealing problems in hydraulic applications, and automotive applications are the main topics discussed.

B-012
Bargh, K.A.
Teflon Sheet as a Large-area Gas Seal for Gas Flow Radioactivity Counters
Rev. Sci. Instr., V. 29, pp. 536-7, June 1958

A flexible sealing material to confine the flow of the counting gas. A plastic foam sheet covered with a 0.006 inch layer of teflon self-adhesive sheet was found very suitable.

B-013 T
Barnes, G.C., Ryder, E.A.
Progress Report on Rubbing Seals
S.A.E. Paper No. 46, 11 pp., Jan. 1957

A seal test rig, using easily procured test pieces has been devised to provide a fast rough screening test for materials that might be used for rubbing seals in aircraft gas turbines. The test rig is designed to determine dry load carrying capacity of pairs of material and to approximate wear rates and temperature limitations of compatible mating surfaces.

B-014 T
Barrett, T., English, D., and others
Improvements in or Relating to Shaft Seals
Great Britain Patent 827,015, published Jan. 27, 1960

Viscosity groove type shaft seal to provide against the leakage of pressurized gas by means of a liquid such as oil. The design is claimed to be applicable to plate type and cylindrical type seal designs.

B-015
Barton, D.M.
Vacuum O-Ring Seals
Vacuum, V. 3, No. 1, pp. 51-53, Jan. 1953

A new groove design for use with O-ring seals. Groove has trapezium cross section, with parallel edges perpendicular to the sealing face to provide suitable retention of the O-ring while position and inclination of the groove base controls both, required compression and cross sectional area. Drawings and design data included.

B-016 T
Barwell, F.T.
Research on Friction and Wear
Instn. Engrs. and Shipbuilders in Scotland, V. 95, part 2, pp. 64-91, 1951-52. Also Engineering V. 172, pp. 649-651 - Nov. 23, 1951, pp. 697-699 - Nov. 30, 1951

Experiments carried out in Thorntonhall Laboratories reveal that eccentricities have no noticeable effect on formation of vortices. Therefore G.I. Taylors' theory may be applied directly to an eccentric bearing.

B-017
Bashta, T.M.
Design of Seals for Rotary Type Hydraulic Machine Sets

(Voprosy Konsruirovaniya Uplotneii Gidravlicheskikh) Agregatov-Vrashchatelnikh Tipov, Aug. 60-13 pp. RTS 1145 LC or S.L.A.-translation 60-23793

B-018 B
Baskey, R.H.
A Research Programme on the Investigation of Seal Materials for High Temperature Applications
W.A.D.C. Tech. Report 58-181-74, pp. 12, ref., June 1958, (P.B. 151451) (Preprint 591C-8, A.S.L.E., 5 No. Wabash Ave., Chicago 2, Ill.)

Novel rotating seal materials were developed by powder metallurgy techniques for potential aircraft applications at high speeds and high temperatures. A systematic wear study without lubrication included several commercially available materials, also pure refractory hard metals, binary alloys.

B-019
Bass, H.G., Fischer, W.W. (G.E. Co., Sch'dy., N.Y.)
Estimation of Transient Temp. of Nose Door Seal H.A.-3B
PB 145-917 - Off. Tech. Ser., 3 pp., Oct. 6, 1952
(Hermes Heat Transfer Memo. No. 53, declassified)

The object was to estimate the during-flight temperature of the rubber seal of the nose doors for the H.A.3B missile. No attempt is made herein to determine the overall satisfactoriness of the design investigated.

B-020 T
Baudry, R.A.
Gland Seal for Rotating Shafts
U.S. Patent, 2,246,912, June 24, 1941

Shaft seal, for use on a hydrogen cooled machine, is accomplished by injecting sealing oil around solid sealing rings which fit close to the shaft. The oil supply and gland operation prevent low pressure regions from occurring in the seal clearance space, and air is not sucked into the seal.

B-021 T
Baudry, R.A., Winer, B.B.
Gland-Seal Bearings for Gas-Cooled Equipment
U.S. Patent 2,501,304, March 21, 1950

A combination bearing and seal for use on gas-cooled machinery. Lubricant is injected in an annular groove behind a self-centering sealing ring. Most of the lubricant flows across the bearing to the air side, while a much smaller quantity is diverted by the sealing ring to the gas side.

B-022 T
Baudry, R.A.
Seal for Hydrogen Cooled Generators
U.S. Patent 2,636,754, April 28, 1953

A system designed to substantially reduce the amount of make-up hydrogen required in the hydrogen cooled machines. A small amount of hydrogen is bled off the hydrogen side of the gland seal and is used to remove air and moisture from the air-side oil before the oil is recirculated through the seal. This reduces the amount of air entering the machine, making the hydrogen make-up requirements less.

B-023 T
Baudry, R.A., Winer, B.B.
Journal and Thrust Bearing Practice on Large Rotating Electric Machines
Lubrication Engng., V. 10, pp. 327-335, Dec. 1954

Bearings used on electrical machines must have very effective oil seals to prevent oil mist from passing into the machines. Slow speed machines can use simple felt seals, but high speed machines require a more elaborate sealing system. The author describes a double labyrinth oil seal which uses a split flow of air to prevent leakage of oil mist into the machine.

B-024
Baudry, R.A. and Curtis, L.P.
Gland Seal Systems for Modern Hydrogen Cooled Turbine Generators
AIEE Trans., Power App. & Sys., V. 76, pp. 328-337, June 1957

With the advent of hydrogen pressures above 30 psig made possible by inner cooling of the coils, a re-evaluation of the gland seal design was indicated. This paper covers such a development.

B-025
Baudry, R.A., Peterson, G.E., and others
Oil Seals to Provide Positive Lubrication on Large or High Speed Thrust Bearings
ASME Trans., V. 80, pp. 819-825, May 1958

Paper discusses the application of positive oil seals which effectively prevent aeration of the oil and oil leakage.
Paper concludes that proper application of laminar fluid seals at the bore and periphery of the thrust-bearing runner, and the maintenance of positive pressures at these seals effectively prevent aeration of the oil in large and high-speed thrustbearings, thus assuring adequate lubrication. Discussion follows paper.

B-026
Baumann, R.
Pressure Tests with Vulcanized Fiber, Ebonite and Metal for Stuffing-box Packing. (In German)
Zeit des Ver deutsches Ing., June 1913

Description of tests and their results.

B-027
Baumbach, W.
Stuffing Boxes for Agitators
Brit. Chem. Eng., V. 3, pp. 89-90, 1958

A description of three new types of stuffing boxes used for shaft sealing in chemical vessels fitted with agitators
(1) Stuffing box with plain bearing attached
(2) Stuffing box with outrigger bearing
(3) Extended stuffing box.

B-028
Beacham, T.E.
Rotary and Oscillating Seals
IME Proc., V. 160, pp. 532-536, 1949

Several types of seals, e.g., soft packings, rubber seals, metal-to-metal, metal-to-carbon, journal seals, face seals, etc. are discussed.

B-029
Beacham, T.E., and Towler, F.H.
Hydraulic Seals
IME Proc., V. 160, pp. 532-569, 1949

Presents a detailed account of rotary, oscillating and reciprocating seals; materials normally used in each, and operating conditions.
Purpose of paper an initiation of a comprehensive discussion of existing knowledge on subject.

B-030
Beacham, T.E.
Rotary and Oscillating Seals
The Engineer, V. 187, pp. 228-229, Feb. 25, 1949

Automatically adjustable shaft seals. This includes cross-sectional views of ten metal and carbon face seals. For use with high-speed shafts, they are suitable for particularly difficult sealing problems; diagrams.

B-031 B
Beakbane, H.R.
Leather Seals for Hydraulic and Pneumatics
Proc. 2nd European Jl. Pwr. Conf., pp. 14-22, April 1960

Use of leather may be considered a little old fashioned, but for many purposes it is superior to other materials. The paper considers the properties of leather, with regard to resistance to heat and power fluids, its qualities when impregnated with various waxes and resins, and the effect of corrosion.

B-032 B
Beatty, J.R., Juve, A.E.
Stress Relaxation of Some Rubber and Synthetic Rubber Vulcanizates in Compression
India Rubber World, V. 121, 5. pp. 537-43, Feb. 1950

B-033
Beck, E.C., and others
Designing Seal Rings for Today's Car Transmissions
SAE Journal 70, 71-2, Feb. 1962 (Based on SAE paper 475E)

Physical properties and stress limitations of the material used control design of a sealing ring to a great degree.
Discussion: Cast Iron - mean stress 30,000 psi, max. 55000 psi; Aluminum - mean stress 25,000 psi, max. 40000 psi
Operational problems, including an eleven point check list, and wear compatibility of carrier, bore, and ring.

B-034
Becker, E.
Flow Processes in Annular Gaps (Labyrinth Seals)
Translated for Oak Ridge National Lab., Tenn. from VDI Zeitchrift, V. 51, pp. 1133-41, July 20, 1907. (AEC-TR-4960)

Turbulent flow processes in annular gaps having irregular cross sections are studied with particular references to labyrinth seal applications.

B-035
Becker, Reinhard
O-Ring Seals in Hydraulics
Microtecnic, V. 12, pp. 289-290, Oct. 1958

A reliable hydraulic construction depends on long lasting sealing material. This in the face of high stressing by variable pressure, high or low temperatures and the most varied kinds of movements.
O-rings are recommended for the above mentioned requirements.
Applications up to 350 Kg/cm^2 are possible as well as for a high vacuum.
For all moving part seals, material of 80-90 scleroscope units of hardness is used. If little or no lubrication is present graphite O-rings are used.

B-036
Beecroft, R.I. and Swenson, C.A.
Behavior of Teflon Under High Pressures
J. Appl. Phys., V. 30, pp. 1793-8, 1959

Compression measurements have been made on samples of Teflon at various temps. 75-380°K, and at pressures up to 21,000 atm.

B-037
Bell, H.F.
Automatic Detection of Pump-seal Failure
Oil and Gas Journal, V. 54, pp. 169-170, Oct. 10, 1955

B-038 T
Bell, K.J., Bergelin, O.P.
Flow Through Annular Orifices
Trans., A.S.M.E., V. 79, pp. 593-599, 1957

Summary of the results of an experimental and analytical investigation of orifice coefficients. The theoretical results, derived in a thesis by Bell, are restated for eleven cases, including laminar, transition, and turbulent flow for various types of annular orifices. Experimental results are presented in graphical form.

B-039
Bell, M.
Improving Packing Performance
Power Plant Engineering, V. 40, pp. 726, Dec. 1936

Application of a split ring, around a valve rod, under the follower, to allow greater compression on the valve packing.

B-040
Bell, M.
Solution of all Packing Problems
Power Plant Engineering, V. 40, No. 121, pp. 710-11, Dec. 1936

Explains the technique and use of three items for packing problem solution. Joint paste, made principally of manganese; a sheet of fine gauge wire; a can of shredded metallic gland packing.

B-041
Benjamin, C.H.
Friction of Steam Packings
Mech. Eng., V. 4, pp. 838-839, Dec. 9, 1899

The experiments described in this paper were made at the Case School by senior students under direction of the writer.
Test equipment, data, and results presented.
Soft rubber, graphite, dry and lubricated were used with various pressures.
The loss of power varies almost directly with the steam pressure in harder varieties of packing, but is nearly constant with softer kinds.

B-042
Bentley, F.W.
Pneumatic Cylinder-Packing Leathers
Power, Oct. 7, 1913

Illustrates directions for replacing pneumatic cylinder packing, showing defects resulting from some methods employed.

B-043
Berens, A.S.
New Elastomer - Impregnated Leather Gives Complete Porosity Control in Sealing Applications.
Materials and Methods, V. 38, p. 127, Oct. 1953

A continuing problem in leather seals has been control of porosity. Performance tests on a new elastomer impregnated leather, called Compor, indicate porosity control has been accomplished. Properties of this new product are discussed.

B-044
Berg, R.J.
A Review of Seal Materials for Guided Missile Applications
California Inst. of Tech., Jet Propulsion Lab., JPL-PR-20-340, Contract DA-04-495-Ord-18, March 7, 1958

The currently available seal materials which have proved useful in guided-missile applications are described. A summary is presented in tabular form, of the results of experience with various seal materials.

B-045 B
Bergholm, A.O., and Swartz, P.W.
Sealing Action of Packing Rings
J. Franklin Instn., No. 253, pp. 253-6, March 1952

Determination of the manner in which pressure is transmitted through V-ring packings. Photographs of stress patterns.

B-046
Berk, S.
Fungicidal Treatments for Cork Gaskets
Ind. & Engng. Chem., V. 41, pp. 627-633, March 1949

Seven fungicidal formulations applied to protein and resin bonded cork were evaluated on the basis of a number of criteria for moldproofing automotive gaskets.

B-047
Bertolet, E.C., Jr.
Use of Teflon Packings and Seals in Hydraulic Operations
National Conference of Industrial Hydraulics, V. X, pp. 56-68, October 18, 19, 1956

Presents physical properties of Teflon; Hydraulic application as gaskets, packings; "Vee" and "U" types "L" cup, flange packings, and piston rings. Describes their performance. Test results are discussed.

B-048
Bethel, C.
Waste-packed Bearing Design and Operation
Elec. Jl., V. 21, pp. 115-8, March 1924

Information regarding oil flow and action of waste packing.

B-049
Beyer, K.
Joint Packings for Pipes
Arch. f. Warmewirtsch, '16, pp. 123-126, May 1935

The choice of suitable materials for insertion as packings in the flange joints of pipes is of importance in preventing operational troubles and leakage losses. Various types of packings are described, and an analysis is made of the materials available, and the pressure and temperature limits within which they can be employed for pipes carrying water, steam, air, and oil.

B-050
Bialkowski, L.S. and Le Blanc, J.
Metallic Seal Design for Advanced Systems
SAE-Paper 523B for meeting April 3-6, 1962

All-metallic seal produced by B.F. Goodrich Co. is adapted for rod or piston service, dynamic or static applications, and operates successfully at -65 to 500°F, pressure to 35,000 psi, concept is that of blunt knife edge that forms radial or axial edge of flexible diaphragm.

B-051
Billet, A.B.
High Temperature Sealing Studies of Missile Hydraulic Components
SAE paper 50T from meeting of Mar. 31, 1959, 10 pp.

Environmental requirements for static and dynamic seals used in guided missiles and manned aircraft for outer space; shaft and static seal development; use of silicone and Viton A appears to give best sealing characteristics in 400 to 550 F range to date; seals for hot gas servo systems; examples of applications, such as actuator, servo valve, dual pump, and miniature vane pump.

B-052
Billet, A.B.
High Temperature Hydraulic Pump Seals
Machine Design, V. 31, p. 152+, Oct. 1, 1959

Describes one of the major areas of development in high-temperature hydraulic sealing studies, that of the dynamic seal of the unit drive shaft. (Viton A - lip type radial shaft seal. 400°F/100 hrs. Glass - Teflon filler, carbon-rubber surface with Viton A static O-ring) Static seal development, fluid compatibility, seals for hot servo systems, also discussed.

B-053 T
Billet, A.B.
Hydraulic Seals in Long Term Space Exposures
International Conference on Fluid Sealing-paper G2, April 1961, British Hydromechanics Research Association, Harlow, Essex, Eng.

This paper discusses the effects of long time exposure of combined space environment on hydraulic sealed systems and component. Test results of dynamic seal elements operated successfully at 1000 psi, 2000 ft/min. surface speed for 20 hours. High temperature sealing ranges from 400°F to 1000°F.
Static seals - 300°F to 500°F Hot gas sealing 2200 psi at 2000°F
Gamma dose - 4.5×10^5 R, natural radiation 5.5×10^{14} N TV for 500 hours, etc.

B-054
Billett, E.A., Bishop, J.
A Greaseless Vacuum Seal for Rotating Shafts
Journal Scientific Instrum. V. 35, p. 70, Feb. 1958

This note describes a variation on the Wilson seal, in which the grease is dispensed with by using P.T.F.E. washers on a steel shaft, taking advantage of the very low friction between these materials. The model shown uses four washers of P.T.F.E., with highly polished steel spacers separating them, both washers and spacers being 0.062" thick. Seals have been used at temperatures up to 200°C and have shown no sign of leaks greater than 10^{-5} l/sec.

B-055
Bingham, A.E.
Hydraulic Seals for Extreme Temperatures
Shell Aviation News No. 286, pp. 14-21

Factors to consider in design of high temperature seals used in aircraft and similar applications are outlined; tabulation of fluids and elastomers and their properties; comparison between labyrinth and elastomeric glands.

B-056
Bingham, A.E.
Hydraulic Seals for Extremes of Working Temperatures, with Special Reference to Aircraft
Proceedings, I.M.E., V. 176, No. 17, pp. 409-420, 1962

Report on goals, test conducted to date, and test equipment used. Tables of fluids and their properties, elastomers and their properties, gland configuration, and labyrinth glands.
(Goals include 70° to 170°C, -26° to -40°C, 4000 lbs/in.²)

B-057
Birch, F., Robertson, C., Clark, S.P., Jr.
Apparatus for Pressures of 27,000 Bars and Temperatures of 1400°C
Industrial and Engineering Chemistry, V. 49, No. 12, 1965, 1966, Dec. 1957

Temperature, pressure and other variables may be controlled and measured in the reaction zone with unusually high precision.
Neoprene washer packing and Omega steel washer backing.

B-058 T
Blair, R.W.
Controlled Gap Seal
S.A.E. Paper - 678A - 3 pp., 7 fig., Jan. 1956
S.A.E. Journal - pp. 44, Nov. 1956 (abstract)

The controlled gap seal is a form of labyrinth for sealing rotating shafts and consists of a carbon sealing ring capable of sliding radially within a housing. Experiments with such a seal are described in this report.

B-059
Blair, R.W. and others
Metal Bellows Seals
Lubrication Engng., V. 17, pp. 470-475, Oct. 1961

Study of parameters of metal bellows as used in end face seals. Gives advantages and disadvantages of two basic types of metal bellows; the hydraulically formed type and the welded type.
These seals are suited for the temperatures, high surface speeds and pressures found in the cryogenic liquid pump seal and the hot turbine gas seal in liquid fuel missiles.

B-060
Blech, K.
New Balanced Expansion Stuffing Boxes (In German)
Archiv fuer Waermewirtschaft, V. 42, p. 73, March 1938

Brief illustrated description of German patented "reaction compensator"; only two stuffing boxes are required; new type of metal packing for stuffing box also described.

B-061
Bleyle, G.A., Jr., Crosby, H.W., Kendall, R.E.
Liquid Nitrogen Pump and Vaporizer
Industrial and Engineering Chemistry, V. 49, pp. 1955-1958, Dec. 1957

To satisfy a need for dry, oil free gaseous N at pressures up to 10,000 psi, a reciprocating piston pump using unlubricated packing at -320°F was selected. Packing consisting of graphitized shredded Teflon interspaced with Teflon washers has been developed. Photographs and diagrams.

B-062
Blick
How Flanges Work and Why They Leak
Petroleum Refinery, 1953

B-063 T
Blok, H.
Discussion on Gland Seals
Int. Conf. on Lubrication and Wear., pp. 775-7, Oct. 1-3, 1957, Proc. Instn. Mech. Engnr., 1 Birdcage Walk, London, SW 1

Commenting on a paper by Jagger. An attempt is made to explain two possible mechanisms of film formation in lip seals. First, an eccentric radial pressure distribution set up by the ever present slight imperfections of machinery is the source. Based on an assumed initial radial pressure distribution and hydrodynamic theory, the author shows that there is a minimum film thickness for any corresponding film profile. Secondly, inertia effects, might create a wedging action.

B-064 B
Bloom, J.C.
The Design and Development of Rubber and Teflon High Pressure Sealing Devices for Aircraft Hydraulic and Pneumatic Systems Components
Douglas Aircraft, (Proc. 15, NCIH., V. 13, pp. 194-205, Oct. 1959)

Paper describing the development of AN 6290 O-ring for use on AND 10050 Boss and related fittings.

B-065
Bloomquist, C.R.
Gasket Seals
Final Report F-A 1858, Franklin Inst. Lab. Res. & Dev., Sept. 1956

B-066
Boeddinghaus, H.
Felt-sealed for Service
Elect. Mfg., V. 27, pp. 72, 74, 76, 94, June 1941

Wool felts serve as a good material for sealing bearings and shafts against seepage of lubricants and intrusion of vagrant matter.

B-067
Boller, L.C. and others
High Temperature Resistant Sealant Materials
USAF WADC TR 56-155, (AD110633), Dec. 1956, 59 pp.

Development of a formula for a sealant compound which is unaffected by JP-5 jet fuel for a limited period of time when fuel vapor temperature is 540°F and liquid fuel temperature is 380°F; retains flexibility and adhesion; and withstands proof testing under flexing and pressure at the liquid and vapor fuel temperatures required.

B-068
Bollinger, P.R.
Power Feed Seals
Electromechanical Design

For high pressure high temperature helium.

B-069
Boltz, R.W.
Bellows Sealing Device Protects Bearings from Abrasives
Product Eng., V. 15, p. 614, Sept. 1944

Spring actuated brass bellows which maintains continuous seal contact has been designed for planetary type miller to exclude abrasives, dust, water, and corrosion when bearings are loose, misaligned or eccentric.

B-070
Boltz, R.W.
Gaskets in Design
Mach. Design, V. 17, pp. 151-156, March 1945

Article discusses the general steps that may be followed as an aid in the selection of compression gaskets, regardless of nature of equipment in which they are to be used or the details of the closure design.

B-071
Bondroit, F.
New Developments in Packings Resistant to Chemicals (In German)
Chemie-Ingenieur-Technik, V. 22, pp. 236-8, June 14, 1950

Properties and applications, with special reference to metallic packing and combined asbestos-metal packings; valve rings; seals for shafting; illustrations.

B-072
E Boon, E.F.
Seals for Rotating Shafts (In Dutch)
De Ingenieur, V. 62, pp. 43-50, July 21, 1950

The development of the mechanical seal shows clearly that a tight seal is possible. Various factors leading to a tight seal with a normal stuffing box are described. A very precise assembly of a packing is necessary to obtain a tight seal. This is possible with special tools, drawings of which are shown. Special construction to cope with excessive radial movement of the shaft are described.

B-073 T
Boon, E.F.
A Few Remarks on the Sealing of Rotating and Reciprocating Machine Parts (In Dutch)
De Ingenieur, V. 64, No. 12, pp. 33-40, 1952, (BHRA Translation No. T459, Feb. 1953; AEC Tr 5109)

Various formulas for leakage through narrow gaps are given and applied to two sealing configurations, the piston and sealing ring, and mechanical face seal. Laminar flow is assumed. An equation is presented for the leakage in a mechanical seal with a varying height gap. An effective gap, height is presented for the case where any variation is small and continuous. It is shown that contact between face and seal cannot be avoided.

B-074 T
Boon, E.F., Honingh, S.
Some Notes on Seals for Rotating Shafts
Proc. Fourth World Petroleum Congress, Section VII/A Paper 1, Rome 1955, 18 pp.

A survey of several types of dynamic shaft seals indicating general requirements and their practical deviations. Discusses the advantages and disadvantages of the packing box seal, narrow faced mechanical seals, visco seals, and dual diameter visco seals. Elementary equations are developed and experimental results are shown.

B-075 T
Boon, E.F.
Some Notes on Shaft Seals for Chemical Pumps (In Dutch)
De Ingenieur, V. 67, pp. 31-38, 1955

Tests of power consumption and leakage of soft and metal packings are described. The heat transmission through the shaft and packing is calculated. Some construction of pumps without a packing box are discussed.

B-076 B
Boon, E.F., Kriggsman, A.
Some Principles of Seals (In Dutch)
Chemisch Weekblad, V. 52, 27 pp. 526-32, July 7, 1956

This article gives some of the more elementary explanations of sealing mechanisms. The author discusses some more recent work on a flange type joint sealed with a packing, and gives useful practical details.

B-077 T
Boon, E.F., and Tal, S.
Hydromechanic Sealing of Rotating Shafts
Chem. Ing.-Tech. V. 31, 3 pp. 202-12, 1959 (In German) (Main Library Translation - T.F. 5825 - GE Co., Schenectady, N.Y.)

The pressure when generated by a screw pump is given by $p = \mu \omega d \ell / S^2 X$ when p is the pressure, μ is the fluid viscosity, ω is the shaft angular velocity, d is the shaft diameter, ℓ is the length of the threaded portion of the shaft, S is the radial gap between the shaft and the casing, and X is a coefficient which is a function of thread geometry. The author develops the exact values of X for a variety of thread shapes, and examines diffusion when the device is used as a seal.

B-078
Boreham, K.R.
Hydraulic Oil Seals
Scientific Lubrication, V. 12, pp. 47-52, 46, May 1960

Static and dynamic seals for oil-powered hydraulic equipment. Seals are mainly synthetic rubber together with some thermoplastic materials such as Fluon (PTFE). These materials can be combined with various fabrics. The description and operation of the various seal forms are presented.

B-079
Bos, J.R., Weber, E.B.
Composite Elastomer-Metal O-Ring Seals
WADC Tech. Report 59-749, 22 pp., 14 fig. - 1959

Performance evaluation of the prototype composite seals in hydraulic fluids at elevated temperatures revealed that rolling or spiralling, a failure frequently experienced with conventional O-rings, was virtually eliminated. The spring cored O-ring performed much better in static impulse tests than in dynamic reciprocating tests. Data is presented and results discussed.

B-080
Boudreau, W.F., Taylor, E.R.
Shaft Seals for the EGCR Reactor
Nuc. Sci. Abst. #28385, (V. 16, No. 20, Oct. 31, p. 3713), (TID7631 (Pis-25))

A design description of the E.G.C.R. shaft seals is presented. Specifications for H_e leakage from the EGCR cooling system limit the loss to 1 percent per day. The discussion is devoted to a description of the floating-bushing seal, buffering and H_e removal system, gas transfer paths, back-diffusion, predicted operating parameters, and seal configuration.

B-081 T
Boudreau, W.F., Taylor, E.R.
Appraisal of the Applicability of Liquid-buffered Floating Bushing Shaft Seals to the Circulators for a Gas Cooled Reactor
Int. Conf. on Fluid Sealing Paper C6, April 17, 19, 1961, (12 pp.), British Hydromechanics Research Association, Harlow, Essex, England

The requirements which must be met by the shaft seal for a gas cooled reactor are presented, and the basic configuration of the seal is discussed. The relationship between the seal and circulator design is reviewed briefly, and an appraisal is presented on the selection of the buffer fluid and on the requirements for the gas purification system. Some anticipated design, requirements for shaft seals for future reactors are discussed.

B-082
Boulloud, J.P., and Schweitzer, J., (In English & French)
Experimental Study of Metal Gasketed Joints for Ultra-high Vacuum
Vide, V. 14, No. 82, pp. 241-249, July-Aug. 1959

B-083 B
Boussand, T.
The Problem of Seals
Tech. Sci. Aeronaut., No. 1, pp. 35-9, 1947 (In French)

The article describes various types of joints such as metallic, plastic and resilient joints. It compares the value of different materials such as natural rubber Buna S, and GRS., Neoprene, Perbunan, "Thioplastes," mixtures of butyl rubber, polyvinyl alcohol, and plastic materials. A table is given showing the swelling of the various types of rubber.

B-084 T
Bowden, A.T., Martin, G.H.
Design of Important Plant Items
Int. British Nuclear Energy Conference V. 2, pp. 156-67, April 1957

A discussion of the gas seal in the centrifugal gas circulators in the Calder Hall Atomic Reactor. The seal is comprised of a labyrinth, a floating bushing, and a "stand still" seal which is an elastomeric ring forced into contact with the rear of the centrifugal impeller when the compressor is stopped.

B-085
Bower, E.S., and Vandermar, B.C.
Chemical and Mechanical Development of Elastomeric Piston Seals for Automatic Transmissions
SAE Paper No. 1A, 8 pp., Jan 13-17, 1958

Characteristics of butadiene-acrylonitrile copolymers; requirements target specification for screening compounds, rubber compounding of right butadiene-acrylonitrile ratio; fabrication and testing of seals; compounds based on fluorocarbon rubber.

B-086
Bower, E.S., Vandermar, B.C.
Chemical and Mechanical Development of Elastomeric Piston Seals for Automatic Transmissions
Rubber World, V. 138, p. 115, April 1958, SAE Journal, V. 66, p. 130, April 1958

B-087
Bower, E.S., Vandermar, B.C.
Seal Changes Solve Special Problems
SAE Journal, V. 70, 48-9, June 1962

Modifications to the standard lip seal design solved these problems for one manufacturer.
(1) The seal section was deepened to keep it in place.
(2) The back of the seal was chamfered to get it away from the area of extrusion.
(3) The stretch fit was tightened up to keep the seal from lifting off the bottom of the groove.
(4) The compound used for seal material was adjusted for extrusion resistance. Drawings included.

B-088
Bower, E.S., Vandermar, B.C. (Acadia Div., Western Felt Works)
Reciprocating and Static Seal Designs of Elastomeric Compounds
S.A.E. Paper 475G, September 1962

B-089
Bowman, H.A. and Cross, J.L., et al.
Impregnated Teflon as a Packing Material at 150,000 PSI
Rev. Sci. Instr. V. 27, p. 550, 1956

Teflon impregnated with 5 wt. percent MoS_2 proved to be satisfactory packing material in the high-pressure moving seal of differential area pressure intensifiers.

B-090
Box, J.R., Weber, E.B.
Spring Cored O-Ring Investigation - Quarterly Progress Report No. 1
Report No. ESR - 12212, Chance Vought Aircraft, Dallas, Tex., 24 pp., April 8, 1959

Reports evaluation of a spring-cored O-ring with Viton A elastomer compound. Pressure impulse test in the CVL-2249 cylinder assembly. System pressure 400 psi and tests would be conducted at 400°F. Impulse pressure cycles -40 per min. Table of Survey of Elastomeric materials for use with spring cored O-rings.

B-091 T
Boyd, J., Robertson, B.P.
Oil and Temperature Relations in Lightly Loaded Journal Bearings
Trans. Am. Soc. Mech. Engrs., V. 70, pp. 257-262, April 1948, Lubrication Engineering, V. 4, pp. 58-62, April 1948

The oil flow and operating temperature were studied in a series of tests of high speed, lightly loaded full journal bearings. The oil flow through a lightly loaded bearing having a circumferential supply groove would be quite similar to the oil flow in a buffered bushing seal. Good agreement between theory and experiment was reported.

B-092
Boyd, W.W.
Spiral Wound Gaskets on Flanged Joints
Pet. Refiner, V. 30, 120-3, Dec. 1951

B-093
Bradford, B.W., and Skinner, W.J.
Improvements in and Relating to Seals
British Patent 809,281 Feb. 18, 1959

The preparation of graphite seals for rubbing use in UF_6 vapor is described.

B-094
Bradford, B.W., and Skinner, W.J.
Molded Sealing Element
US Patent 2,879,247 March 24, 1959

Molded sealing elements suitable for use under conditions involving exposure to uranium hexafluoride vapor are described.

B-095 T
Bradford, L.J.
Oil Film Pressures in End Lubricated Sleeve Bearing
Technical Bulletin No. 14, School of Engineering, the Pennsylvania State College, 1931 (pp. 90-104)

This report deals with an early experimental study of the film instabilities in low clearance end-lubricated journal bearings. Cavitation as such is not mentioned in the report, but the conditions observed and reported on seem to suggest this type of instability which was more pronounced when the lubricant supply was to only one end of the bearing.

B-096
Bragdon, C.T., and Monich, M.T.
Integral Seal Ball Bearings
Prod. Engng., V. 26, pp. 179-185, Nov. 1955

Analysis of types of seals; relation of combination of bearing and performance; seal materials; effect of temperature and speed; design suggestions.

B-097 B
Brenza, J.J., Fuchslüger, J.H.
Evaluations of Designs and Materials for High-speed, High Temperature Shaft Seals for Turbojet Engine Application
Part II-WADC Tech. Report 56-267, 23 pp., 14 fig., 11 tab, Dec. 1958

Various designs of both circumferential and radial face seals were subjected to test during the program of work described in this report, and their relative merits are discussed. 163 different materials were evaluated, 1000 hours life, 10,000°F, 200 lbs/sq.in. 30,000 ft/min rubbing speed.

B-098
Bresee, J.C., Watson, C.D., Watson, J.S.
Gamma Radiation Damage Studies of Organic Protective Coatings and Gaskets
Oak Ridge Natl. Lab. (U.S.A.E.C.), ORNL-2174, 30 pp., Nov. 1956

Qualitative results of screening studies, quantative results of decontamination test and physical property measurements.

B-099
Bridgman, P.W.
Methods of Handling Very High Pressures
Comp. Air Mag., V. 27, pp. 17-19, Jan. 1922

Explains methods of packing pistons and joints to withstand pressures up to 500,000 psi.

B-100 T
Brillie, H.
Viscous Fluids at High Velocity and Their Effects on Bearing Surfaces, Propeller Blades etc. (In French)
Le Genie Curl, V. 114, pp. 10-13, 33-35, 1939

The author notes the erosive damage which has been reported on the surface of bearings, propeller blades, and hydraulic turbines, and suggests that the only rational explanation of this effect is cavitation.

B-101
Brkich, A.
Mechanical Seals, Theory and Criteria for Their Design
Product Engng., V. 21, pp. 85-89, April 1950

Effects of surface tension, liquid back pressure, viscosity, spring force and face proportions on leakage through mechanical joints.
Qualitative basic equations are derived for leakage through gaps and then analyzed to explain the performance of seals with volatile and non-volatile liquids.

B-102 T
Brkich, A.
Sealing Device for Relay Pump Shafts
US Patent 2,710,205 June 7, 1955

The use of a flow restricting bushing between two annular chambers sealed at the outer ends by identical mechanical face seals. Buffer fluid at a pressure greater than that of the substance to be sealed is used. By suitably restricting respective flow rates, the pressure drop across the two mechanical seals are each substantially less than the total pump pressure.

B-103 B
Brkich, A., Allen, R.E.
Development of Floating Ring Type Stuffing Boxes for Eddystone Boiler Feed Pumps
A.S.M.E. Paper 59-A-259, 5 pp., 9 figs., Nov-Dec. 1959

Recent increases in the size, pressures and operating temperatures of boiler feed pumps have required new designs of stuffing boxes, to be undertaken and the authors report on their successful work in this field. 1750 fpm shaft, 4.5 in dia., sealing 650 lb/sq. in. 435°F.

B-104
Britt, J.R.
Elastomer Compounds for High Vacuum Application
U.S. Naval Research Laboratory, 12 photos., diagr., July 1951, PB 104729, Off. Tech. Ser.

A compounding study using butyl, neoprene, and butadine-acrylonitrile type elastomers as high vacuum seal is presented. Vacuum ranges 10^{-5} to 10^{-6} MM for the rapid pumping systems used. Test data and results.

B-105
Broadbent, E.R.
Performance of Linear LTZ-70 Hydraulic O-Ring Packing used with Chrome Retanned Leather Back-up Rings at 3000 PSI Pressure.
Douglas Aircraft Co. Report, SM-11809, April 4, 1946

B-106
Broadway, N.J., and Palenchak, S.
The Effect of Nuclear Radiation on Seals, Gaskets, and Sealants
Battelle Memorial Inst., Radiation effects Information Center, Columbus, Ohio., NR-7096, Project No. 2133, November 30, 1958, Contract AF33 (616)-5171

Various sealing materials were irradiated at temperatures up to 450°F and evaluated. A few of the materials tested - nitrile, neoprene, Teflon, Viton A and thiokol rubbers.

B-107
Broce, J.R.
How to Inspect and Maintain Mechanical Seals in H.T.W. Pumps
Power, V. 105, 190-3, Dec. 1961

Mechanical seals is equipment of a high order built to very close tolerances, and for proper operation it requires careful installation. In hot water applications, temperature control is a must. Detailed description of installation, operation, and maintenance. Sixteen steps with pictures for each.

B-108
Brodsky, J.
Consider the Boiler Gasket
Power Plant Engineering, V. 40, No. 12, pp. 712-13, Dec. 1936

General commentary on surface preparation, gasket selection and application as maintenance procedure.

B-109
Brothman, A.
Stuffing Boxes; Pressure Sealing Devices for Rotary Shafts
Product Engng., V. 11, pp. 417-419, Sept. 1940, pp. 520-522, Nov. 1940

Types of stuffing boxes are illustrated. A table lists materials utilized for packings and seals v.g., asbestos hemp, rubber, babbitt, white metal, carbon rings etc. Norms for selection and maintenance of stuffing boxes are discussed.

B-110
Brown, E.D., Jr.
Investigation of Carbon and Carbon Substitute Seal Materials
R58 Ch SD-117, GEL Tech. Data Center, G.E. Co., Sch'dy., N.Y., 4 pp., Dec. 29, 1958

Lists criteria of seal materials as materials which must be able to operate dry or nearly dry; partly lubricated; or lubricated, all without damage to seal material or contacting shaft. For conventional lubricants in use, notably petroleum fluids and diesters, most satisfactory material is carbon. Carbon additives which are discussed, make it possible for operation at 1300°F and hotter.

B-111
Brown, Wm., Raxdow, A.
Reusable Seals for Electronic Equipment
P.B. 131194 - Off. Tech. Ser. (94 pp., photo, dwg. diag. 8 r) May 1957

This report covers the investigation, evaluation, and establishment of the most effective type of reusable seals for airborne electronic equipment. Fused seals, methods and procedures were developed for hermetically sealed enclosures of electronic assemblies, subassemblies, and equipment, so they can be opened fifteen times for repair and resealed without destroying the contents therein. The methods developed are satisfactory for field service application.
AD 130772 Project 4526, Task 41602 - Summarizes research from June 1, 1955, Oct. 31, 1956 under contract AF33 (616) 294 AF WADC TR 56-616.

B-112 T

Browne, L.H.
Rotary Seal
US Patent 2,230,881, Feb. 4, 1941

A seal through which fluids may be fed to a rotary shaft for distribution to a pressure actuated device. Two opposing face shields are provided with a lubricant to prevent leakage of the pressurized fluid to the atmosphere. A visco seal or labyrinth seal retards the leakage of this lubricant into the pressure passage.

B-113 T

Browne, L.H.
Rotary Seal
US Patent 2,270,927, Jan. 27, 1942

The device allows a fluid to be passed through a stationary collar with an inner annulus through radial holes to the center of a rotating shaft. The seal is a double mechanical face seal. The annular space between the seals forms a collecting volume for the fluid in transit.

B-114 T

Browne, L.H.
Shaft Seal
US Patent 2,270,928, Jan. 27, 1942

A method sealing a joint between two tubes or hollow shafts which must have relative motion while transporting a fluid medium. In one design, fluid pressure forces a sealing ring jointly against the inner and outer race of a ball bearing. Another design used fluid pressure to force a sealing ring against the perpendicular end of one member with which the ring is in relative motion.

B-115

Browning, W.E., and Zizzo, S.G.
Life Expectancy of Organic Gasket Material at the Transition-precessor Position Within the Mark I Accelerator Shielding
Calif. Res. & Devel. Co., Livermore, Calif., LWS-22505, CRD-A17-41, 6 pp., Jan. 15, 1953

The life expectancy of elastomers vs gaskets between the transition-precessor position are predicted suitable for 10^3 hours to 10^4 hours under the anticipated neutron flux conditions of 2.7×10^7 neutrons/cm^2 sec.

B-116

Brueschke, E.E.
A Leak-free Method of Transmitting Motion into Ultra High Vacuum
Vacuum, V. 11, No. 5/6, pp. 255-259

A description is given of a seal which is leak-free and can be used to transmit rotary and limited translational motion into ultra high vacuum. The seal is accomplished by using a low vapor pressure liquid in combination with a rubber gasket. The seal is useful at 10^{-9} Torr and below and therefore has excellent application for testing of mechanisms, bearings and materials in ultra high vacuum.

B-117

Brueschke, E.E.
Ultra-high Vacuum Seal for Space Simulation Systems
Rev. Sci. Inst., V. 32, pp. 732-4, June 1961

The operation and construction of a leak-free seal, by means of which limited translational and unlimited rotational motion in a vacuum can be obtained, is described. The seal is accomplished by using a rubber gasket in combination with a liquid which is in equilibrium with internal and external pressure. This seal can be operated from zero to several thousand rpm and can be used at 10^{-9} mm Hg and lower without difficulty. The seal has distinct advantages for use in space environment simulation chamber and other ultra-high vacuum apparatus.

B-118

Bruno, J.O.
Investigation of Hydraulic Low Friction Teflon Cap Seals
PB 133979, Off. Tech. Sev., 61 pp., photo, dwg., diags., graphs

A low friction rod seal was required for hydraulic actuators which would increase O-ring life and minimize rod seal leakage. A test actuator, rod seals, and corresponding hydraulic impulses and driving circuits were designed and built. Both static and cycling tests were conducted on various size seals with varying cross sectional O-ring squeeze.
Final Design criteria are shown.
AD 118213, Project 1371, Task 13495. Contract AF33 (601) 28637, AF WADC TR 57-163.

B-119

Bryan, J.D.
Valve Seals Prevent Oil Loss
Diesel Power, pp. 86-87, Sept. 1959

Positive sealing of valve stem-to-guide clearances saves oil, cuts maintenance costs, improves engine performance.

B-120 T

Bryant, J.M.
Sealing Drives and Anti Friction Bearings
Design News, V. 12, No. 71, pp. 156-57, April 1, 1957

The author describes the types of seals available (clearance or labyrinth, lip or axial seals, and face seals) and lists applications of each type.

B-121 T

Bryant, J.M.
Types and Characteristics of Bearing Seals
Machine Design, V. 29, pp. 127-132, April 4, 1957

This abstract of an American Society of Lubrication Engineers Paper compares the sealing characteristics of clearance seals and closure seals as required for sealing rolling-element bearing.

B-122

Bryant, J.M.
Match Drive and Bearing Seals to Ser Conditions
Iron Age, V. 179, pp. 136-138, No 'une 13, 1957

Standard and special seals for mech :ives and anti-friction bearings recommend choice of effective seal.

B-123 T

Bryant, R.W., Dukes, W.A.
Some Sealing Materials for Threaded Joints
International Conference on Fluid Sealing, Paper B3, April 1961, British Hydromechanics Research Association, Harlow, Essex, England

The development of special sealing compositions, both lutings and cement for parallelthreaded joints is summarized and their performance is described.
The most important single factor in the sealing of threaded joints is the quality of the composition.
Desirable qualities are discussed.
Cements which dry merely by evaporation of solvent are greatly inferior to solvent-free cements and lutings.

B-124

Brymner, R., and Steckelmacher, W.
Demountable Vacuum Seal for Operation at Temperatures from - 188 to 800 C
Jour. Scientific Instruments, V. 36, pp. 278-281, June 1959

Static metal seals made from flat gasket sheets of soft copper, iron, stainless steel, aluminum or nickel. These are compressed to a cone shape by metal flanges of mild steel or stainless steel. The seal depends on a surface-friction effect instead of compression. The same seal may be used many times on the same pair of flanges.

-125

Buckley, D.H., and Johnson, R.L.
Halogen Containing Gases as Lubricants for Crystallized-Glass-Ceramic-Metal Combinations at Temperatures to 1500°F
NASA TND-295, 19 pp., Oct. 1960

Pyroceram 9608 (a crystallized glass ceramic) has been considered for use in high temperature bearing and seal applications. One of the problems encountered with Pyroceram is the lack of availability of lubricants for the temperature range in which this material becomes practical. Experiments were conducted with Pyroceram sliding on various nickel- and cobalt-base alloys using reactive halogen-containing gases as lubricants. Friction and wear data were obtained as a function of sliding velocity and temperature.

B-126

Buckshee, W.
New Hydro Gland Packing Really Works
Power Engng., V. 63, p. 70, April 1959

New seal for hydroelectric turbines provides effective solution to problem of shaft cutting by tight packing, or leakage when packing is not sufficiently tight; silicone "Klosure" impervious to water temperature in Northern Canada has given two years successful trouble free operation; Al or bronze adapting rings are used in gland.

B-127

Bunn, J.
Best Methods for Preparing Journal Box Packings
Railway Master Mech., April 1901

Illustrated description of vats for treating packings.

B-128 T

Burger, F.E.
Mechanical Seals for Rotating Shafts
Aircraft Engineering, V. 19, pp. 93-4, March 1947

The author's design of a special shaft sealing ring is described. The seal consists of two or more carbon rings of varying cross-sectional area, having a gap cut at the minimum area section. As each ring is pressed on the shaft, its inside dimensions remain circular. A mathematical treatment is presented to arrive at numbers for frictional torque and horsepower loss. No data presented.

B-129

Burger, F.E.
Mechanical Seals for Rotating Shafts
Engrs. Digest, V. 4, pp. 313-314, July 1947

Early concepts of the labyrinth seal and soft packing seals is reviewed, together with their limitations. Surface finish of shaft and selection of seal materials with low coefficient of friction recognized as critical. Synthetic carbon containing metal oxides and other admixtures selected as affording one of most suitable substances.
A developed seal utilizing this material is described together with mathematical treatment.

B-130

Burns, David L.
Leakage of Water from Pump Mechanical Seals
Nuclear Science Abstract 20564, (DP-666) (Du Pont de Nemours & Co., Savannah River Lab., Contract AT (07-2)-1), 11 pp., Nov. 1961

Measurements of water vapor leakage past mechanical seals for pump shafts showed that commercially available mechanical seals do not leak more than about 30 lb/yr of vapor. Both vapor and liquid leakage were measured in bench-scale tests on mechanical seals from three manufacturers for pump shaft sizes of 2.625- and 3.563 with diameter, shaft speeds from 500 to 3600 rpm, and pressures from 250 to 1000 psig. Leakage of D_2O would be recovered.

B-131

Burns, R.C.
Cleaning and Lubricating of Brake Cylinder Packing Leathers
Railway Review, July 28, 1917

B-132

Burrows, G.
Improvements Relating to Sealing Glands
P.P. Starling, E. Metropolitan-Vickers Electrical Co. Ltd., Br.Pat. 703696

A rotary seal is described which allows relative movement between a stationary and a rotating part of a high vacuum system. The two parts end in flanges between which are two concentric sealing rings of some flexible material; these are fixed to one flange and slide on the other. The interspace between the two rings is connected through the fixed flange to a source of rough vacuum so that the pressure difference across the inner ring is small and the leakage negligible.
A ball-race may be fitted between the flanges to space them apart and the clearance is adjusted by screws to give optimum sealing.

B-133
Butler, C.A., and Sutter, R.S.
Pump Stuffing Box for Obnoxious or Inflammable Liquids
Oil & Gas J., V. 49, p. 117, Aug. 17, 1950

In problem involved, operator could not stay in pump house with even slight drip from packing; liquid was solvent which made grease sealing impossible; gland could not be sealed with water; in solution of problem, principle of liquid-sealed centrifugal shaft packing was combined with action of backvanes on pump impeller; dry air was used to keep pumped liquid from coming in contact with stuffing box; diagram of actual installation.

B-134 B
Butler, C.A., Sutter, R.S.
A Leak-proof Stuffing Box
Chem. Engg., V. 57, 9, p. 117, September 1950

Description, drawing, and application of a leak-proof stuffing box.
Air pressure keeps liquid away from shaft, and liquid annulus keeps air out of pump.

C-001
Cable, J. A.
Selection and Maintenance of Centrifugal Pump Seals
Power Eng., V. 57, p.82-3, Nov. 1953
Some tips on selecting the proper pump seal for the job and giving them proper maintenance when installed.
Subjects discussed: - packed stuffing boxes; difficult liquids vs. boxes; repacking hints; mechanical seals; difficult liquids vs. seals; replacing mechanical seals.
Photographs and drawings.

C-002 T
Cable, J.A., Kristy, O.M.
How to Maintain Stuffing Boxes That Have Mechanical Seals
Power Engineering, V. 62, p. 88, 1958
Practical tips on mechanical seal maintenance.

C-003
Cady, E.L.
Packology
Mill & Factory, V. 36, pp. 134-136, 325 etc. March, 1945.
Evaluates and describes the application of the more common packing materials for pumps, valves, expansion joints, etc.

C-004 B
Cameron, D.H.
Nylon Pipe Joints for Use in Hydraulic Services Up to 3,000 lb/sq. in.
Proc. 14th B.I.S.R.A. Plant Engg. Conf. (Hydraulics in the Steel Industry)
pp. 87-91 disc. pp. 91-3. March 1957

C-005
Campbell, J.A.
Fundamental Characteristics of High Performance Hydraulic Systems
PB 114178, Off. Tech. Serv., (258p; photo, dwgs, diag. gr. June 1950)
Results from literature surveys, theoretical studies, and experimental work include: variations in fluid viscosity and density with temperature and pressure; frictional characteristics of O-Rings at various squeezes, pressures and temperatures; hydraulic damping or resistance factors for tubing, fillings, and buses, etc.

C-006
Candee, R.
Oil Seal Applications in Farm Tractors
SAE Paper n 23, 7 p., Jan. 14-18, 1957
John Deere Waterloo Tractor Works, procedure in approaching operating conditions and securing recommendations of manufacturers of seals; applications on right-hand crankshaft bearing; experience with three basic types on front wheels, namely; packing, lip, and face-type seals.

C-007
Cannizzaro, S.
Republic Tests Seals for 1000°F Hydraulics
Space Aeronautic, V. 35, pp. 103-104, 106, 108, 110, April, 1961
Composite metallic seals for use in hydraulic systems having an ambient temperature of 1200°F and a liquid temperature of 1000°F are developed using Ag impregnated Mo, 430 stainless steel filled with 84Ag - 16 In brazing alloy, Nu Die V and Rex AAA tool steels and Haynes Stellite.

C-008
Cannizzaro, S., Lee, Jr. Schroder, R.
Sealing in Severe Environments
National Conference on Industrial Hydraulics, V. XV, pp. 187-199, October, 1961
This paper discusses design techniques, and results of three programs concerned with sealing in severe environments, recently completed, or currently conducted. 275°F pneumatic system, 600°F hydraulic system. Elastomeric dynamic seals; non-elastomeric dynamic seals; static seal testing, metallic boss seals. Tables of test data and results discussion.

C-009
Cannon, C. H.
1000°F Pneumatic Servo System
SAE Journal, V. 66, pp. 70-71, April, 1958
Design requirements of a new Lockheed aircraft pneumatic system operation at 1000°F and 5000 psi for 500 hours life. Leakage and lack of high temperature materials with required mechanical characteristics are major problems encountered. The system will be used to operate the dive brakes of a supersonic aircraft. (Indicates steps being taken - no data results)

C-010
Cannon, R.
How to Choose and Use Today's Gaskets in Today's Equipment
Petroleum Processing, V. 12, pp. 95-110, March 1957
Gasket construction, functions, material, and designs.

C-011
Carlotta, E.L.
Synthetic Rubber O-Ring Seals
Prod. Engrg., V. 22, pp.130-135, June 1951
For sealing of static or moving parts in hydraulic systems, "O"-rings have become widely accepted.
Large number of synthetic rubbers available increases this acceptance.
Synthetics include butyl, GR-S rubber, polysulphide rubber, silicone rubber, nitrile rubber to mention a few. These rubbers are compatible with hydraulic fluid, aromatic fuels, dilute acids and alkali, Freon gas, etc. Design and installation problems are discussed.

C-012
Carlotta, E. L.
Materials for O-Ring Seals
Materials and Methods, V. 38, pp. 104-107, Dec., 1953
Material requirements for O-rings are exacting.
Natural and synthetic rubbers, silicon elastomers, and plastic elastomers meet many of these prerequisites which are discussed in detail.

C-013
Carlotta, E.L., and Hobein, E.M.
Limitations of Synthetic Rubber Packings For High Temperature Applications
Rubber Age, V. 74, pp. 85-90, 134, Oct. 1953
Heat resisting potential of available elastomers for use as molded seals in various components of hydraulic and pneumatic systems on military and civilian aircraft; discussion covers compounding, design of rubber-containing units for operation at elevated temperatures, and comparative properties.

C-014
Carlson, A.F.
Interim Report of Conical Seals for Flared Tubing
Bulletin No. B707J, Chrysler Corp., Detroit, Michigan

C-015
Carlson, A.F.
Test of Conical Seals for Flared Tubing
Bulletin No. B-767J, Chrysler Corporation,
Detroit, Michigan, March 10, 1961

C-016
Carlson, N.J., Black, W.S.
Joints for High-Pressure High-Temperature Piping
Trans. ASME, V. 73, pp. 237-46, 1953

Outlines briefly the difficult problem of obtaining leakproof joints for high-pressure, high-temperature piping. It includes the results obtained from an extensive test on three full-sized joints for connecting ferritic alloy main steam piping to austenitic-alloy turbine equipment of 150,000 kw capacity. Description of joints. Performance tests-procedures. Pressure seal joint.

C-017
Carlson, R.W.
Friction Characteristics for Hydraulic Seals
Product Engrg., V. 27, p. 221, May, 1956

Four-lipped seal developed to replace standard O-ring and gaskets under listed conditions. Comparison data shown.

C-018
Carter, R., Karassik, I.J.
Designing, Constructing and Maintaining Centrifugal Pumps; Mechanical Seals
Water and Sewerage Works, V. 103, pp. 27-30, Jan., 1956

The basic principles of mechanical seals are presented, and a few of the various types of construction are described, along with their limitations.

C-019
Cass, E. and McCuistion, T.J. and others
Applied Hydraulics & Pneumatics, V. 13, pp. 132-133, Oct., 1959

Description of test procedure, and result of investigation to determine amount of force required to compress O-rings; log-log plot of compression force vs deflection enables rapid calculation.

C-020
Chandler, W.L.
Bellows in the Refrigeration Industry
Refrig. Engrg., V. 42, pp. 297-299, Nov., 1941
pp. 381-382, etc., Dec., 1941

The principal bellows applications are bellows-type seals for rotary shafts, motor elements for expansion valves, thermostat elements for expansion valves and switches and bellows-type packless valves for shut-off and other service. These are described in part 1.
Part 2 describes the following properties of bellows:
1. Spring rate or flexibility 2. Pressure resistance
3. Mean effective area 4. Fatigue.

C-021
Chapman, C.S.
Sealing Problems in Current Passenger Car Automatic Transmission
S.A.E. Paper 475D - September, 1962

C-022
Charlton, G.J.
Proper Method of Packing and Lubricating Journals
Railway Master Mechanic, June, 1914

Outlines the quality of waste and oil and describes the proper method of packing.

C-023
Cheney, L.E.
O-Ring Friction Characteristics
Mach-Design - V. 21, pp. 245-6, April, 1949

The recently developed O-ring shows many advantages over V-ring and U-cup packings. The effect of pressure, 500, 1000, 1500 psi were checked. It was found that an increase in pressure resulted in a small increase in running friction. Rod finish was checked 2.8-3.2, 6-8, 10-12, 16-18 micro inches rms, and results evaluated.

C-024
Cheney, L.E. and McCuistion, T.J.
Aircraft Hydraulic Packings; Factors Influencing Their Performance
Mech. Engrg., V. 70, pp. 675-679, Aug., 1948

The several factors influencing the performance of hydraulic packings for aircraft are discussed; both of a design and application nature.
Packings are V-ring, T-ring, O-ring, and U-cups, with special emphasis on O-ring packings made from natural rubber GR-S, GR-I (butyl) and Buna N, a nitrile type of rubber. Experience with thermoplastic materials negative.

C-025
Cheney, L.E. and Mueller, W.J., and Duval, R.E.
Frictional Characteristics of O-Rings with Typical Hydraulic Fluid
Am. Soc. Mech. Engrs.-Trans. V. 72, pp 291-7, April, 1950

O-ring packings employed in aircraft hydraulic systems are subject to numberous operating variables, which affect the friction involved in their operation. A method for studying the friction is described, and the effects of several variables-- pressure, time delay, squeeze, stroke speed, ring size, and surface finish of the moving metal part-- have been determined. Charts showing friction versus pressure, squeeze, time delay, surface finish, stroke speed and reproducibility.

C-026
Chinnery, R.
Bellofram Rolling Diaphragms
Hydraulic Power Transmission V. 8, pp. 325-7, May, 1962

Diaphragm design for frictionless packing for sealing pressurized fluid; applications

C-027
Chirkin, V.S.
Acid-resistant Packing Made of Polyvinyl Chloride
Khimicheskaya Prom., V. 2, p. 20, 1945

Packing for glands in systems carrying acids is made from plaited strips of polyvinyl chloride. To prevent raveling, the ends of the braid are fused together by heat.

C-028
Chittenden, W.A., Hoveke, G.F.
Heavy Water Reactor Plant Leakage
Atomic Energy Commission Report SL 1874, June 30, 1961

Test program results to appraise D_2O leakage rates from turbine shaft seals, pump shaft seals, and valve stems. Mechanical seals, packing seals, throttle bushings, and floating ring seals.

C-029
Christian, G.L.
Seals Designed for Supersonic Speeds
Aviation Week, V. 64, pp. 97-98, June 11, 1956

External sealing and molding for canopy cracks, flap edges, and other uses are now being designed and manufactured with the same extreme care that goes into any part of a supersonic airframe. Demanding seals which hold their shapes are highly scuff-proof, and resist rubber ruining ozone and temperature extremes ranging from - 100°F to 500°F. Silicone rubber family developed. Data is given, applications are discussed. Photographs.

C-030
Chupp, W.W.
Flanged Joint Sealing Gaskets
US Patent No. 2,532,891, Dec. 5, 1950,
U.S. Atomic Energy Commission

The gasket seal described in this patent comprises a pair of rings of compliant material arranged concentrically and in spaced relation to form an annular cavity. In association with the rings are one or more mechanical stops adapted to limit the deflection which the rings may undergo during compression and induce proper alignment of the surfaces which contact the rings.

C-031 B
Cianetti, E.
A Study of Behavior of Some Elastomeric and Plastormeric Materials for Packings
Min. Difesa Publ., pp. 23-35, 1956
In Italian MOS Translation TIL/T 4937, Oct., 1958

This report contains the results of a comprehensive study of various properties of several elastomers and plastomers. Particular attention has been given to their behavior when exposed to liquid hydro-carbons and low temperatures and to the change in their characteristics due to aging.

C-032
Clark, A.F.
Careful Design Prolongs Packing Life
Machine Design, V. 5, pp. 29-30, 39, June 1933

Selection of material for particular application; materials available are leather, felt, cork, cotton, asbestos, and rubber.

C-033
Clark, A.F.
Sealed and Shielded Bearings Aid in Design Simplific[illegible]
Machi[illegible] De[illegible]gn, V. 6, No. 8, pp. 22-24, Aug., 1934

Bearings with shields and seals combine several necessary details into a single unit, resulting in longer life, and ease of application. Advantages, limitations and applications.

C-034
Clark, E.H.
Mechanical Packing Design; Cup and Flange Types
Prod. Engrg., V. 16, pp. 474-475, July, 1945

Design features and certain basic rules for designing cup and flange packings are presented.
Cup packing used mainly on pistons, plungers or rams on hydraulic or pneumatic service.
Flange packings used in packing recesses both for reciprocating plungers and as seals on rotating shafts. Types are well illustrated.

C-035 B
Clark, F.E., Ball, W.P., and others
Molding of O-Ring Packings
Rubber Age, (NY) V. 62, 61, pp. 652-4, March, 1948
Also, P.B. 94015 Off-Tech. Serv., March, 1948

C-036
Clark, J.F.
Sealing Studies on Cartridge Actuated Devices
PB-138726, Off. Tech. Ser., 33 pgs., (Rept. No. R-1469), Sept. 1958

Investigations were conducted using quad rings as sealing media. Each of these products was found to be effective in specific applications; however, the O-ring was particularly adaptable to existing cartridge actuated devices and was used almost exclusively in the modification made under this program.

C-037
Clark, P.M.
Freeze Seals
Knolls Atomic Power Lab., Gen. Elec. Co., Schenectady, N.Y., Contract W-31-109 Eng-52, 6 p., July 14, 1952

Results are presented from tests on an air-cooled freeze seal suitable for use in submarine thermal reactor service lines. The seal is considered a one-way freeze seal because it will pass gas in one direction only without emptying the liquid from the seal.

The test arrangement is illustrated and performance data are summarized.

C-038 T
Clark, P.M.
Mechanical Pumps for High Temperature Liquid Metals
Mechanical Engineering, V. 75, pp. 615-8, August, 1953

Most of this article is devoted to the seals developed for obtaining zero leakage of sodium-potassium mixtures at temperatures greater than 750°F. A long overhung, oversized shaft was used with an inert gas sealed labyrinth seal which was surrounded by a cooling oil jacket. To seal the gas, which might absorb NaK vapors, a rotary face seal was employed above the labyrinth seal. The seal was lubricated with approximately 30 drops of oil per day which was collected in a trap and drained periodically. Seal life was generally more than 2000 hours. Test to determine best seal was run.

C-039
Clark, R.A. and Cheyney, L.E.
Compounding of O-Ring Hydraulic Packings
Rubber Age (NY), V. 65, pp. 561-6, 1949

The requirements of O-ring hydraulic packings for service in aircraft, automobiles, locomotives, hydraulic presses and many other types of equipment and machinery are reviewed with special reference to Army-Navy.
Aeronautical Specification AN-P-79.

C-040 B
Clark, R.A., Kell, R.M.
Swelling and Drying of Fuel O-Rings
Rubber Age (NY), V. 75, 6, pp. 851-7, September, 1954

The purpose of this investigation was to shorten a full O-ring test procedure by using temperatures up to 250°F to accelerate the rate of O-ring swelling and drying. Free and confined samples from five commercial sources were tested.
30% aromatic fuel - 65° to 250°F.

C-041
Clark, R.A.
TFE Plastic O-Rings
Machine Design, V. 34, pp. 221-222, Apr. 26, 1962

Tetrafluoroethylene plastic O-rings are used for static and dynamic seals, but are frequently unsatisfactory for dynamic seals because of poor wear characteristics. Their surfaces are easily damaged, so they must be installed carefully and against smooth surfaces to avoid leaks.
They have good chemical and heat resistance, low coefficient of friction, flexural strength and toughness. They can be heated, stretched and shrunk fit on diameters as much as 30% greater than the O-ring groove.
Groove design for the rings, and their deformation under load are discussed. To avoid creep failures a means of maintaining pressure such as heavy initial loading, spring-loading, use of adjustable glands, or use of filled TFE O-rings should be employed.

C-042
Cline, R.W.
Packing Gland Redesign Reduces Pump Maintenance
Applied Hydraulics, V. 8, pp. 100, 102, 104, October, 1955

System developed for Goshen Rubber Co., Goshen, Ind.; adapter made of bearing bronze fills space originally occupied by packing; one groove contains O-ring operating as dynamic piston rod seal, other containing second O-ring operating as static seal; application to presses in rubber molding shop, powered by weighted accumulator system charged with water at 3000 psi from Triplex pump.

C-043 T
Clinebell, B. J.
O-Rings - An Annotated Bibliography
India Rubber World, V. 27, pp. 74-78, 1952-53

Eighty-nine references are listed.

C-044 B
Clond, R.W., Philip, S.F.
Vacuum Test of Rubber, Lead, and Teflon Gaskets and Vinyl Acetate Joints
Rev. Sci. Instrum., V. 21, pp. 731-3, Aug. 1950

Rubber, Teflon, and Vinyl Acetate sealed systems are compared, by an approximately quantitative method, with a lead gasket sealed system.
Time pressure curves, data, tables presented.

C-045
Close, G.C.
Improved Rubber O-Ring for Hydraulic Service
Matls. & Methods, V. 42, p[illegible] 1955.

New molding technique, cal[illegible] developed at Plastics and Rub[illegible] has practically eliminated interna[illegible] id stresses within structure of O-ring[illegible] ages of new molding method are that time lag[illegible] ig die cavity has been reduced to vanishi[illegible] t and amount of material introduced into die ca[illegible] i exactly equal to amount of material requi[illegible] fc part.

C-046
Cochran, R.P., and Curren, A.N.
A Balanced-Pressure Sliding Seal f[illegible] Tra[illegible]er of Pressurized Air Between Stationary and Ro[illegible]ting Parts
NACA RM E56E11, Jan. 9, 1957, p. 17

This seal was designed in connection with experimental investigations of air-cooled turbine rotor blades in turbojet engines where high sliding velocities and high ambient seal operating temperatures are encountered; sliding velocities up to 10,000 feet per minute and air pressures up to 35.3 PSIA.
Rate of wear on the seal rings was immeasurable on the inner ring and about 0.0005 inch per hour on the outer ring.

C-047
Coffman, B.T.
Seal for Rotating Shaft
U.S. Patent 2,835,986 to A.E.C.

A seal is described for a rotatable shaft that must be highly effective when the shaft is not rotating but may be less effective while the shaft is rotating.

C-048 T
Coffey, J.J.
Sealing a Centrifugal Pump Shaft
Power, V. 93, pp. 118-9, Sept., 1949

This is a descriptive article covering stuffing box seals, lantern rings, mechanical seals, and mechanical seals with auxiliary gland packing and vent. Application to centrifugal pumps are considered.

C-049
Calhoun, C.E.
Stopping a Turbine Steam Chest Leak by Special Gasket
Power, V. 61, p. 997, June 23, 1925

Brief article describing the reasoning and results of replacing ring gaskets by a full-face gasket. Drawings included.

C-050
Collier, M.
A Feed Screw Powder Seal For The Reactor Tower
Machine Design, V. 31, p.118, Nov. 26, 1959

The two-way sealing of a feed screw for a fine powder conveyor is accomplished with a split gland. The correct installation of the gland is discussed.

C-051
Cosma, G., Lungen, S., Simonescu, C.
Metal-to-Metal and Ceramics-to-Metal Bakeable Vacuum Seals
Am. Vac. Soc., 1961 Trans., Eighth Nat. Vac. Symposium, Second International Congress, V. 2, pp. 1319-1322

Two methods are described for the construction of vacuum tight and bakeable metal-to-metal seals, soldered together with aluminum gaskets. The metal-to-metal seals are lighter, smaller, and cheaper than the usual flange seals with metal gaskets. They can easily be taken apart.

C-052
Conger, C.B.
Piston Rod and Valve Stem Packing
Loc. Engrg., 1500W., July, 1900

Describes the two general types of metallic rod packing in use, and the care it needs.

C-053
Conklin, E. W.
Ceramic Rings Used For Mechanical Rotary Seal
Materials and Methods, V. 45, p. 157, May, 1957

One of the most critical parts in mechanical rotary seals is the rotating seal ring.
Tests were conducted to find materials which would perform better than metal seal rings.
Alumina ceramics were found to make superior rings.
Have high hardness, resistant to all acids except hydrofluoric and strong caustics. Retain full strength to 2000°F and support own weight to 3000°F. Also useful for speeds of 8000 to 20,000 rpm and high pressure uses.

C-054
Connelly, R.E. and Wolf, J.E.
Development of Seals for Rocket Engine Turbopumps
Am. Soc. Lub. Engrs. Trans., V. 2, pp. 25-31, 1959

The design and development of oxidizer, fuel, gas, and oil seals for use in turbo pumps are reviewed. With the following conditions surface speed of 8000 ft. per min; test pressure of 200 psi and face loading of 100 psi.

C-055
Conway, H.G. (editor)
Aircraft Hydraulics VII Component Design
Chapman & Hall, London (TL 697 H9C76a), 198 pp., 1957

Seals, hydraulic pumps and motors; jacks; valves; hydraulic servo-controls; pipe work.

C-056
Cooke, B.
Overheating in Rotary Oil Seals
Engr., V. 187, pp. 223-225, Feb. 25, 1949

Describes temperature measurements made on a rotary seal, which consisted of three parts, namely, a spreading ring, a "U" seal, and a crown header.

Tests specific for crown header. Material neoprene rubber.

C-057
Cooke, B.
Low Pressure Reciprocating Seals for Hydraulic Control Valves
I.M.E. Proc., V. 165, pp. 262-265, 1951

Technique of designing a synthetic rubber lip type of rotary seal so that it is "run-in" on the shaft was applied to finishing the lip of a seal for reciprocating duty.
Paper illustrates that by specially treating a lip seal it is possible to obtain results that have not been equalled. The running-in process used is hardly a production proposition, and some easier way of producing a clean lip must be devised.

C-058 B
Cooke, B.
Pipe Joints For Hydraulic Power Transmission
I.M.E. Proc., V. 164, 3 pp. 308-23, 1951

Vigorous vibration and shock pressure tests, intended to represent the worst conditions possible in marine practice. Test results presented.

C-059
Cooke, B.
Pipe Joints for Hydraulic Power Transmission
Engineering, V. 171, 413-16, 1951

A discussion of the principal advantages and limitations of eighteen different pipe joints, and test results of same. The effect of vibration and shock on joints and sealing materials used.

C-060 B
Cooke, B.
New Gland for Hydraulic Stop Valve Spindles
Engineer, V. 191, 4958, pp. 170-1, Feb. 2, 1951

Short article describes the replacement of normal packed glands on valve spindles by sleeves fitted with O-ring seals. The constant supervision and tightening of soft packed glands is avoided. O-rings successful at pressures of 1500 lbs/sq. in. and do not increase the effort to operate the valve.

C-061
Cooke, B.
Suggestions for Leakage Control in Hydraulic Systems
Applied Hydraulics, V. 6, pp. 38-40, Feb. 1953

Ideas are presented on the proper use of packings which will help one to reduce oil loss and machine downtime.

C-062
Coopey, W.
How To Seal Rotating Shafts Against High Pressures
Chem. Engrg., V. 58, pp. 116-117, July, 1951.

High pressure shaft sealing generally is solved with one of two approaches. Paper describes these approaches and illustrates concepts involved with the description of two rotary seals.

C-063
Coopey, W.
How To Pack Reciprocating Rods Against High Pressures
Chem. Engrg., V. 58, pp. 164-166, Nov., 1951

To design adequately for the severe service conditions found in many high-pressure chemical processes, one must understand and appreciate the fundamental principles of packing. These fundamentals are alignment, quality of materials, finish of materials, lubrication and application. These are individually discussed.

C-064
Coopey, W.
Holding High Pressure Joints
Petroleum Refiner, V. 35, pp. 189-193, May, 1956

C-065
Coopey, W.
How To Solve Soft Packing Problems
Chemical Engineering, V. 65, pp. 131-134, Jan. 27, 1958

Problems attendant to the life of the packing in centrifugal pumps is discussed.
Emphasis is placed on the proper application of the lubricant to critical spots and an attempt is made to remove as many human factors as possible attendant to the original installation.

C-066 B
Coopey, W.
Reciprocating Rod Packings Need Not Be Trouble Spots
Chemical Engineering, V. 68, pp. 113-6, August 21, 1961

Discussion explodes popular misconceptions about packings. Lapped edges, rounded, replace serrated edges of packing elements. Drawings - Conventional vs. Spring-loaded arrangements.

C-067
Cornell, R.L. and Lucas, R.S. and Young, H.L.
Friction and Wear For Water-lubricated Seals
Product Eng., V. 32, pp. 38-41, July 24, 1961

Results of comprehensive investigation of seal wear under boundary lubrication, conducted at Massachusetts Inst. of Technology, indicate that graphite against chrome-plated steel gives best combination of low wear rate and low friction.

C-068
Cornish, H.E., Bloom, J.C.
Development of High Pressure Seals For A.N. Straight Threaded Fittings
Applied Hydraulics, pp. 18-24, Nov., 1949

Describes development and test of an improved O-ring boss seal.
Pressure pulsation tests showed superiority of new design.

C-069 T
Cornish, R.J.
Flow of Water Through Clearances With Relative Motion of the Boundaries
Proc. Royal Soc. of London, Series A, V. 140, pp. 227-240, 1933

Description of experiments on the resistance to flow of water through a series of fine annular clearances in which the cylindrical bush forming the inner boundary could be rotated.

C-070
Cord, G.L.
Transmission Oil Seals
SAE Paper No. 119u, 10 pp., Oct. 26-28, 1959

Characteristics of silicone polyacrylic (PA-21) and butyl acrylic (BA21) high temperature materials employed in manufacture of seals; progress in mechanical shape of oil seals, when coupled with new synthetic elastomers; advance made in technique of excluding dirt and mud; application information required by oil seal supplier to assure satisfactory performance from selected seal.

C-071 T
Cowie, D.B.
High Vacuum Seal
Rev. Sci. Instruments, V. 15, No. 2, pp. 46-7, Feb., 1944

Wilson's design of vacuum seal with some translational and rotational motion of tubing through a pair of ceramic, or metal plates. 45° bevel on inner faces of the plates retain a neoprene gasket around lead through tubes.

C-072 T
Cowlin, F.J.
The Lubrication of Steam-Turbine Driven Electric Generators
I.M.E. Proc., V. 143, 1940, pps. 98-109, discussion pp. 175-183

Brief review of systems and problems of the lubrication of large steam turbine generators. Discusses general requirements of lubricating systems and components such as coolers, filters, piping, pumps, valves, controls, etc.

C-073 T
Creed, F.R.L.
Liquid Seals For Gas-cooled Dynamic Electric Machines
U.S. Patent 2,805,090, Sept. 3, 1957

A means of sealing a gas-cooled machine by use of oil pressure to force an annular piston with a thrust bearing surface on one end against a collar on the shaft. Oil, not necessarily at the same pressure is pumped to the thrust bearing surface which forms the actual seal.

C-074
Crego, D.F.
Sealing High Pressure Gas
Machine Design, V. 27, pp. 162-168, Sept., 1955

Article reviews characteristics of shaft sealing elements and systems developed for high pressure gases. Discussion specifically concerned with centrifugal compressor applications but designs illustrated have broader implications. Divided into two sections, article deals first with several basic high pressure seals or sealing elements. Then from standpoint of design and application, a number of gas sealing systems, which employ one or more of these fundamental seals, are covered in detail.

C-075 T
Crego, D.F.
Centrifugal Compressors: Seals And Sealing Systems
Petroleum Refinery, V. 34, pp. 143-6, Jan., 1955
Petroleum Engineer, V. 28, pp. C17-22, Feb., 1956

Straight pass labyrinth, staggered labyrinth, segmented carbon rings, and contact or mechanical seals are discussed. Three types of sealing systems, evacuation, gas injection, and fluid injection, are also discussed.

C-076 T
Cronstedt, J.
Turbine Seal
U.S. Patent 2,410,340, Oct. 29, 1946

Labyrinth seal design to eliminate the effects on the seal elements of shaft flexures and heat transfer through the shaft. The rotating seal element is a sleeve which is mounted on the shaft and forms the shaft bearing surface.

C-077 B
Crosland, H.C.
The Effect of Hydraulic Fluids On Synthetic Rubber
Proc. 10th N.C.I.H., V. 8, pp. 190-4, October, 1954

A discussion of the proper synthetic rubber to be used with each basic type of hydraulic oil and the limitations put upon this compatability of rubber and fluid by temperature, pressure, and other conditions of application. Polysulfides, nitrile rubbers, chloroprenes, styrenes, isoprenes, poly acrylates, silicones, trifluorochloroethylene and tetrafluoroethylene resins.

C-078
Cultra, R.A.
Packing For Hydraulic - Elevator Valves
Power, V. 39, pp. 511-2, April 14, 1914

A description of a method of renewing cup leathers on pilot and main hydraulic elevator valves. Also the manner of making the leather packing cups at the plant. Drawings included.

C-079
Cygan, R.
Project Freeze Seal
North American Aviation Inc., Downey, California
NAA-SR-MEMO-1565 19P. Jan. 25, 1956

Five freeze seal designs for the main SRE pumps were built and tested. The design of seals is presented along with the results of testing.

C-080
Cygan, R. and Stelle, A.M.
Frozen Slugs Seals Shafts
Chem. Engrg., V. 63, pp. 124, 126, No. 3, March, 1956

Operating principle of new seal is formation of barrier between liquid-metal system and outside by freezing slug of metal in narrow passage;frozen metal is held in passage, which is usually annulus, by friction and adhesion.

C-081 T
Cygan, R., Stelle, A.M.
Design and Operation of Freeze-Seal Valves and Pumps
Chemical Engineering Progress, V. 52, pp. 157-9, April, 1956

This article describes the development of such seals for pump and valve shafts. Apparently there is no maintenance needed, provided the liquid metal is prevented from oxidizing by an envelope of inert gas. The starting torque is high, but figures given in the article show that running torque is low if the pressure differential is small.

D-001
Dabney, M.S., Holt, W.W., Jr.
Mechanical Seals Effective on Main Line Pumps
Oil and Gas Journal, V. 49, pp. 57-8, Jan. 4, 1951

A description of test run on mechanical seals as applied to main-line propane pumps is given. Preliminary test on double face seals with seal oil between, lubricated single seals, and unlubricated carbon to metal single seals.

D-002
Dahlstrand, J.Y.
Packing Glands for Steam Turbines
Power Pl. Engng., V. 29, pp. 740-741, July 15, 1925

A discussion of various types of gland packings used in steam turbines and difficulties experienced with each kind.

D-003
Daniels, W.B., and Hruschka, A.A.
Seals for Pressures to 10,000 Atmospheres
Review of Scientific Instruments, V. 28, pp. 1058-1061, Dec. 1957

This note discusses the use of "armored" - O-ring seals for the pressure range to 10,000 atmospheres. In addition a high-pressure pump piston seal and a proprietary surface treatment to reduce liability of galling of steel parts are described.

D-004
Datz, S.
Packing Materials for Hydraulics and Pneumatics
Applied Hydraulics & Pneumatics, V. 12, pp. 115-118, July 1959

Listed are materials marketed for use in hydraulic and pneumatic components along with application information for each type.
Materials include; synthetic rubbers, fluoro-elastomers, polyacrylates, leathers, metals, asbestos. Temperature ratings of each and compatibility with liquids indicated.

D-005
Davenport, F.E.
Pointers on Asbestos Gaskets
Power, V. 83, p. 189, April 1939

Brief directions for their practical use in connection with power plant piping.

D-006 B
Davies, A.J.
A Note on the Use of Polytetrafluoroethylene in Vacuum Seals
Journal Scientific Instruments, V. 35, pp. 378-9, Oct. 1958

A short article describing the procedure used to obtain a seal that may be cooled and reheated repeatedly without deteriorating.

D-007
Davies, H.
Sealing Elements for Rocket Application
Mach. Design, V. 26, 333-4, April 1954

Sealing problems encountered in rocket engines are discussed. Desirable conditions are pointed out.
Metal to-metal seals, semi resilient seals, resilient seals.

D-008 T
Davies, M.G.
Effect of Eccentricity and Whirl on Back Diffusion in Sleeve Glands
United Kingdom Atomic Energy Authority, IG-R/CA 159, 1959, 18 pp.; also Int. Conf. on Fluid Sealing, Paper C4, April 17-19, 1961, British Hydromechanics Research Assn.

The diffusion of one gas against the stream of another gas, in annular space between a sleeve gland and a rotating shaft is considered. Concluded experimentally, that the simple diffusion theory is inadequate when shaft runs eccentrically or whirls in sleeve; effective length can easily be halved. Explanation suggested.

D-009 T
Davies, M.G.
The Generation of Lift by Surface Roughness in Radial Face Seals
Int. Conf. on Fluid Sealing, Paper E-4, April 17-19, 1961, B.H.R.A.

The idea is developed that large pressures of questionable origin in oil filled clearance between surfaces of radial face seal, is due to hydrodynamic lubrication effects in small wedge shaped cavities constituting surface roughness. The consequence of the theory in regard to leakage and seal design are discussed.

D-010
Davis, P.C., Gore, T.L., Kurata, F.
High Pressure Metal-to-glass Fittings
Ind. Engr. Chem., 43-8, pp. 18, 26-7, Aug. 1951

The fitting described has been designed to seal glass capillary tubing to a metal fitting and to withstand high fluid pressures and vibrations. Sealing is effected by placing an O-ring between the ends of the metal and the glass tubing which is ground square. The tubes are held together by a locking screw. (3000 lbs/sq. in - 400°F)

D-011
Davis, V.H.
Packing That Pump
Southern Power and Industry, V. 59, pp. 68-9, Sept. 1941

Methods for packing pumps and valves given, which may suggest some new ideas in packing procedure.

D-012 B
Dawson, L.J.
Development and Design of Mechanical Seals for Centrifugal Pumps
Ingersoll-Rand, 1950

A description of the origin and development of seals employing radial mating surfaces serves to show which features of the whole gland inherently cause the most trouble. Illustrations and designs.

D-013
Dawson, P.
Investigation of Cavitation in Lubricating Films Supporting Small Loads
Proc. Conf. on Lubrication and Wear, Paper 49, pp. 93-99, Oct. 1-3, 1957, Instn. Mech. Engrs., 1 Birdcage Walk, London, SW 1

Bearing surfaces in the form of stationary spherical cap and a plane slider have been used to provide visual observations and pressure records of the conditions in the clearance space under full and cavitating conditions. Satisfactory correlation of the pressure records and visual records has been obtained. A departure from predictions is noted in the form of a down stream shift of the oil film rupture point.

D-014
Dawton, R.H.V.M.
High Vacuum Shaft Seals, Flanged Joints and the Gassing and the Permeability of Rubber-like Materials
Brit. Jour. App. Physics, V. 8, pp. 414-421, Oct. 1957

Object of paper is to record the results of some vacuum measurements, e.g., gassing rates and permeability rates, on shaft seals and flanged joints, and also to measure the effect of rubber-like materials when used on continuously pumped high vacuum equipment.
Materials tested were rubber type J.1260 from Premier Waterproof and Rubber Co., Ltd.; cured rubber from Greengate and Irwell Rubber Co., Ltd.; neoprene GN from Spencer Moulton and Co.; and butyl of code number Bu 12000, from Firestone Tire and Rubber Co.
A silastic gasket was tested at 200° to 300°C.

D-015 B
Decker, A.L.
Mechanical Seals for Nonlubricating Hydrocarbons
A.S.M.E. Paper No. 57-P.E.T.-2, See also Oil & Gas Journal, V. 56, No. 1, pp. 93-96, Jan. 6, 1958

A brief review of the basic types of mechanical seals, and mechanical seal installations. Test program is described for selection of seals for the less common services. Test group was comprised of 36 centrifugal pumps, 28 of which were handling fluids composed of 90 to 99 percent ethyl chloride, with varying quantities of ethane, hydrogen chloride, and ethylene dichloride.

D-016 T
Decker, A.L.
Sealing of Hazardous Fluids
Int. Conf. on Fluid Sealing, Paper H2, April 17-19, 1961 (6 pp., 2 plates), British Hydromechanics Research Association-Harlow, Essex, England

Paper outlines general types of equipment and methods employed by one chemical company in the handling and sealing of several hazardous fluids. Types of equipment and pertinent features outlined and illustrated.

D-017
Dega, R.L. and Symons, J.D.
Seal Testing to Establish Quality Control Specifications Can Reduce Leakage
SAE Jour., V. 68, pp. 127-128, No. 4, April 1960; SAE Trans., V. 68, pp. 440-447, 1960, SAE Paper No. 130B

Investigation of causes for seal leakage and determination of basic design criteria; major parameters of seal application which affect its efficiency and life as determined by controlled laboratory testing are shaft surface roughness, machining lead, assembly and seal. Most important parameters are seal diameter control, lip pressure, and eccentricity; air gage developed called "Sealector."

D-018 B
Dega, R.L.
Recent Advances in Lip Seal Technology
National Conference Industrial Hydraulics, 14 pp., 154-63, Oct. 1960

Presents the variables which affect seal operation; shaft; assembly; seal, lip dimension variables, diameter, pressure, eccentricities; lip material quality. Also types defects with photos. Presents data curves, temperature vs. speed, hardness variation due to curing time, friction coefficients for various hardness and surface finishes, etc. and discussion.

D-019
Dega, R.L.
Lip Seal Failures - Cause and Prevention
Design News, V. 16, pp. 8-9, June 5, 1961

Establishes that seals leaked because of tolerance excesses in the following lip variables; diameter, pressure, eccentricity, and material quality.

D-020
Deich, V.E., Samoilovich, G.S.
The Comparison of Experimental and Design Data on Leakage Through Labyrinth Seals
(DSIR LVU) (12 pp., 1961) (M3090), Order from OTS or SLA Translation No. 61-27552, Trans. of Mono. Aerodynamic Principles of Axial Turbomachines

Steam turbines, turbines, disks, shafts, gas flow, seals, gas seals, design, tests.

D-021
Dempsey, R.G.
O-Ring Maintenance
Applied Hydraulics, V. 8, pp. 106, 108, 111-113, Oct. 1955

Procedure for obtaining optimum characteristics of packings used in commercial aircraft hydraulic systems; schedule for inspections, replacements, and rework.

D-022 B
Denny, D.F.
Experiments on the Static Friction of Rubber Seal Materials
British Hydromechanics Research Association

D-023
Denny, D.F.
Friction of Flexible Packing
I.M.E. Proc., V. 163, (WEP No. 57) 98-102, 1950

At any time there are anomalies and difficulties in the prediction of friction, and these are accentuated if one of the surfaces is deformable, as in flexible piston seals, since the surface strain introduces an additional variable. Such surfaces, although exhibiting the three usual kinds of friction-dry, boundary, and film, show peculiar characteristics at low speeds, when friction is usually less than at higher speeds. The static friction itself depends on the time which the surfaces have been in contact, and preliminary work indicates that there is a law connecting these two quantities, the probable form of which is given in this paper.

D-024
Denny, D.F.
Experiments on Radial Face Seals
British Hydromechanics Research Association, Report No. R.R. 425, 1952

D-025
Denny, D.F.
The Effect of Shaft Eccentricity on the Performance of Soft Packed Stuffing Boxes
British Hydromechanics Research Association, RR 444, 15 pp., 11 fig., Dec. 1952

Describes experiments with shaft rotated eccentrically by various amounts. Includes a description of test on a device designed to prevent the excessive leakage that accompanies eccentricity.

D-026 B
Denny, D.F.
The Influence of Load and Surface Roughness on the Friction of Rubber-like-materials
Proc. Phys. Soc., B66, pp. 721-7, 1953

Experiments with rubber and gelatin under lubricated conditions confirmed that with rubber-like materials the load dependence of friction is consistent with elastic deformation of surface asperities and partial adhesion over the true area of contact. However, the relation that the friction force is proportional to the two thirds power of the normal load found by previous investigators pertains only over a limited load range. At very low loads the friction is very nearly proportional to load, while at very high loads contact between specimen and track is complete, and the friction remains sensibly constant. Where rubber slides on rough tracks there is a large additional friction force unrelated to true contact area.

D-027 T
Denny, D.F.
The Problem of Fluid Leakage
Fluid Handling, V. 4, pp. 62-64 - March 1953, pp. 92-94 - April 1953, Compressed Air Engineering, pp. 15-21, January 1953

A discussion of reciprocating seals and rotating seals. Included are U-rings, stuffing boxes, synthetic rubber square and round cross section seals. Also circumferential seals, mechanical face seals and oil seals. Presents the operating conditions which can be met with currently available materials.

D-028 B
Denny, D.F.
Tests on Packed Glands for Rotating Shafts
British Hydromechanics Research Association, RR 487, 5 pp., 6 fig., July 1954

D-029 B
Denny, D.F.
Test with O-Rings and Lip Seals on Rotating Shafts
British Hydromechanics Research Association, RR 525, 4 pp., 7 fig., Feb. 1956

Describes some tests made to determine the suitability of O-rings for sealing rotary shafts, when fitted in canted grooves.

D-030 T
Denny, D.F.
A Force Analysis of Stuffing-box Seal
British Hydromechanics Research Association, R.R. 550, 1957

D-031 B
Denny, D.F.
Further Tests on Packed Glands for Rotating Shafts
British Hydromechanics Research Association, RR 561, 3 pp., 9 fig., July 1957

Measurements of fluid pressure gradients have been made on stuffing-box seal fitted to a rotating shaft.

D-032 B
Denny, D.F.
Leakage and Friction Characteristics of Some Single-Lip U-Seals Fitted to Reciprocating Shafts
British Hydromechanics Research Association, RR 595, 8 pp., 16 figs.

Fifteen designs of single-lip U-seal have been compared, for sealing reciprocating shafts against leakage of high pressure oil.

D-033 B
Denny, D.F.
The Work of the Hydromechanics Laboratory With Special Reference to Work on Seals
Proc. 14th British Iron and Steel Research Association (Hydraulics in the Steel Industry), pp. 94-107, disc. pp. 107-8, March 1957

Description of work at the British Hydromechanics Research Association.

D-034
Denny, D.F.
The Lubrication of Fluid Seals
IME Proc. of Conf. on Lubr. & Wear, pp. 392-395, Oct. 1, 1957

Ideally a seal should allow none of the fluid to escape from its container, but with sliding seals, whether for rotary or reciprocating shafts, it is normally necessary to provide reasonable lubrication of the sliding parts if adequate life of seal is to be achieved. Practical implications of the type of lubrication are discussed and it is demonstrated that while no seal can be expected to prevent leakage completely, some by their very nature must allow appreciable leakage flow.

D-035
Denny, D.F.
Recent Research on Hydraulic Seals
Scientific Lubrication, V. 10, pp. 12-17, Sept. 1958

Gives examples of research at the British Hydromechanics Research Assoc. Examples discussed are measurement of rubber friction; striction effect in rubber; leakage of rubber seals; O-rings for rotating shafts and oil-seals for eccentric shafts.

D-036 B
Denny, D.F.
Time Effects in the Static Friction of Lubricated Rubber
Wear, V. 2, pp. 264-272, May 1959

The considerable increases in static friction that occur when lubricated rubber remains at rest under load are shown to be due to squeezing out of the lubricant films from between the surfaces.
The time-scale of the phenomenon appears to be directly related to the bulk viscosity of the lubricant and to the elastic modulus of the rubber.

D-037
Denny, D.F.
Some Measurements of Fluid Pressures Between Plane Parallel Thrust Surfaces and Special Reference to Radial Face Seals
Wear, V. 4, No. 1, pp. 64-83, 1961

D-038
Denny, D.F.
Choosing Right Seal
Hydraulic Power Transmission, V. 8, pp. 32-6, Jan. 1962

Application of appropriate types of seals for cylinder design, characteristics of seal types including metal piston-ring seals, double acting piston seals, etc.

D-039
Denny, D.F., and Turnbull, D.E.
Inverted Stuffing-Box
Engineer, V. 205, pp. 617-618, April 25, 1958

When leakage in a conventional stuffing-box is reduced by tightening the gland bush, there is often overheating, binding and excessive shaft wear. These are usually avoided by permitting a large cooling leakage. However, by inverting the stuffing-box (allowing the packing to rotate with the shaft and the rubbing to take place between the packing and casing) cooling of the rubbing surfaces is greatly facilitated. The inverted arrangement permits operation with a tighter gland bush. Leakage rates comparable with mechanical seals are achieved. Test results are given.

D-040
Denny, D.F., and Turnbull, D.E.
Sealing Characteristics of Stuffing-Box Seals for Rotating Shafts.
IME Proc., V. 174, pp. 271-291, No. 6, 1960

There are two distinct modes of operation of a stuffing-box seal. One mode occurs when the axial pressures produced in the packing by tightening the gland-bush are greater than the sealed fluid pressure and the other when they are less. Under former conditions the fluid pressure distribution is an exponential function of the packed length and the friction torque increases rapidly with this quantity. With high fluid pressures, however, the relation between fluid pressure and packed length is no longer exponential and most of the fluid pressure drop occurs over the last 10 percent of the packed length. In addition very little increase in frictional torque occurs when the packed length is increased. Theoretical expressions derived show good agreement with experimental results.

D-041
D'Eustachio, D.
A Useful Seal for Dynamic Vacuum Systems
Rev. Sci. Instrum., V. 16, pp. 377-8, Dec. 1945

An easily constructed vacuum seal is described. The seal, which uses an ordinary rubber stopper, has been found useful down to 10^{-6} mm Hg. It can be broken and remade rapidly. Precise location within the vacuum system is possible, and the adjustment can be made from outside the vacuum systems.
It has been used for introducing electrodes into, and connecting glass tubing onto dynamic vacuum systems.

D-042
Dezlih, C.J., and Hill, J.J. and others
Design Data for O-Rings and Similar Elastic Seals
Boeing Airplane Co., WADC-TR-56-272(Pt. III), Contract AF33(616)-2867, AD-151181, Feb. 28, 1958, 96 pp., Project 7340

A complete literature survey on O-rings, high and low temperature design, relaxation and volume change tests; with and without back-up rings, etc.

D-043
Diefenbach, G.
Fields of Application for Packings Without Stuffing Boxes. (In German)
Chemie-Ingenieur-Technik., V. 23, pp. 491-4, Oct. 28, 1954

Examples given to show how stuffing boxes can be replaced by automatic spring and slip ring packings; application to pumps, dryers and mixers; illustrations.

D-044 T
Diefenbach, G.
Slide Ring Seals for Rotary Shaft (In German)
Chemie-Ing-Technik, V. 26, No. 7, 1954

Slide ring seals are increasingly used for rotary shafts, because they seal just as well and automatically as gaskets, they can also be used in the case of hard materials. The application of slide ring seals for sealing against pure liquid or homogeneous viscous substances has already been mastered to a very high degree. (Slide ring seals are referred to as mechanical face seals in the U.S.)

D-045
Diegmann, H.
Proper Use of Stuffing-Box Packing (In German)
Maschinenschaden, V. 11, pp. 5-8, 1934

Aspects governing choice of packing; suggestions for carrying out quality tests; packing for wet and superheated steam plants, cold-and-hot water pumps, feed pumps, ammonia compressors, etc.

D-046
Diegmann, H.
Proper Design of Metal Stuffing-Box Packings (In German)
Maschinenschaden, V. 12, pp. 94-6, 1935

With reference to water, steam, gases, liquids under pressure, cooling water, and flue gases, chemical apparatus and machinery and condensers; illustrations.

D-047
Diegmann, H.
New Methods for Manufacture of Stuffing-Box Packings (In German)
Gummi-Ztg., V. 49, p. 496, May 17, 1935

Improvements in design of rapid braiding machines briefly described and illustrated.

D-048
Diegmann, H.
Rules for Application of Various Packing Materials. (In German)
Maschinenschaden, V. 13, pp. 153-5, 1936

Notes on metal base packings for stuffing boxes; steam engines, soft packings according to build up system, asbestos, metal base packings, klingerit packings, installation and replacement of piston rings.

D-049
Diegmann, H.
Series of Exerpiments Regarding Various Packing Materials Functioning under Hydraulic Pressure. (In German)
Werkstattstechnik u Werksleiter, V. 31, pp. 133-6, March 15, 1937.

Characteristics of different packing materials & tabulations of types according to pressure applied, applications.

D-050
Diegmann, H.
New System of Stuffing-Box Design for Movable Machine Parts Using Synthetic Rubber (In German)
Werkzeugmaschine, V. 42, pp. 309-12, July 31, 1938

Various designs illustrated.

D-051
Diegmann, H.
Use of German Materials in Packing (In German)
Schiffbau, V. 40, pp. 83-5, March 1, 1939

Examples of application described and illustrated, with special reference to so-called ferrolastic packing, synthetic rubber packing etc.

D-052
Diegmann, H.
Substitute Materials in Packing (In German)
Kunststoffe, V. 31, pp. 93-6, March 1941, also Glasers Annalen, V. 65, pp. 117-9, March 15, 1941

Synthetic rubber perbunan used for shafting packing instead of leather; comparison between packing properties of leather and its substitute.

D-053
Diehl, K.
Progress in Using Buna Rubber in Engine Construction
PB 24338, Off. Tech. Serv., 7 pp., 1940

This article discusses the most important characteristics of buna rubber, when used for engine construction especially; (1) moisture resistance, (2) heat resistance, (3) mechanical properties, (4) sliding properties, (5) suitability for packing. Finally various types of rubber packings are discussed; especially the heat resisting newly developed B.S. packing. Photographs and graphs are included.

D-054 T
DiPrima, R.C.
The Stability of a Viscous Fluid Between Rotating Cylinders with Axial Flow
Jnl. Fluid Mechanics, V. 9, pp. 621-631, 1960

Attack on stability problem by superimposing a rotationally symmetric disturbance on the steady flow equations. Results compared with those of Kay and Elgan, Fage, Goldstein, and Cornish. Results disagree with last three, but in qualitative agreement with Kay and Elgan. Found that critical Taylor number increases with increasing Reynolds number.

D-055 T
Dollin, F., Brown, W.S.
Flow of Fluids Through Openings in Series
Engineer, V. 164, No. 4259, pp. 223-224, Aug. 27, 1937

Beginning with the general polytropic ideal gas equation, continuity equation, and work-kinetic energy relation, derived a general flow equation. The equations of Martin, Callendar, and Stodola for labyrinth flow as well as flow equations incompressible flow and adiabatic expansion are all obtained from the basic equation. No experimental or design data are given, but the theoretical treatment is straightforward and clear.

D-056
Donald, M.B., and Salomon, J.M.
Behavior of Compressed Asbestos - Fibre Gaskets in Narrow-Faced, Bolted, Flanged Joints
IME Proc., V. 171, pp. 829-833, 1957

To determine the conditions necessary to maintain tightness in narrow-faced, bolted, flanged joints. Experiments were carried out on standard 300 lb. per sq. in., 900°F screwed-on and welded-on flanges fitted to two and one-half inch bore pipes. 'Klingerit,' rubber bonded asbestos gaskets of various dimensions were tested.
Stresses in the bolts were measured with strain-gauges. Results reported.

D-057 T
Donald, M.B., Salomon, J.M.
The Behavior of Narrow-Faced, Bolted, Flanged Joints Under the Influence of Internal Pressure
IME Proc., V. 173, No. 17, pp. 459-468, Nov. 17, 1959

Experimental investigations indicate that during the application of hydraulic pressure to a narrow-faced, bolted, flanged joint; the bolt load can either increase, decrease, or remain constant, depending upon the dimensions and elastic properties of the joint components.

D-058
Dorr, G.N.
Leather Packings for Hydraulic Systems
Product Engng., V. 6, pp. 139-141, April 1935

Grades of leather and design features for various types of leather packings suitable for different ranges of pressure and conditions of service; drawings showing application of U type, split ring, multiple V and concentric ring packings.

D-059
Dorr, G.N.
Packings for Hydraulic Equipment
Mach. (N.Y.), V. 58, pp. 170-175, Sept. 1951

Packings for hydraulic presses, machine tools and other equipment are used in a wide variety of types and different materials for specific purposes. Those used most extensively are described in this article together with recommended applications.

D-060
Dorr, G.N.
Packings for Hydraulic Equipment
Mach. (Lond.), V. 80, pp. 236-239, Feb. 7, 1952

Material from which a packing is made, is also of great importance and must be given careful consideration when selecting seals for any particular purpose. Installation hints given together with operating characteristics of leather as a packing material.

D-061
Doty, W.R.
O-Ring Seals for Glass Apparatus
Review Scientific Instruments, V. 30, pp. 1053-4, Nov. 1959

To construct the joint, glass tubing of the proper size is heated, gathered, and then tooled with forming pliers. The face of the joint contains a groove of the proper diameter and depth to hold an O-ring which makes the seal. Successfully used at temperatures up to 300°C and pressures down to 10^{-6} mm Hg. Drawing, photographs, and specifications for O-ring joint.

D-062
Doughty, L.E.
Composite Elastomer-Metal O-Ring Seals
WADC Technical Report 59-749, Part II, 40 pp. 14 fig., June 1961

A revised fabrication procedure was developed to embed a metal spring concentrically in an elastomer O-ring. Hydraulic dynamic tests were performed at 4000 psi and temperatures up to 500°F. Test results and data presented and discussed.

D-063
Dreibelbis, L.L.
Pressure Traps Between Packings
PB 16610-Off. Tech. Serv., 8 pp., March 19, 1942 (Army Air Forces Expt. Eng. Memo. Rept. EXP-M-51/M752, Add 9)

Laboratory tests are reported on O-ring packings when installing more than one ring on a piston head. As a result of tests it is recommended that O-ring packings be installed with only one ring in each gland, while "V" rings should be installed back to back. Appendix contains table, performance curve, and drawings.

D-064
Dresden, D.
On the Theory of Packing Gland for Reciprocating Machines
Engrs. Digest, V. 11, pp. 342-344, Oct. 1950

Practical information concerning packings for glands can be obtained from standard handbooks. These, however, do not explain why one type of packing is preferable to another under certain conditions. In this article, an attempt is made to develop a theory of the principles involved.

D-065 B
Dunkle, H.H.
Gasket Design and Selection
Proc. 4th National Conference of Industrial Hydraulics, V. 2, pp. 61-71, 1948

Some principles to be observed and practical notes on performance of several types of gaskets are given, but little in the way of actual design.
A classification is given of suitable gasket materials for increasing temperatures, and also of common types of cross-section used.

D-066
Dunkle, H.H.
Gaskets for the Oil Industry
Oil and Gas Journal, V. 49, p. 194+, October 5, 1950, Petroleum Refiner, V. 29, pp. 135-9, November 1950, Abstract; Power, V. 94, p. 136+, November 1950

Description of basic materials used for gasket service in the oil industry; compressed asbestos sheet packing-1/16" thick; corrugated soft iron; corrugated iron jacketed asbestos; spirally wound stainless steel and asbestos; flat iron jacketed asbestos; soft iron ring or octagonal shape. Temperature and pressure ranged indicated.

D-067
Dunkle, H.H.
Metallic Gaskets
Machine Design - The Seals Book, pp. 103-112, Jan. 19, 1961

For any gasket application the choice of material will depend upon the operating conditions, the mechanical features of the flanged assembly, and the gasket characteristics. In general the operating conditions govern the choice of gasket materials and the dimensional and mechanical features of the flanged assembly control the selection of the gasket type.
A guide to be used to determine the choice of the metallic or nonmetallic gasket is covered.

D-068
Dunkle, H.H., and Fetter, E.C.
Chemical and Heat Resistance of Gasket Materials
Chemical Engineering, V. 53, pp. 102-109, Nov. 1946

Gasket constructions in general use and gasket materials both metallic and nonmetallic, their heat and chemical resistance.

D-069
Dunn, R.
Shaft Seal Development Program for the HTGR
Nuc. Sci. Abst. 28388 (Vol. 16, No. 20, p. 3713, Oct. 31, 1962), (Clark Bros. Co., Div. of Dresser Industries, Inc.)

Development of the shaft seal for the HTGR recirculator is reviewed. An interlocking labyrinth was developed which is installed on the shaft from the drive end. This type labyrinth provides a lower leakage rate than straight or stepped labyrinths for a specified shaft size.
Other design aspects are discussed.

D-070
Dunsmoor, R.J., Tipton, F.W.
Testing of Fluid Seals for Extreme Temperatures
Boeing Airplane Co., Proc. of Joint Army-Navy-Air Force Conference on Elastomer Res. & Dev., October 1958

D-071
Dyckes, G.W.
Designing Silicone O-Rings
Rubber World, V. 146, pp. 73-5, April 1962

Information on dimensional stability of molded silicone rubber items, effects of cross section diameters on shrinkage factors, and variation of these factors under normal processing conditions; 7 commercial silicone compounds were investigated, with single compounds of natural rubber, neoprene and butyl rubber for comparison, guide for establishing O-ring tolerances is proposed.

D-072
Dymkowski, J.V.
Pneumatic System Design
Aeronautical Engineering Review, V. 17, pp. 50-55, 66, March 1958

Duct connecting flanges are flat-faced machined bolted type with recessed grooves for O-rings, metallic and silastic, used for sealing. Slip, ball and slip and ball type joints.
Deals with tension and compression type duct system.

E-001 T
Eastman, F.B., McMahon, H.O.
High Temperature Seal for Rotating Shafts
Chemical Engr., V. 54, pp. 157-8, May 1947

Use of teflon shavings dusted with a lubricant such as graphite as a packing for shaft seal. Temperatures up to 700°F are permissible with this packing. Packing must be spring loaded to overcome thermo-moulding of the teflon.

E-002
Eaton, W., Hees, G.W., and Sech, J.
Knife-Edge Vacuum Seal
Vacuum, V. 4, pp. 438-44, 1954

The seal is of a demountable gasket type, made by forcing 2 knife edges into a metal gasket; various knife-edge materials were used including stainless steel, low-C steel, ceramics, quartz and glass.

E-003
Eddy, R.W.
Low Temperature Seals
Chrysler Corporation Missile Division, P 1-31, April 1958

Memorandum describing evaluation of five types of seals to qualify the most reliable for use in LOX lines of Redstone Missile. Includes Normah Coneseal, stainless O-rings, silicone O-rings. Skinner spring seals, temperature compensating coupling stress distribution.

E-004
Edwards, P.L.
Rolling O-Ring Seal
Review of Scientific Instruments, V. 31, p. 1356, Dec. 1960

A very convenient cover and air tight seal called the rolling O-ring seal, is useful where the chamber pressure is below, or only slightly above the outside pressure. Drawing and example included.

E-005 B
Edwards, T.W.
Controlled Leakage Seal; One Answer to High-Pressure Stuffing Box Problems
Power, V. 104, pp. 68-9, November 1960

Editorial article on controlled leakage seal. Solid bushing, and floating ring are the two seal types. Materials, advantages and limitations discussed.

E-006
Edwards, W.L.
Designing Soft Copper Gaskets for High Pressure Equipment
Chem. & Met. Engng., V. 44, pp. 134-137, March 1937

The design and application of soft copper gaskets: for sealing various kinds of pressure vessels.

E-007 T
Egli, A.
The Leakage of Steam Through Labyrinth Seals.
ASME Trans., V. 57, pp. 115-122, 1935

Derivation of flow rate equation from Saint Venants equation for a single sharp edge throttling process, as applicable to labyrinth seals. Graphical method outlined, method devised of connection for application to straight-through labyrinths. Test data included.

E-008 T
Egli, A.
The Leakage of Gases Through Narrow Channels
ASME Trans., V. 59, pp. A63-A67, 1937

R stance coefficient for compressible flow in the a li. Graphs of pressure ratio vs. dimensionless m. s flow. Steam and air leakage test reports, and plots.

E-009 B
Eichenburg, R.
Design Considerations for A.W.H.E.M. 15,000 PSI Flanges
ASME Paper 57-PET-23, 14 pp. inc. 4 fig., Sept. 1957, Mechanical Engineering - V. 80, 3, pp. 66-8 (Condensed), March 1958

This paper provides the necessary information for design of high pressure flanges and gaskets as used by the Standardization Committee of the Association of Well Head Equipment Manufacturers in the design of their 15,000 psi flanges. The design is useful in the extension of the present 10,000 psi API group of flanges to larger size, as required.

E-010
Eiseman, B.J., Jr.
Effects On Elastomers of Freons and Other Halohydrocarbons
Refrig. Engng., V. 57, pp. 1171-1174, 1179, Dec. 1949

A basis for estimating the swelling and elasticity characteristics of gasketing materials.

E-011
Eller, S.A.
Stress Relaxation Versus Seal Ability of MIL-R-900 Gasket Materials
Rubber Age, V. 79, pp. 455-458, No. 3, June 1956

Relationship between stress relaxation and seal aging properties of buna-S stocks, S-3-11 and E-162-1488, compounded to requirements for class 2 material of specification MIL-R-900 and neoprene W stock E-156-575 and buna-N stock E-194-542, compounded to requirements for class 1 and class 5 material, respectively, of specification JAN-R-1149.

E-012 B
Eller, S.A.
An Apparatus for Preparing O-Ring Tensile Specimens
Rubber Age, (NY), V. 84, 5, p. 812, Feb. 1959

E-013
Eller, S.A., and Stein, A.A.
O-Rings for Cylinder Liners of Diesel Engines
Rubber Age, V. 85, pp. 443-448, June 1959

Evaluation of neoprene, nitrile, and silicone, rubber O-rings under conditions simulating service such as in engines aboard naval vessels; load retention properties were higher for nitrile and silicone rubbers than for neoprene; standby service has greater deleterious effect on load retention of neoprene and nitrile rubber O-rings than active service, while for silicone there is little difference.

E-014
Ellis, E.G.
Sealing Problems of High Speed, High Altitude Aircraft and Guided Missiles
Scientific Lubrication, V. 11, pp. 10-12, 14, May 1959

Hydraulic fluids to be sealed are hydrocarbon fluids to 275F, diester blends and silicate esters to 450F, and modified silicones to 700F.
For static seals, O-ring design studies are presented with the following materials: synthetic elastomers, fluoroalkyl silicones; Viton A and Viton A-HV.
On dynamic seals for pneumatic and hydraulic circuits, elastomers may be used up to 120 seconds, but metallic seals must be used for extended running.
Both static and dynamic seals for propulsion systems with high energy fuels, mono-propellants and cryogenic fluids are briefly discussed.

E-015
Elonka, S.M.
Fuel Oil Pumps and Packing
Marine Engineering, V. 44, p. 434, Sept. 1939

Brass fitted pumps should not be used for pumping fuel oil because some fuel oils contain a sulphur base, others an asphalt base. Reasons, corrections, discussions are presented. Operational evidence presented.

E-016 T
Elonka, S.
Fix Seals Before They Knock Out Turbine
Power, V. 95, pp. 112-3, August 1951

Brief commentary on, a) how glands work, b) gland trouble and cures. Maintenance procedure.

E-017 B
Elonka, S.
Manual on Piston Rings
Power, V. 96, 7, pp. 103-26, July 1952

What rings do, guide to ring types, controlling cylinder tube, making emergency rings, liner and piston ring wear, installing and renewing rings.

E-018
Elonka, S.
Gaskets
Power, V. 98, pp. 105-124, March 1954

Author presents a round-up of the latest information on sealing materials and methods, time proven ways as well as new ones.

E-019
Elonka, S.
Mechanical Packing
Power, V. 99, pp. 107-130, March 1955

Article points out that packing is not an exact science since there are too many variables. Experience of engineers and specialists of packing makers, users of packing, and research laboratories where packings have been tested for over thirty years, for packing know how.

E-020
Elonka, S.
Mechanical Seals
Power, V. 100, pp. 109-132, March 1956

Mechanical seals. How they work, materials, finish, cooling systems, selection, installation and care.

E-021
Elonka, S.
This Mechanical Seal Comes Ready to Slide Over Shaft
Power, V. 100, pp. 120-121, No. 12, Dec. 1956

Seal comes fully assembled for direct installation on pump shaft; device known as unitary design, is for pressure to 100 psi, temperatures to 250F and shaft speeds of 4000 fpm. Solves problem of seal faces being damaged by inexperienced help when installing mechanical seal into machine.

E-022 T
Elonka, S.
Not Springs, but Magnets Hold This Mechanical Seal Together
Power, V. 102, pp. 120-1, Dec. 1958

Description of how the seal works. Allows more even seal face pressure so wear on faces is more accurately controlled. Seals used up to 5000 psi, 550°F, speeds up to 20,000 f.p.m. Magnetic shield for gland described.

E-023 T
Elonka, S.
This Mechanical Seal is Also a Pump
Power, V. 103, pp. 220-221, Sept. 1959

Editorial comments on keeping seals cool.

E-024 B
Elonka, S.
Metallic O-Rings Have Come of Age
Power, V. 103, 12, pp. 200-1, Dec. 1959

These rings seal a variety of chemicals, and gases, i.e., liquid sodium at 1300°F. 0.006" - 0.010" thick resistance welded tube rings. Stainless steel, 304, 316, 321, 347, Monel, Inconel, Inconel X, may be coated with silver, Kel F, or teflon. Basic types, pressures, flange loading. Experimental results.

E-025 B
Elonka, S.
Handbook of Mechanical Packings and Gasket Materials
Mechanical Packing Association, 110 pp., 1960

E-026
Elonka, S.
What's New in Seals
Power, V. 106, 189-93, May 1962

Gaskets: Molded-in-place and pressure-lock types are new.
Fluorocarbon resins team up with inorganic fibers.
Mechanical packings: Rod seal adjust itself; wiper-scraper design is complete in one unit.
Mechanical Seals: Fluid-balanced and welded metal bellows types widen sealing ranges.
Piston rings: Teflon is filled with carbon or glass.
Nylon wear rings also work without lubrication.
O-rings and Lip Seals: Fluoroelastomers, butyl and polyurethanes give rings new temperature scope.
Diagrams, drawings - etc.
Cork and cork rubber, cork and silicone rubber, teflon and asbestos, etc.

E-027
Elsworth, L., Holland, L., and Laurenson, L.
Further Experiences With Aluminum Wire Seals for Bakeable Vacuum Systems
Journal of Scientific Instruments, V. 37, pp. 449-451, Dec. 1960

Aluminum wire clamped between stainless-steel flanges and baked at 300°C formed more adherent joints as the surface finish was decreased from 50 to 10μ in. The Al joint would not cold weld with a force of 4 tons per in. of 20 SWG wire using flanges with a surface finish of 10μ in. and the flanges were indented by the wire. With constant surface finishes, the strength of the Al joint was directly proportional to the baking temperature above a forming temperature (250°C at 10μ in.), and was greatest when the Al wire was perpendicular to the direction of machining marks. Seals made between flanges with a 10μ in. finish had an adhesion strength of 354 lb/in. of wire after forming with a force of 5000 lb per in. of wire and baking at 400°C.

E-028 B
Engelking, F.S., Keys, M.C.
Resilient Face Seals for Tractor Final Drives
S.A.E. Transaction 66, pp. 75-80, 1958

Reports on efficient performance of a tractor final-drive seal, dependent on bellows, bellows-boot operation, seal load and area, seal material, wear washer, and gasket structure.
Curves and test data presented.

E-029
Engle, R.W.
Leaktight Seals for High-Pressure Valves and Piping
Product Engineering, V. 32, pp. 72-75, April 24, 1961

Seal choices for pressures above 1500 psi, and at a temperature range of -20 to 275F.

E-030
Engstrand, W.D.
High Pressure Sealing Fasteners
Assembly and Fastener Engineering, V. 4, pp. 35-37, Feb. 1961

A fastener designed to seal against the leakage of liquid or gaseous media or the transfer of pressure differentials.

E-031
Evans, E.C., Babelay, E.F.
Removable Bowl Cold Trap
Rev. Sci. Instrum., V. 23, pp. 249-250, May 1952

The bowl of the trap has a ground flange at the open end. This flange fits a metal flange of the system's tubulations which retains a neoprene or gum rubber gasket and is sealed to the system by action of the atmosphere. In order to improve vacuum tightness it is recommended to coat the gasket with Apiezon grease.

E-032 T
Evans, R.G.N.
Methods of Sealing Sleeve Bearings
Product Eng., V. 8, No. 3, pp. 110-1, March 1937

Twelve drawings on seals as applied to plain or sleeve type bearings for retaining lubricant or excluding foreign material.
Presentation covers leather packings, bronze rings, "V" formed spreader ring, and labyrinth seals. No technical data.

E-033
Evans, W.P., Landen, E.W.
Radioisotopes Can Do the Otherwise-Impossible; Gasket and Seal Evaluation
SAE Journal, V. 67, 74C-74D, Aug. 1959

A radioactive tracer, cesium 134, was added to the coolant, for an accurate check for leakage into the crankcase, or other leakage paths. Engine parts suspected of allowing leaks could then be autoradiographed on X-Ray film after engine disassembly. Data results, photographs, and curves presented.

E-034
Everett, M.H., and Gillette, H.G.
Squeeze Type Molded Packings
Machine Design, The Seals Book, pp. 68-76, Jan. 19, 1961

Most common type, the O-ring. They work under the principle of controlled deformation. Pressure from the combined fluid produces the deformation which causes the elastic O-ring to seal.
Applications include reciprocating, oscillating and rotating devices.
Operating range -65° to + 225°F for polysulfide polymer and -130°F or lower and as high as 500°F for silicone polymers.
Hexafluoropropylene and vinyldene fluoride copolymers are also used. They show excellent resistance to a great number of fluids and chemicals, -40°F to 600°F.

E-035
Everett, M.H. and Gillette, H.G.
Static O-Ring Seals
Machine Design, The Seals Book, pp. 100-102, Jan. 19, 1961

All static O-ring seals are classified as gasket type seals. Easier to apply than dynamic O-ring seals. Pressures as high as 25,000 psi can be sealed. Once the O-ring is installed, the operating pressure keeps the O-ring sealed against the metal clearance areas. Some types of silicone materials will function within the -130 to +500°F range.

E-036
Exline, P.G.
Leakage in Capillary Seals of Hydraulic Valves and Pumps
Prod. Engng., V. 17, pp. 290-296, April 1946

Formulas for calculating the volume of leakage of viscous fluids through concentric and eccentric radial clearance spaces are given.

F-001
Faber, P.
Designing Labyrinth Stuffing Boxes For Steam Turbines
Power, Sept. 8, 1908

Gives a method for calculating packings of labyrinth stuffing boxes, graphical solution.

F-002
Fabian, R. J.
Materials For Gaskets, Packings And Seals
Mat. in Design Engrg., V. 50, pp. 111-126, Dec., 1959

Hundreds of materials and composites are available for sealing fluids under static or dynamic conditions.

F-003
Fairbanks, F. L.
Packing For Ammonia Compressor Stuffing Boxes
Power, V. 43, pp. 179-81, Feb. 8, 1916

Various kinds of ammonia packing are illustrated and described. An asbestos fabric and rubber, self-centering ring is a sensitive and excellent packing. A fibrous sectional wedge ring with cushion gives good service on the rod of a pump discharging 200 gal. per min. of anhydrous ammonia, 24 hours per day, with no odor of ammonia at the stuffing box or from the rod.

F-004
Faley, R.L., Long, J. F.
High Temperature Seal For Gas Chromatography Detectors
Analytical Chemistry, V. 32, p. 302, Feb., 1960

Describes the modification of the Crawford Swagelok tube fittings to mount a gas chromatography hot wire detector. Temp. up to 500 C, and pressure to 50 psig

F-005 T
Fanestiel, A.
Mechanical Shaft Seals For General Refinery Service
Oil and Gas Journal, V. 49, pp. 56-61, Feb. 1, 1951

Discussion of application of three seal types to refinery service: stuffing box with cage ring, double mechanical face seal with seal oil between, and single mechanical face seal.

F-006 B
Farman, R. G.
Studies Of Relaxation Characteristics Of Non-metallic Gasket Materials
India Rubber World, V. 123, 5, pp. 679-82, & 685, March, 1951

Describes a relatively simple apparatus designed for determining the relaxation characteristics of gasket materials at room and elevated temperatures to stimulate actual field conditions.
Data curves presented.

F-007
Farnsworth, S. H.
Metallic Packing For Valve Stems
Power, V. 41, 648, May 11, 1915

A modification of packing on Corliss type valve stem.
Drawings for two types

F-008
Faunce, S.F.
Metallic Seal And Gland For Static Sealing Applications In Aircraft Power Plant Fluid Systems
WADC Tech. Rpt. 56-378, June, 1956

F-009 T
Fawcett, J.R.
Fluid Seals For Rotating Shafts
Mech. World Engr. Record, V. 130, No. 3369, pp. 121-5, Aug. 10, 1951

The choice of seals to prevent lubricant from escaping along a rotating shaft depends on the nature of fluid, its pressure and temperature, shaft speed, and permissable leakage. Discussions of seal arrangements, to give protection in two directions and synthetic rubber oil seals with reinforcing washer to withstand higher pressures are included.

F-010 T
Fawcett, J.R.
O-Rings And Their Application
Machinery (London) V. 82, No. 2108, pp. 677-8, April, 1953

Article discussing efficiency and requirements of effective sealing up to 40,000 lbs. per sq. in. pressures. Ratios for size, hardness and housing finishes discussed.

F-011 B
Fawcett, J.R.
Pressure Seals For Reciprocating Parts
Hyd. Pur. Transm. Part I - V. 1, 2, pp. 97-100 - Feb., 1955
Part II - V. 1, 3, pp. 154-57 - Mar., 1955
Rotating Seals Part III — V. 1, 5, pp. 293-5, May, 1955
Part IV - V. 1, 6, pp. 249-52, June, 1955

F-012
Fawcett, J.R.
Pressure Sealing By O-Rings
Engineering, V. 185 pp. 314-315, March 7, 1958

Static O-ring applications are illustrated with pressures to 3500 lb. per sq. inch.
Dynamic applications are also discussed together with their limitations.

F-013
Feather, L.E.
Gasketed Joints For Electrical Equipment
Product Engineering, V. 28, pp. 183-188, March, 1957

Requirements imposed on gaskets and gasketed joints for all types of electrical equipment are so diverse that no single gasket material can be used universally; in fact, the ideal material is seldom found for any application.
Choice of material and joint design involves four basic criteria; these togehter with specific gasket materials are discussed.
Application suggestions are likewise covered.

F-014
Fedor, J.V.
A Sommerfeld Solution For Finite Bearings With Circumferential Grooves
A.S.M.E. Paper No. 59 - Lub - 4, 6 pages

The author develops a method of solution of the Reynolds differential equation for full journal bearings that circumvents the usual algebraic complexity. The bearing considered by the author is similar to a buffered bushing type seal. Calculation of bearing end flow.

F-015
Fekete, K.
Similarity Considerations For Radial Face Seals
Escher Wyss News, V. 34, No. 2/3, pp. 46, 47, 1961

The characteristic non-dimensional parameters or dynamic similarity criteria of the radial face seals can be determined by dimensional analysis. Formula development, schematic drawings, test results, data curve are presented.

F-016
Felter, L. E.
Babbitt Packing For Steam Joints
Popular Mechanics, V. 34, 448-9, Sept., 1920

F-017
Fenner, A. J., Willoughby, G.
Gas Seal For High Temperature Extensometer Tensile Tests In Inert Atmospheres
Journal Scientific Instruments, V. 31, pp. 472, 3, Dec., 1954

A silicone rubber moulding closes the lower end of a tube, and stretched over the lower straining bar of the machine. The extensometer arms are then passed one by one through the appropriate rectangular apertures and attached to the ends of the test piece. 400°C for 100 hours, 600°C for 70 hours. Discussion, data curve and photo.

F-018
Field, H., Jr.
Design Of Hydraulic Systems, Packings And Their Application
Product Engineering, V. 16, pp. 100-4, February, 1945

The problem of selecting a fluid seal or packing is is discussed as to materials, types available, purpose of the seal, installation problems, and advantages and disadvantages of each type of packing. Compression packing, U-cup packing, V-ring packings, O-ring packings, high pressure packings. Asbestos, yarn, cotton yarn, hemp, cork, leather, rubber, lead, tin, graphite, etc. used separately and combinations. Drawings and detailed descriptions.

F-019
Finkel, J.
Schwabe Stuffing-box
Zeitschi. Vereines Deutsch Ing., V. 47, pp. 1049 - 1051, July 18, 1903 (communicated at the meeting of Westfälisches Bezirksverein Nov. 18, 1902)

In this stuffing box, specially designed for high pressures, and super heated steam at 320°C, the packing rings are made of cast iron owing to the ease with which this acquires a fine glass-hard polish. Five or more such rings, split at three places on the periphery, are carried enclosed between outer cast-iron rings in the chambers bored out for them, play in all directions in the plane of the rings being obtainable. A device is shown by which the leakage past such a stuffing box is measured on a high-pressure cylinder. The mean temperature of the interior of the cylinder was 225°C. Diagrams show the pressure and temperature distribution in the stuffing box.

F-020 T
Fisher, E. W.
Considerations In The Development Of Roll Neck Oil Seals
Iron & Steel Engr., V. 27, 5, pp. 92-5, 96-8, May, 1950

A brief review of some of the failures and limitations of lip garter seals, and considerations for improvement.

F-021
Fisher, E. W.
Seals And Closures
Lubrication Engrg., V. 9, pp. 190-194, Aug., 1953

Seals and closures retain lubricants in bearings. Two general classifications (a) contact and (b) non-contact. Examples are given of each type and advantages and disadvantages of each are listed together with design considerations. New synthetic materials listed are silicone rubber and teflon, advantages and disadvantages of each are given.

F-022
Fisher, E. W.
Packing Construction Is Usually Tailored To The Specific Application
The Plant, V. 15, pp. 57-59, May, 1957

Braided packings for shaft seals; their construction, materials and applications. Types discussed are fiber, including jute, cotton, ramie, rayon and flax; asbestos; metal braids; teflon plastic; asbestos plus teflon; fiber plus rubber; metal foil wrapped around fibrous or non-metallic cores; "plastic" packing, called plastic because it is formable and not because of its material; and ring packings.

F-023
Fisher, E.W.
Take Care Of Oil Seals For Longer Bearing Life
Iron Age, V. 181, pp. 110-111, March 13, 1958

Recommendations on the handling of oil seals prior to installation are made together with general installation and maintenance instructions.

F-024 T
Fisher, M.J.
An Analysis Of The Deformation Of The Balanced Ring In High Pressure Radial Face Seals
Intl. Conf. on Fluid Sealing, Paper D4, April 17-19, 1961
British Hydromechanic Research Association, Harlow, Essex, England

The deformation has been determined by applying cylindrical shell theory, and for a particular seal it is comparable to that of the stationary carbon face. It is shown that this deformation depends markedly on the position of the static seal between the ring and the shaft.

F-025
Fitch, L.E.
Development Test On Hydro-Pneumatic Equilibrators For 155 MM Gun Carriage M1,(2nd Report, 30th Report On Ordnance Program No. 5084)
P.B. 33947, Off. Tech. Serv., 47 p. Oct., 1943

Tests were made to determine the effect of sub-zero temperatures on the functioning of the complete hydro-pneumatic equilibrators and the effect of the loss on hand wheel efforts. A comparison of hydro pneumatic equilibrators having standard silver packings, with pneumatic equilibrators having experimental neoprene packing, indicated that the latter are superior both in design and performance. Drawings and photographs are included.

F-026
Fitts, J.S.
Testing Of 304 Stainless Steel Metal Boss Seals
General Electric Co., Aircraft Nuclear Propulsion Dept., Cincinnati, Ohio
DC60-2-112, Contract AT(11-1)171, Feb. 9, 1960, 18P

A series of tests were performed to determine the sealing effectiveness and reusability of MS24382 304SS metal boss seals. Operating pressures up to 3000 psig and at -65 to 600°F.

F-027 B
Fletcher, P. J.
Assessing The Performance Of U Rings Used As Piston Seals In Hydraulic Pit Equipment
Hydraulic Power Transmission, V. 3, 33 pp. 534-9, Sept. 1957

F-028
Flick, F.
Hydraulic Seals
Machine Design, V. 20, pp. 139-44, Nov., 1948

Factors to consider in selecting sliding types of packing for hydraulic cylinders of types used in machine tools; development of leakproof seal assemblies; discussion of seal types, efficiency, cause of chatter, advantages of cup seals, metallic seals, piston rod seals and static seals.

F-029
Fomesi, R.
Mechanical Seals For Slurry Service
Industrial & Engrg. Chemistry, V. 51, pp. 59A-60A, Nov. 1959

A slurry is a mixture of a liquid and undissolved solids, either amorphous or crystalline. Amorphous solids are usually not abrasive but will gum up seal springs. Springs must be located out of contact with such slurries. Crystalline solids are usually abrasive. Three mechanical dynamic seal configurations are presented for abrasive slurries: (1) a single seal with a clear liquid injected at a pressure slightly above the pressure on the stuffing box, thus isolating the seal from the abrasive pumpage; (2) double seals, one at each end of the stuffing box with an outside source of liquid to lubricate the seal faces from within the seal cavity; (3) single seal with a quench style gland for use only with slurries where abrasive conditions are caused by contact of the liquid with the atmosphere.

F-030 T
Forrest, J.
Oil Seals For Rotary Shafts
Trans. of the Institute of Rubber Industry, V. 20, pp. 212-214, 1945

This paper records advances made in the use of molded synthetic rubber sealing members.

F-031
Forrester, P.G.
Frictional Properties Of Oil Seal Materials
Engrg., V. 164, pp. 121-124, Aug. 8, 1947

Tests were made to determine amount of friction between an oil-seal lip and the rotating land for restricted areas of contact and low sliding velocities. The oil seal materials used were four types of leather and two types of synthetic rubber and three lubricants of standard service grade.
Experimental results are shown in table form.

F-032
Forrieres, R.
Deterioration Of Rubber Gaskets
Gas Journal (London), V. 207, p. 362, Aug. 15, 1934

F-033
Forte, T., Harrington, D.B
Demountable Metal Vacuum Seal
Rev. Sci. Instr., V. 28, pp. 585-6, July, 1957

Flanges are serrated on their inner surface, and are mechanically bonded to the pipe by expanding the pipe into the flange. A small length of pipe is allowed to protrude beyond the flange, and the diameter of this short length is fixed accurately during the simple expanding process by using a retainer plate. The seal is made by pinching a 0.20" flat ring of 2S aluminum between the turned ends of the pipe.
Rubber, teflon, copper, silver, or gold may be used.

F-034
Fox, Z.
Seals For Preventing Oil Leakage In High Speed Superchargers
Prod. Engrg., V. 17, pp. 158-162, Feb. 1946

Methods of preventing oil leakage through openings for shafts in gear cases and bearing housings of high speed superchargers where pressure differentials exist across the seal.

Design features and performances of ring, labyrinth and contact types of seals are discussed and their relative advantages compared.

F-035-
Frazier, E. C.
Design For Economical Gasketing
Prod. Engng., V. 22, pp. 139-144, July, 1951

The design, selection of material, and manner of specifying gaskets has a significant effect on joint cost and effectiveness. Factors involved are charted and illustrated to further the discussion.

F-036
Frazier, E. C.
New Paper Materials Find Use In Gaskets, Oil Filters, And Embossed Sheeting
Mat. & Methods, V. 38, pp. 92-93, Aug., 1953

A new family of paper materials composed of fibers, synthetic rubber or resins, and sometimes ground cork, have been developed by Armstrong Cork.

F-037
Frazier, E. C.
How To Select, Specify And Use Nonmetallic Gaskets
Mach. Design, V. 26, pp. 158-188, Nov., 1954

Manual deals specifically with gaskets. Its purpose is to give specific design guidance in their selection and use. Fundamental principles of gasketing are stressed, and significant characteristics of the numerous available gasket materials outlined. Materials covered are principally those having a broad field of use, as contrasted to specific purpose materials.

F-038
Freedman, R. A.
What You Should Know About O-Rings For Air And Oil Circuits
App. Hydraulics & Pneumatics, V. 12, pp. 128-130, July, 1959

From maintenance standpoint, O-rings involve problems of hydraulic leakage and pneumatic lubrication; maintenance inspection of seals revolves around two questions; how many drops of fluid must seep past hydraulic O-ring before they indicate objectionable leak and how must air seep past pneumatic O-ring before there is indication of damage; to understand hydraulic problem of leakage and pneumatic problem of lubrication, it is necessary to consider special features of O-rings separately.

F-039 B
Freeman, A. R.
Gaskets For High Pressure Vessels
Mechanical Engineering, V. 74, 12, pp. 969-72, Dec., 1952

The gaskets described in this paper are suitable for pressures of the order of 50,000 lb/sq. in. The most successful seals ensure that the contact pressure is increased as the fluid pressure increases. Rubber O-rings, delta rings, and other gaskets also described.

F-040
Freeman, A. R.
Eight Gaskets For High Static Pressure
Product Engrng. Annual Handbook, J. 30- J. 33, 1955

Methods of sealing the heads of vessels subjected to high static internal fluid pressure are described.

F-041
Premlin, J. H.
A Quick Release Seal For Vacuum Desiccators
J. Sci. Instrum., V. 29, p. 236, July 1952

Many all-glass vacuum desiccators are sealed simply by greasing the surface of the lid and the body. After evacuation, and in cold weater, it is often difficult to remove a lid sealed in this manner. The use of a rubber ring is suggested stuck to one of the surfaces between the lid and body. The rubber, compressed during sealing, expands slightly after release of the vacuum, and consequently the lid can be lifted away freely. Rubber of 40 Shore hardness should be used.

F-042
Premlin, J. H.
A Flexible Vacuum Joint
J. Sci. Instrum., V. 29, p. 267, Aug. 1952

Design developed which provides for a flange on the main vacuum system, and a flange on the moving shaft, joined by a length of rubber tubing, which is supported internally by a helical wire spring. The rubber tube is clamped to the flanges by brass bands. A floating ring in the tube prevents it from collapsing. Butyl rubber with a leak rate of 0.05 microliters per sec were used. The diameter of the hub should be 10% smaller than the system, and shrunk on to the flanges.

F-043
French, E. P., Newton, A.
A High Vacuum Rotary Seal
Nucleonics, V. 10, p. 66, Mar. 1952

The seal consisted essentially of a rubber diaphragm clamped between the upper and lower half of the housing, the center of which allowed a wobble shaft connecting the driving shaft on the atmospheric side with a driven shaft on the vacuum side to pass. The seal was effective down to a pressure of 3×10^{-5} mm Hg.
Drawing included.

F-044
Froede, W. G.
Seals Among NSU Developments On Rotating Combustion Engines
S.A.E. Journal, V. 69, pp. 50-56, April, 1961

Reports progress on seals for Wankel engines, as well as on spark plugs cooling, fuel consumption and overall durability. Drawings, data curves, pictures.

F-045 T
Prolich, M.
Turbine For Cooling A Fluid By Expansion
U.S. Patent, 2,910,328, October 27, 1959

Design provides a labyrinth which can be supplied with gas from an accumulator upon stopping the turbine. This prevents hot oil vapor from leaking from the bearing cavity into the turbine.

F-046
Prossel, W.
Hydrodynamic Shaft Seals
Engrs. Digest, V. 18, pp. 18-20, Jan., 1957

Describes a threaded shaft seal providing sealing by dynamic means, without wear and with minimum friction, as there is no direct contact between the stationary and rotating parts. Although primarily developed for liquids, this seal, fitted in a sleeve is also suitable for gases. Tests were run at speeds of 100 to 2000 rpm. Seals only when equipment is running.

F-047 T
Prossel, W.
High Speed Lubricating Oil Pump (in German)
Konstruktion, V. 12, No. 5, pp. 195-203, 1960

Determination of the effect of thread form on the viscosity type pump. The application was for lubricating oil pumps for high speed machinery. Tested V thread, trapezoidal thread, truncated threads, each in three sizes, 0.03 to 0.5 mm, clearance at 1500 to 15,000 rpm, in 50 mm dia. pump. Surface speeds [illegible] 3.93 to 39.3 mps, oil viscosity 20 cs at 20°C. Test results shown.

F-048
Fujiwara, E. J. and others
Compounds For High Temperature Fuel Seals
Rubber Age, V. 82, pp. 1016-1020, No. 6, March, 1958

Apparatus for determination of physical properties of elastomeric materials at temperatures to 400°F. While immersed in aircraft fuels; machine incorporates fixed metal bellows filled with hydraulic fluid as unit in which pressure will reflect stress applied to O-ring; of materials tested, Viton A vulcanizates appeared to be of most interest for high temperature seals.

F-049 T
Fuller, D. D. (Editor)
Gas Lubricated Bearings
Proc. First International Symposium on Gas-lubricated Bearings, Office of Naval Research, ACR-49, Oct. 26-28, 1959

Twenty papers on gas-lubricated bearings. The appendix B is a revised version of "A Bibliography on Gas-lubricated Bearings" by E. B. Sciulli.

F-050 B
Furuhama, S.
A Dynamic Theory Of Piston Ring Lubrication
Bulletin of Japan Society of Mechanical Engineers
Part I - V. 2, 7, pp. 423-8, 1958 (Analysis)
Part II - V. 3, 10, pp. 291-7, May, 1960 (Experiment)

It has been found that at high engine speeds the oil film between a piston ring and the cylinder wall is not ruptured at T.D.C. when the piston speed is zero.

F-051
Fuschillo, N.
Sliding And Rotating Mechanical And Electrical Feed-through Seals Into A Vacuum
Am. J. Phys., V. 26, p. 400, Sept., 1958

Low cost rubber seals for use in vacuum systems at pressures down to 10^{-6} MM Hg are described. The seals permit almost unlimited rotational and longitudinal freedom as well as provide a reliable and easily demountable feed-through system for electrical connections.

G-001 B
Gahus, J.
Centrifugal Pumps In The Chemical Industry
Chimie et Inc., V. 63, 1, pp. 34-9, Jan. 1950 (In French)

This article is a discussion of packings for centrifugal pumps used in the chemical industry and the author shows that from a common sense point of view as well as from experience pump stuffing boxes should be allowed to leak. Use of the radial-face seal (mechanical) is advised in the most difficult cases where leakage is unacceptable.

G-002 B
Galloway, C.W.
Evaluation Of High Temperature Hydraulic Seals To Temperatures Of 550°F - Part I Mechanical Evaluation
W.A.D.C. Tech. Reports 57-76, 143 pp., Feb. 1958

Report describes test to evaluate and develop seals for use in hydraulic systems using OS-45 or MLO-8200 fluids at temperatures of 400°F and 550°F respectively. The scope of the program includes seals of various shapes and forms as well as their method of installation and the material from which they were manufactured.

G-003
Galloway, C.W.
Designing A Seal For The DC-8
Hydraulics and Pneumatics, V. 14, pp. 89-91, Feb. 1961

Description of a teflon slipper seal with a square rubber loading ring, developed to prevent leakage of hydraulic fluid in the DC-8 landing gear.

G-004
Galpern, H.N.
Aircraft Gasket Compounds
Rubber Age, V. 90, No. 4, pp. 621-626, Jan. 1962

An evaluation of the compatibility of a fluorocarbon elastomer with various silicone lubricants under a wide range of temperature conditions.

G-005
Garba, J.
Preliminary Evaluation Of Helium Couplings For The 6K Motor
Jet Propulsion Laboratory, P 1-3, April 21, 1959

Seals, O-ring design, and test of special O-ring tube joint.

G-006
Gardner, A.R.
Beat The Heat With Pyrolytic Graphite
Prod. Engng., V. 33, pp. 72-75, Jan. 22, 1962

High temperature strength and two kinds of electrical and thermal conductivity make this a material to watch.
Chart compares its properties to tungsten, copper, stainless steel (310) and commercial graphite.

G-007
Gardner, D.B.
Recirculating Water Systems And Mechanical Pump Seals
Air Conditioning, Heating & Ventilating, V. 57, pp. 57-9, Dec. 1960

The performance of mechanical pump seals can be controlled by certain application and installation procedures which can be utilized in both closed and open recirculating water systems. Seal failure cause and prevention. Photograph examples of seal failures. Carbon and ceramic mating rings.

G-008
Gardner, W.W., and others
Hydrostatic Thrust-Type Shaft Seal For Hydrogen-Cooled Generators
Power Apparatus and Systems (AIEE)Trans.), V. 78, pp. 653-659, Aug. 1959

The seal is essentially a hydrostatically lubricated thrust bearing in which the thrust load to be supported by the oil film consists of the spring force, the force produced by the hydrogen pressure acting on the end of the seal ring, and the O-ring and support-pad friction forces resulting from axial movement of the shaft. These forces are balanced by the oil film pressure distribution maintained by the oil-supply system at the face of the seal. The seal has a babbitted face. It is suitable for hydrogen pressures above 30 psi and up to 75 psi.

G-009
Garner, L.P.
Machined Metal Stuffing Box Seals Adapted To High Vacuum Technique
Rev. Sci. Instruments, V. 8, pp. 329-332, Sept. 1937

Vacuum seals machined from metals and adapted to vacuum technique are described and shown in several forms suitable for joining metals and glass components. Precautions in construction and use are given.
Dimensioned working drawings for several sizes of one type are shown.

G-010
Garrod, R.I.
A Compact Sliding Vacuum Seal
J. Sci. Instrum., V. 28, p. 187, June 1951

The author describes an O-ring type of joint allowing for translatory movement. The flange of one of the two pipe sections is fitted with a V-shaped groove to accommodate the O-ring. The flange of the adjoining pipe section is recessed over a certain width of the flange, to a depth that 7 percent compression permits the two flanges to touch.

G-011
Gartmann, H.
Boiler Feed Pumps For Supercritical Pressure
Mech. Engng., V. 80, pp. 51-54, Jan. 1958

Seals for a high pressure pump in a 125,000 kw steam plant designed for a supercritical steam pressure of 4500 psig are discussed.
The sealing of the high pressure pump, with its suction of 2065 psig and 510.9°F temperature, required careful consideration.
The seal bushings selected were of the floating type with a close tolerance between the shaft and bushing, and a large clearance between the bushing OD and the surrounding housing. One end was sealed by a lapped radial face against a matched face in the case.
Satisfactory results are obtained when a fluid having good lubricating properties is used as a sealing medium.

G-012
Garvin, J.M.
Designing A Seal For Very High Pressure
Applied Hydraulics, V. 10, pp. 141-145, May 1957

To seal 45,000 psi. Careful consideration must be given to every detail of design and construction. Starting with the basic principles of seals and weighing every step on the way.
Step-by-step description of how such a problem was solved, with insight into some of the reasons why packings work or fail.

G-013
Gastineau, R.L.
Hollow Metallic O-Rings
Machine Design, The Seals Book, pp. 113-114, Jan. 19, 1961

Hollow metallic O-ring gaskets possess certain characteristics under operating conditions that static rubber O-rings do not. Under very high pressures, clearances for rubber O-rings must be prohibitively small to prevent rubber extrusions. Temperatures are limited to temperatures below the decomposition temperature of the rubber. At the other end of the scale, under a high vacuum, rubbers give off gases of their own, thereby destroying the vacuum.

G-014
Gel'man, L.J.
Centrifugal Pump For Amalgam
Order from OTS or S.L.A. - Translation No. 61-27179

The construction of shaft packing for centrifugal pump was made for the purpose of eliminating contact between the amalgam and atmosphere's air and test was made to determine the efficiency of pump shaft packing.
Experimental data.

G-015
Gerberich, D.E., Kurlovich, D., and others
Experimental Investigation Of Optimum Inclosure For "O" Section Synthetic Rubber Seal Ring
WADC Tech. Report 53-319, Aug. 1953

G-016 T
Gercke, M.J.
Flow Through Labyrinth Packing
Die Warme, V. 57, pp. 513-517, 1934
Abstract-Mech. Engn., V. 56, pp. 678-680, 1934

A flow equation is derived by considering a series of nozzles. The final equation does not assume constant flow area, but if constant area is assumed, the result is identical to Martins equation. The use of the equation under different operating conditions is discussed, and the influence of a direction of flow with radially arranged labyrinths is considered.

G-017
Gercke, M.J.
Labyrinth Packings and Centrifugal Force (In German)
Waerme, V. 58, pp. 529-34, Aug. 17, 1935

Analysis of effects of centrifugal force on flow-through packings as basis for quantitative determination of effect in any particular case, taking steam pressures into consideration, special reference to steam-turbine packings.

G-018
Giaimo, E.C., Jr.
Ring-Type Teflon Gasket
Review of Scientific Instruments, V. 26, p. 520, May 1955

The gasket consists of a teflon ring, machined from the edge resulting in a "U" shaped cross section. A toroidal helical spring is loosely laid in the groove around the outside of the ring. After the initial loading has taken place, the spring becomes partially imbedded in the teflon, which prevents it from moving. The sealing action is accomplished by producing a radial force as a result of an axial one. Design data and service results presented, as well as drawing.

G-019
Gilchrist, R.R.
Application Of Cold Drawn Seamless Steel Tubes For Hydraulics
Compressed Air & Hydraulics, V. 23, pp. 159-163, No. 263, Feb. 1958

Discussion of seamless hydraulic ram cylinder manufacturing methods, surface finishing techniques, seal types, performance, and operating characteristics; dependence of operating cycles, on surface finish, and effect on seal failures; recommended surface finish tolerance for medium and heavy duty cylinders.

G-020
Gilhart, J.S.
Graphite-To-Metal Seal Screening Tests
Babcock and Wilcox Co., Atomic Energy Div., Lynchburg, Va., BAW-1053, Contract AT(30-1), 1940, 21 pp., Sept. 1958

A seal which will hold bismuth with a practical interface contact pressure was developed between graphite and steel.

G-021
Gill, J.S., Marshall, W.L.
High Pressure Vessel Incorporating A Teflon Gasket For Use To 500°C And 1000 ATM
Rev. Sci. Instrum., V. 32, No. 9, 1060-2, Sept. 1961

A small vessel is described, believed to be leakproof at a temperature of 500°C and at the hydrodynamic pressures up to 1000 atm. Details of the design are given and the use of the vessel in the determination of solubilities in aqueous systems is described.

G-022
Gillespie, L.H.
How To Design With Teflon Packings
Applied Hydraulics and Pneumatics, V. 12, pp. 121-124, July 1959

Advantages of teflon for seals are; its almost complete chemical inertness, wide useful temperature range, and the lowest coefficient of friction of any solid material. Since no lubrication is needed, a possible source of contamination is removed.
Design considerations are given for using teflon in lip seals of the Cup, Rod, U, and V-ring types; also for squeeze-type seals of the square, wedge, and O-ring types.

G-023 B
Gillespie, L.H.
How To Design With Teflon Piston Rings
Appl. Hydraulics and Pneumatics, V. 12, 10, pp. 128-30, October 1959

Article covers properties, starting friction, design factors, reinforced resins, piston ring wear, tolerances and finishes, and rotating seals.

G-024
Gillespie, L.H.
Mechanical Seals - Design, Materials, Installation
Applied Hydraulics & Pneumatics, V. 13, pp. 90-92, Feb. 1960

Brief description of the various types of dynamic shaft seals and a short glossary of dynamic seal terms.

G-025
Gillick, T.J., Jr., and Magnant, F.J.
Synthetic Fiber Felt
Prod. Engng., V. 28, pp. 178-182, July 1957

Properties of synthetic felts for filters, seals, wicks, and gaskets at service temperatures above 200°F; major types are simple interlock, woven insert, pre-stretched and shrunk, and thermoplastic bonded; tables show tensile strength and elongation of fibers; effect of structure on properties; coefficient of friction of synthetic fiber felts against glass, deformation and set at 58.6 psi load, wicking and oil absorption and physical properties of seven fibers.

G-026
Giordano, F.
Less Leakage Now With Shaft Seals
Product Engineering, V. 31, p. 15, Dec. 19, 1960

New seal design and a new method of finishing the surface of a rotating shaft have combined to reduce seal leakage at high pressures and operating speeds. The back up ring is prevented from rotating, but free to move a small distance radially. The back up ring, therefore, tends to follow rather than oppose shaft movement. Test results discussed.
Drawing included.

G-027 T
Glaeser, W.A., Allen, C.M.
The Friction And Wear Of Refractory Materials For Use In High Speed Mechanical Seals
Int. Conf. on Fluid Sealing, Paper F2, April 17-19, 1961 (30 pages), British Hydromechanics Research Association, Harlow, Essex, England

The role of physical properties of ceramic and cermet materials in relation to their high speed, high temperature friction, wear, and surface degradation, as applied to unlubricated radial face-seals. Speeds up to 360 ft/sec. temp. 1000°C.

G-028
Goldoftas, T.
Shape Of Seals
Hydraulics and Pneumatics, V. 14, pp. 75-82, Nov. 1961

Tabular charts listing seals by shape, and divided into elastomeric, leather, and metallic seals. These charts are limited to pressure sealing types for static and reciprocating applications. Chart information includes fluid medium, temperature ranges for various seal materials, maximum pressure, surface finish required, and seal size range.

G-029
Gough, H.J.
Pipe Flanges, Research Committee, First Report
I.M.E. Proc., V. 132, pp. 201-340, 1936

The experiments reported fall into four distinct but related groups. Test made to determine the limiting pressure conditions necessary to maintain tightness of a joint at air temperature, the behavior of metal joints, reinforced asbestos packings, metallic-packing rings, and corrugated sheet have been studied. Assemblies of studs, bolts, nuts, and washers have been subjected to prolonged loading at 975°F and their contribution to the total creep deformation studied. The elastic flexibility of two pair of flanges for 8-inch pipe has been determined. Tests have been made on complete steam pipe assembly curves, data, diagrams, etc.

G-030
Gould, J.R.
Water Seal For Stirrers Used In Mixer-Settler Type Extractor
Knolls Atomic Power Lab., Gen. Elec. Co., Sch'dy., N.Y., KAPL-104, Contract W-31-109-Eng., Jan. 30, 1956

A water seal was developed to prevent vapor leakage out of the mixer shaft standpipes of a horizontal mixer-settler type extractor.

G-031
Graham, J.W., and Zimmerman, W.F.
What's Happened To Cermets?
Metal Progress, V. 73, pp. 89-91, March 1958

A cermet is a metal ceramic combination. Developed as a material which would combine the strength of the metal with the heat resistance of the ceramic.
Al_2O_3 bonded by chromium resisted oxidation up to 2200°F for long periods and 2750°F for shorter periods.
Titanium carbide and chromium combinations were also tried. These were successful between 1400°F and 2000°F.
Zirconium boride cermets are not attacked by molten aluminum.
Nuclear energy applications include fuel elements, control rods, bearings and seals in alkali-metal heat transfer media.
Systems for handling liquid fuels for rocket propulsion have made use of their unique properties, especially in seals at necessary joints or glands.

G-032
Grasby, R.C.
Seals For Passenger Car Automatic Transmissions
SAE Paper, No. 715, 4 p., April 16, 1956

Seals fall into two groups; spring loaded oil seals for rotating shafts, and synthetic rubber cup packings for activating clutches; new materials on which there has been test or production experience are silicone rubber, polyacrylate rubber, butyl-acrylate rubber, and teflon; so called piston packings or lip seals, are basically synthetic rubber cup packings.

G-033 T
Grasby, R.C.
Seals For Passenger Car Automatic Transmissions
SAE Journal, V. 64, pp. 20-1, October 1956

Four new seal materials show promise in seals for automatic transmissions. Silicone, polyacrylate rubber, butyl-acrylate, and teflon are discussed as to their advantages and limitations.

G-034
Gray, S.
Bearing And Seal Development
Mechanical Engng., V. 81, pp. 76-80, April 1959

Study of a high-speed rotating-face-type shaft seal normally used to seal the jet fuel in an aircraft pump. The seal is required to operate dry at an overspeed of 13,000 fpm and at 500F. Then it returns to its original jet fuel operation with no decrease in sealing efficiency. For rotors of titanium and tungsten carbide, seals of the same carbides and also of "1000-F" carbon graphite were tested, with test results tabulated. (In the ASME paper, but not in the condensed version, the development of a Freon-12 shaft-seal design for an aircraft-type refrigeration compressor was discussed.)

G-035
Green, L.A., Miles, H.T., Richardson, A.C.
A Dismountable Vacuum Joint For Bakable Glass Systems
Journal Scientific Instruments, V. 36, pp. 324-325, July 1959

Describes a joint in which a metal gasket can be used between glass tubes without the need for a metal flange; the area of metal exposed to the system gas is thus kept to a minimum. The joint is based on a type made commercially for joining sections of glass pipeline. A butt joint is made between two glass tubes of the same diameter by compressing a wire ring between the ground ends of the tube.

G-036
Green, R.A.
Test Of Packing Rings Proposed For Use In The B26 Airplane Main Landing Gear Retracting Strut Swivel Joint
PB 30514, Off. Tech. Services, 6 p., March 1942, (Army Air Corps Material Div., Memo. Rept. Exp. M-51/B840)

The result of life and sealing tests of hydraulic packing rings. The packing rings were confined to "U" cup rings. Table and schematic diagrams are included.

G-037
Greiner, H.F.
Rotating Seals For High Pressure
Product Engng., V. 27, pp. 140-143, Feb. 1956

Mechanical seals offer solutions to many extreme operating conditions where conventional packings may prove unsatisfactory.
Design factors are discussed.

G-038
Griffis, C.B., Javier, V.S., Wilson, A.
Dynamic Properties Of Elastomers At High Temperatures
Qm. Res. and Engr. Command, Proc. Joint Army-Navy-Air Force Conference on Elastomer Res. and Devel., Oct. 1958

G-039
Griffiths, C.A.
Measuring Packing Performance By Work-Factor Method
Power, V. 71, pp. 905-907, June 10, 1930

Various testing machines used to duplicate exact service conditions. The objective in simulative service tests is, obviously, twofold: (1) to weed out unsatisfactory material, (2) to evaluate material which has otherwise met certain minimum requirements; comparison on test data given in chart form.

G-040 T
Grinnell, S.K.
Flow Of A Compressible Fluid In Thin Passages
ASME Trans., V. 78, No. 4, pp. 765-771, May 1956

The author rederives the expression by which the data for Fanno flow in the Gas Table are tabulated. A simplified analysis is presented where momentum effects are negligible. Valid for height to length ratios of less than 1/1000. Experimental and comparison data graphically presented. Discussion by Hughes is appended.

G-041
Griswold, G.Z.
Selecting Seals And Packings
Mach. Design, V. 11, pp. 34-38, May 1939

Unit bearing seals-assembled and self-contained parts made for a variety of comparatively standard applications are discussed in this article. A few are: spring-loaded radial type, non-spring loaded, various face seals, seal friction, bearing overload.

G-042 T
Grobel, L.P.
Shaft Seal
U.S. Patent 2,314,207, March 16, 1943

This presents a shaft seal design for application to gas filled dynamo-electric machines. The seal, which is situated adjacent to the main rotor support bearings, consists of a non-floating buffered bushing in which the sealant fluid is introduced between two non-rotating seal bushings.

G-043
Groddeck, K.H.
Problems Of Noncontacting High Pressure Stuffing Boxes (In German)
Forschung and dem Gebiete de Ingenieurwesens, V. 23, pp. 183-95, 1957

Investigation of flow in vortex chambers of labyrinth packings; formulas developed for calculating flow velocities at inlet of sealing slits, and leakage; advantages of new type of stuffing box which permits reciprocal movement of stationary and rotating parts of seal in axial direction; means of reducing leakage in condensing steam turbine.

G-044 T
Groves, G.A.
Study Of Package Seal Housing Design For Liquid Buffered Floating Bushing Type Seals
Oak Ridge National Laboratory, CF61-11-42, 16 pp. Nov. 28, 1961

The study compared package seal design with a seal where cavity and service lines are designed as an integral part of the machinery structure. It is necessary however, in most cases to apply the design concept in the initial design of the rotating machinery.

G-045
Grunwald, H.H.
Glands For The Entry Of Electrical Cables Into Pressure Vessels Containing Or Surrounded By Conducting Liquid
I.M.E. Proc., V. 172, pp. 487-498, 1958

Paper concerned with the theory, design and construction of seals through which one or more electrical conductors pass into vessels containing or surrounded by, non-insulating liquids under pressure.

H-001
Haakonson, A.
Brake Device Working On The Fluid Brake Principle
Trans. of Norwegian Patent 44/909/26-58
(A. Capelo - PO Box 46354 - Hollywood 46, Calif.)

Hydraulic Brakes, Seals

H-002
Haas, M.
Packings For Pistons And Valves
Review de Mechanique, 3500 W, Sept., 1898

Discussing especially piston packings for marine engines, operating with high pressure steam, with many illustrations of actual packings.

H-003
Haas, W.
Integrated Seals Simplify Tartar Design
Space/Aeronautics, p. 89, 90, 92, April, 1959

Description of gland-type, mechanical, integrated seals that are used on the Tartar missile.

H-004
Haberman, N.
O-Ring Test Report
G.E. Co., Aircraft Nuclear Propulsion Dept., DC-59-10-15, Contract AT (11-1)-171, 14 p., Sept. 30, 1959

High temperature and radiation testing of silicone rubber O-rings which are used as pressure seals in the XMA-1A shim scram actuator disconnect coupling is discussed.

H-005
Haenle, S.
Contributions To The Strength Behavior Of Flanges With Welded Neck And The Evaluation Of The Sealing Forces Of Some Soft Gaskets On The Asbestos Basis
Forchung auf dem Gebiete des Ingenieurwesens, V. 23, Ausgabe B, No. 4, pp. 113-34, 1957

A method of calculating the strength properties of bolted flanged connections with consideration of the applied screw forces and of the initial pressure.

H-006
Hagerty, W. W.
A New Factor In The Design Of Hydraulic Seals
Proc. Nat. Conf. Indust. Hyd., V. 1, pp. 122-129, Oct. 16-17, 1947

Paper describes a method of sealing a rotating shaft against leakage caused by an axial pressure gradient. In first part of paper the general theory of fluid motion encountered is presented.
In second part manner in which the theory is applied to the problem is demonstrated and some general examples of seal design are given.

H-007
Hagerty, W. W.
Designing Hydraulic Seal
Applied Hydraulics, V. 1, pp. 26-7, 30, Feb., 1948

Essential features of method of sealing rotating shaft against leakage caused by axial pressure gradient; general theory of fluid motion encountered; application of principles of fluid mechanics; development of sealing element; general design examples and applications of resulting hydraulic seal.

H-008
Hall, C. C.
Care Of Metallic Packings
Paper, V. 28, pp. 17-18, April 13, 1921

H-009
Hamkens, H.
Steam Engine Operation And Maintenance, Piston Rod Packing
Power, V. 52, p. 621, Oct. 19, 1920

Discusses some forms of soft packings, their limitations and application; also soft metals and cast iron packing rings. Describes care to be taken when fitting rings. Drawings included.

H-010 B
Hamlin, C. E.
Seals, The Vulnerable Giants
SAE Paper 50S, 6 p., 7 fig., April, 1959

The effects of pressure, temperature, and unusual fluids on present-day seals are described together with some of the recent research work which is being done in these fields. Diagrams of various types of back-up rings are shown including the delta and bimaterial and channel.

H-011
Hamlin, C. E.
Backup Ring Design Affects O-Ring Life
SAE Jour., V. 67, pp. 81-82, May, 1959. Abstract of SAE Paper 50S, "Seals, the vulnerable giants," April, 1959

Effect of various configurations of backup rings on life and proper sealing of Viton A O-rings at high temperatures.

H-012
Hamlin, C.E.
Metal Hydraulic Seals May Be It
SAE Jour., V. 67, pp. 62-63, June 1959. Abstract of SAE Paper 50S, "Seals, the vulnerable giants," April, 1959

Metal piston and shaft seals for high temperature hydraulic systems are better than elastomers for thermal cycling and high pressures but are more difficult to design. Natorq static boss seals of stainless steel, Inconel X, and aluminum are leakproof from -360 to 1000F and 5 to 10,000 psi for liquids and gases such as silicate esters, liquid oxygen and hydrogen, anhydrous ammonia, etc.

H-013
Hammond, R.
Sealing A Rotating Shaft
Chemical Age, p. 199, Aug. 17, 1946

The design configuration on a shaft seal involving a metal bellows is presented.

H-014
Hanson, A.C. and Inman, G.O.
Limitations Of Tin As Packing Material
Indus. & Eng. Chem., V. 31, pp. 662-3, June, 1939

Allotropic transportation on bearing surfaces of tin racking causes leakage; addition of bismuth in tin will increase hardness and decrease elasticity to marked degree.

H-015 T
Hanson, O. P.
Mechanical Seals For Rotary Shafts
Research, V. 12, pp. 61-67, 1959

Fundamental principles of mechanical seals are discussed by the progressive development of a definition of a mechanical seal. Particular attention given to individual components, balancing, face pressure conditions, arrangement, limitations, and capabilities.

H-016 T
Harney, D. B.
Shaft Sealing Device With Standby Seal
U. S. Patent, 2,818,286, Dec. 31, 1957

This is a packing gland to be used as an inactive standby seal. The packing is prevented from activation by a split annular ring, allowing the use of the packing compression sleeve and its flanges as the mount for the normally activated outer seal.

H-017
Harper, D. B.
Seal Leakage In Rotary Regenerator And Its Effect On Rotary-Regenerator Design For Gas Turbines
Trans., ASME, V. 79, pp. 233-245, Feb., 1957

An analysis of rotary-regenerator seal leakage is presented which predicts the leakage quantity and the variations of pressure gradient under the sealing shoe. The verifying results obtained from tests of a simulated sealing arrangement also are given.

H-018 B
Harrington, R.
How Hot Water Affects Elastomers
Rubber Age (NY), V. 84, 5, pp. 798-811, February, 1959

H-019
Hart, L. P.
Compressibility Of Composition Cork
Elec. Wld., V. 118, p. 1640, Nov. 14, 1942
Information on the properties of composition gaskets for liquid filled electrical equipment.

H-020
Hart, L. P. Jr.,
Evolution Of Transformer Gaskets
General Electric Review, V. 53, pp. 34-41, June, 1950

1) Behavior of rubber-like materials in oil and Askarel
2) Nitrile rubber compounds
3) Contamination
4) Mechanical properties
5) Compression set
6) Permeability to water vapor
7) Service tests
8) Application
9) Summary

H-021
Hartz, F. J.
Crane Company Seal Leakage Program
Westinghouse Electric Corp., Atomic Power Division, Contract AT-11-1-GEN-14, Subcontract 73 (14-349), Jan. 31, 1955. 11 p.

A program was initiated to compile test data on conventional sealing materials and to investigate methods of operating valves wherein forces must be transmitted through a pressure wall. Test results, descriptions of apparatus and measuring techniques, and conclusions and recommendations are included

H-022
Hatfield, N. J.
All Metal Reed Seal Evaluation For 0-3000 psig Pressure, Hydraulic Service
PB-125883, Off. Tech. Serv.

Evaluation tests of all metal reed seals were performed during the first quarter of the year 1955. Leakage rates and total leakage were accurately measured while the test shaft cycles through the reed seals. Physical wear on the seals was determined. At temperatures of 300°F, test results indicate the feasibility of using an all-metal reed seal in aircraft hydraulic systems at normal operation pressures up to 3000 psig. AD 97637, Proj. No. 1371, Contract AF 33(600) - 28968, WADC TR 55-302.

H-023
Hayman, R. L.
O-Ring Packings Adapted For 3000 lb. Pressures
Prod. Engng., V. 16, pp. 56-58, Jan. 1945

A successful method of utilizing synthetic rubber O-rings in reciprocating hydraulic equipment at double the maximum pressure previously considered practical is described.
Problem solved by using leather backing rings on both sides of the O-rings.

H-024
Hayman, R. L.
Performance Of O-Ring Packings In Rotary Seals For 3000 lb. Pressure
Prod. Engng., V. 16, pp. 190-191, March, 1945

Most frequent application in aircraft hydraulics is on swing joints, a term applied to line joints subjected to rotary action. Generally used in lieu of a high pressure hose.
In developing and testing O-ring seals for swing joints, the largest diameter swing joint required in a 3000 psi application was selected as the critical member. Two test joints were constructed to same dimensions. Results of tests given.
Tests are also described on a single O-ring seal at 0-3000 psi., pulsating pressure, at a rate of pressure increase of 200,000 psi per sec.

H-025
Haynes, W. B.
Use Of Metallic Gaskets
Power, V. 47, p. 906, June 25, 1918

H-026
Hayward, A. G.
Simple Rotary Vacuum Seal
J. Sci. Instr., V. 29, p. 410, Dec. 1952

A 1/16-inch neoprene washer with a central hole and lubricated with Apiezon wax is held against a threaded circular base by the rim of a shaft through the central hole of the washer. The shaft is held in position by a knurled nut screwed onto the circular base and locked by a knurled screw.

H-027
Hayward, A. G.
Simple Vacuum Seals
Vacuum, V. 11, No. 3, pp. 262-264, July, 1952

Describes four types: a) simple rotary seal, b) seal for continuous rotation, c) seal for current lead, d) air leak seal
Neoprene washer; ring, ball, and washers supporting pieces. Drawings included.

H-028
Headrick, R. E.
Composite Seal Materials For Extreme Environments
Wright-Patterson Air Force Base, Aeronautical Systems Div., Tech. Doc. Rept. ASD-TDR - 62 - 286, 23 p., Mar., 1962. Also SAE Journal - V. 70, pp. 58-63, Oct. 1962

Evaluation of metal fiber skeletons impregnated with softer organic and inorganic materials, as seals at both static and dynamic conditions, from liquid nitrogen temperatures to 1000°F, and at pressures up to 5000 psi, have yielded unsurpassed results. The metal fiber skeletons (stainless steel or molybdenum) provide the strength and resilience, and the impregnating materials (soft metals, soft metal alloys, or compounded polymeric materials) provide the sealing barrier. In some cases the impregnants have increased the resilience and lubricity of the composite. The limitations, potentials, and the availability of these materials are discussed.

H-029 B
Heaps, N. B.
How To Extend Centrifugal Pump Life Erosion, Packings And Bearings, Causes And Cures.
Chemical Engineering, V. 66, 11, pp. 128, 130, 132, June 1, 1959

A number of considerations for erosion, packing, and bearings, cause and cures.

H-030
Hearst, J. R., Ahn, S. H., Strait, E. N.
Vacuum Seals At Liquid Nitrogen Temperatures
Rev. Sci. Instrum. V. 30, No. 3, p. 200, Mar., 1959

O-rings made of Kel-F elastomer have been found satisfactory at liquid nitrogen temperature. Recommended sizes and application techniques.

H-031
Heathcote, V. A., Read, W. E.
Demountable Seal For High Vacuum Work
Jour. Sci. Inst., V. 34, p. 247, June, 1957

Standard copper steam jointing rings, manufactured by Hulburd Patents Ltd., were compressed between two steel flanges and the space between the contact edges evacuated by means of a rotary oil pump. The seals could be baked for prolonged periods at 450°C without showing any sign of leak. Test data and drawing included.

H-032
Hees, G. W., Eaton, W., Lech, J.
The Knife-Edge Vacuum Seal
Vacuum, V. 4, No. 4, pp. 438-44 (Oct. 1954; Pub. June 1957)

A reliable high-vacuum sealing technique of wide applicability and capable of containing an ultra-high vacuum is described. This seal can readily be assembled or disassembled. It can withstand repeated and prolonged bakeouts as well as liquid nitrogen temperatures. Seals ranging from less than one inch to more than twelve inches in diameter have been constructed. A wide variety of materials have been joined.

H-033 B
Heffner, F. E.
A General Method For Correlating Labyrinth Seal Lake Rate Data
A.S.M.E. Paper No. 59 - Lub - 7, 8 pages, Oct., 1959, See also ASME Trans., Journal of Basic Engineering, V. 82, No. 2, pp. 265-275, June, 1960

Method for correlating test data which allows calculation of gas leak rates for an entire family of labyrinth seals, from test of two characteristics; accuracy three-percent. Two dimensional graphs are plotted for simultaneous solution. Valid for both staggered and straight-through labyrinths.

H-034 T
Heher, W. M.
Determination Of Coolant Loss Through Double Gland Turbine Seal
Goodyear Atomic Corporation, G.A.T.-T.811, 5 pages, May 27, 1960

An infra-red analyzer is arranged as a detector for leakage of a refrigerant from power recovery turbine seal. The method of leak measurement during seal test is discussed.

H-035
Heher, W. M.
Development And Performance Of The Prototype Seal For The Power Recovery Turbine
Goodyear Atomic Corporation, Report GAT-388, 17 pages, May 12, 1961

A double-gland, positive contact seal was developed for use in a power recovery turbine operating on coolant 114 vapor (dichlorotetraflourethane). The average leak rate during 12,500 hours of successful operation was 0.235 pounds per day.

H-036
Heinrich, J. T.
Simple Vacuum Coupling
Rev. Sci. Instrument, V. 29, p. 1053, Nov., 1958

Work with metal vacuum systems often requires a coupling that is readily machined, quick to assemble, and does not pose difficult alignment problems. Such a coupling may be fashioned from a standard flare fitting. Fittings are selected 1/8" larger than the OD of the tubing used. The conical surface of the male portion is machined to form a flat face. Greased O-ring is pressed against the flat surface and the tube by gasket (which is stock part in flare fittings) and a flare nut.

H-037
Heizen, M.B.
Desert Tests (1952) Experimental Oil Filter Gaskets.
PB 159 620, Off. Tech. Serv., 31 p., 2 ref., 23 Sept., 52. (Rept. 39 on Proj. No. T.B. 5-1401, Aberdeen Proving Ground)

Eight experimental gaskets were installed on trucks ranging in size from 3/4 ton to 5 ton. These trucks were operated approximately 10,000 miles each in ambient temperatures as high as 123°F. Only one oil filter element change was made on each vehicle during the course of operation, but at the conclusion of the test the gaskets were not considered serviceable for additional use.

H-038
Hemeon, J. R. and others
Leakage Control
App. Hydraulics, V. 10, P. 53 between p. 81 and 168, No. 5, May, 1957

The following are included:
What they say about leakage control
Practical design ideas
Testing packings
Port seals
How O-rings seal air and oil components
How teflon packings behave
Designing seals
Control of contaminants
Static sealing with metal O-rings
Selection of flareless fittings, etc.

H-039
Hemphill, J. W.
Packings For Power Plants
Nat. Engr., V. 41, pp. 678-80, Dec., 1937

Suggestions regarding important factors to be considered in selection of packings; hints on construction of stuffing boxes; application of packing.

H-040 B
Hemphill, J.W.
Do's And Don'ts Of Packing Practice
Paper Mkr. & Brit. Paper Trade J., pp. 37-8, 40, 42 Midsummer, 1951 (Reprint from Paper Industry USA)

Practical advice on obtaining the best performance from gland packings. Every gland should be looked upon as an energy-absorbing unit and thus should be adjusted so that the power loss is a minimum.

H-041
Hepworth, J.L.
Sealing Rings For Wet Cylinder Liners
Sci. Lubrication, V. 9, pp. 24-26, No. 3, March, 1957. Also abstract in Chartered Mech. Engr. V 4, pp. 20-21, No. 1, Jan., 1957

Three basic types of wet liner for engine cylinders; only types X & Y are considered, these being sealed at top by compression of flange, while lower end is sealed by rubber rings; functions of sealing ring; measurement and calculation of wall pressure; results of tests and design data; sealing ring material and ring finish; method of assembly.

H-042
Herle, F.
Novel Design Of Metallic Packing Stands Test Of Time
Power Plant Engineering, V. 29, p. 915, Sept. 1, 1925

Assembled segments of metallic packing, cast from babbitt, interlock and are held together by coil springs. Still operating successfully after ten years.
Drawings of construction detail and assembly.

H-043
Heyer, K.
Testing Sealing Agents For Pressurized Cabins
Lilienthal Ges. Luftfahrtforsch, p. 56, Rept. 29
Spec. Libr. Assn. Trans. 3214 (17p), V. 3, No. 8, Aug., 1957

H-044
Heywood, W. A.
Report On Static Seal Test
Knolls Atomic Power Lab., General Electric Company, AECD-3081, March 12, 1951

Tests were made on a gasket-type seal of neoprene rubber. From 4 to 12 standard 1/2-13 holddown bolts in steps of 2 were tried. Data given on tests.

H-045
Heywood, W. A., and Hilbert, C. J.
Control Rod Gas Seals
Knolls Atomic Power Lab., General Electric Co., KAPL-669, Jan. 14, 1952

Apparatus designed to test gas seals for the KAPL Intermediate Power Breeder reactor control rods is described. Specifications and drawings of the seals are included and test data given.

H-046
Heywood, W. A.
Frozen Sodium Shaft Seal
Knolls Atomic Power Lab., General Electric Co., KAPL-1265, Contract W-31-109-Eng-52. Aug. 4, 1954

The feasibility of using a frozen sodium annulus as a shaft seal was investigated. The torque required to turn a shaft in a frozen annulus, the leakage of sodium past a frozen annulus was studied.

H-047
Heywood, W. A.
Control Rod Gas Seals
Knolls Atomic Power Lab., General Electric Co., KAPL 1161, Contract W-31-109-Eng-52, Jan. 25, 1955, 12 p.

The experimental work to develop a seal with a H_e leak rate of less than $2x10^{-6}$ cc/min is described, and the mechanism controlling the lower leak rate limit for elastomer seals is demonstrated.

H-048 T
Hibbard, R. L.
Carbon Bearing Seals
Materials and Methods, V. 45, pp. 110-114, March, 1957

The three types of carbon generally used for bearings and seals are pitch coke-bonded lamp black, pitch-coke bonded graphite, and impregnated graphite. Properties of carbon are compared with other materials and areas of superiority outlined. A guide to the design of carbon bearing and seals is included.

H-049
Hildenbrand, J. L.
SAE Jour., V. 65, pp. 78-79, Sept. 1957. Also in Soc. Automotive Engr. Paper - N21, Jan. 14-18, 1957

Research and development program for improvement of bearing seals for track rollers of crawler tractors; study carried out over five years at International Harvester Company; factors in design of test equipment to evaluate seal characteristics; three stages of laboratory tests and field tests.

H-050 B

Hill, M.
Performance Tests Of Seals, Oil, Plain Or Plain Encased For Shaft Sizes 0.875, 2.625, and 3.50 inches
NAVORD Reports 5529-5531 and 5533-5534, Apr-May, 1958 (Pbs 143236-143240)

Five reports describe results of leakage rate and torque tests on a range of lip seals made by five different U.S.A. seal manufacturers. Operating conditions described 70° - 74°F, - 20 to -65°F, 2000 ft/min., eccentricity (shaft) 0.008 on 3.5 in. dia.

H-051

Hill, R. M.
Cork-Rubber Materials
Product Engrg., V. 10, pp. 239-241, June, 1939

Various applications of cork-rubber compositions, load deflection properties; molded packings for use in both static and dynamic sealing; illustrations of types.

H-052

Hinkle, R. J.
What The Users Of Mechanical Packings And Rubber And Asbestos Products Should Know
India Rubber W., V. 82, p. 65, Sept. 1930

H-053

Hinkle, R. J.
Designs For Applying Press-Fit Seal Units
Product Eng., V. 12, pp. 370-1, July, 1941

Sketches showing how bearings seal units of standard bore simplify bearing design and lower costs.

H-054

Hintenberger, H
Experience With Metal Foils As High Vacuum Seals
Zeits fur Naturforschung, 6A, pp. 459-462 (7p) 1951
Spec. Lib. Assn. Translation #1800, (7p), V. 3, No. 1, Jan., 1957 (AEC-tr 2554)

Details are given of the performance of aluminum foil as a packing material for vacuum tight flanged pipe joints. The design of the flanges and the processing of the foil necessary to give satisfactory results is discussed. The joints can be exposed to ordinary baking temperature up to 280°C.

H-055 T

Hirano, F., Ishiwata, H., and Kambayashi, H.
Friction And Sealing Characteristics Of Oil Seals
Int. Conf. on Fluid Sealing Paper A4 (24 pages) April 17-19-1961, British Hydromechanics Research Assoc. Harlow, Essex, England

Much experimental data on friction and leakage in lip seals and mechanical face seals.
Hydrodynamic analysis
Shaft speed, leakage, friction, seal size, shaft surface roughness, surface pressure and its distribution, are among variables discussed and related.

H-056

Hobbs, R. B.
A Study Of Specifications For Chrome-Tanned Hydraulic Packing Leather
J. Am. Leather Chem. Assoc., V. 41, pp. 573-90, 1946

Variations in the requirements of Federal specifications are discussed. The results of tests of 6 com. leathers are presented.

H-057

Hobline, L.E., Humphrey, S.A., Dinsmon, T. E.
Kinetic And Static Friction Of The O-Ring Type Gasket
PB-120850, Off. Tech. Serv., 29 pgs.

Survey was undertaken to provide supplementary data on the character of the frictional forces to expect when employing O-ring type gaskets, and the results are presented in the form of design work sheets. (NAVORD 2281)

H-058

Hodder, B.H.
Vacuum Or Low Pressure Seal Utilizing Modified Standard Refrigeration-Type Flare TubeFittings
Journal of Scientific Instruments, V. 35, p. 182, May, 1958

The modification comprises a groove machined in the flare fitting to retain standard neoprene O-ring packings. The tube flare is then screwed up to touch the flare fitting. The O-ring is compressed in the groove and the seal made. Drawings and table of relevant dimensions and details for flare fitting modification.

H-059 T

Hodkinson, B.
Estimation Of The Leakage Through A Labyrinth Gland
I.M.E. Proc., V. 141, pp. 283-288, 1940

Reviews methods of calculating leakage for labyrinth seals. Test results presented graphically in the manner of Elgi. Stationary labyrinth test. Analysis assumptions are reviewed and examined for error.

H-060

Holden, J., et al.
Bakeable Vacuum Seals Using Aluminum-Wire Gaskets
Jour. Sci. Inst., V. 36, pp. 281-283, June, 1959

Static seal for use up to 550°C. It is used between steel flanges. Gives information on the minimum compression force which must be applied to an aluminum wire in order to form a cemented joint at elevated temperature.

H-061 T

Hollander, A., Hoover, V.A., and others
Liquid Seal
U.S. Patent 2,295,579, Sept., 15, 1942

A seal for a vertical shaft in which mercury is used as the sealing means. A cup-shaped container rotating with the shaft is filled with mercury. An annular baffle secured to the stationary housing extends downward into the mercury isolating the interior and exterior portions of the housing. A means is provided to keep the mercury in contact with the shaft, opposing the effect of centrifugal force.

H-062 B

Holoubeck, F.
Transitional Frictional Effects In Powered Controls With Particular Reference To Hydraulic Jacks
Royal Aircraft Establishment, Report No. Mech. Engg. 12, Sept. 1949

The phenomenon of "judder" in jacks, when the piston rod moves in a series of increments has been analyzed by considering the jack as an elastic medium. Rubber seals provide friction, the friction under boundary conditions may well be 10 or more times greater than under film conditions.

H-063

Holt, J. B. and Miller, W. S.
Dynamic Seals And Packings
Machine Design, V. 29, pp. 69-98, Oct. 31, 1957

Fluid tightness and service life - the principal measures of seal performance - meet design objectives when seal limitations are accurately matched to service demands. Prediction, not measurement, however, represents the first order concern of the designer. Basic to prediction of seal performance is study of these service conditions.
Pressure, Speed, Temperature, Fluid characteristics.
This article shows how these factors are analyzed to provide guides to selection and application of dynamic sealing methods.

H-064

Horning, D. O.
Vacuum Vessel Construction Techniques And Sealing Methods
Inst. Eng. Res. Univ., Calif. Tech. Rept. No. HE-150-84, 15-5-51

The factors which govern the design of large-size vacuum plant are discussed. (Wind tunnel) (external press. 14.7 lbs. psi - vac 1×10^{-6} mm Hg) Leak detection methods discussed. Permanent and mechanical joints are discussed.
Mechanical seals with gaskets, and O-ring seal teflon material.
Design calculation included.

H-065

Hornschuck, H.
Packing Problems On High-Pressure High-Temperature Pumps
Power, V. 87, pp. 225-226, 280, 281a, April, 1943

Pressures and temperatures involved in centrifugal pump stuffing boxes have surpassed tolerance of packing materials. Control obtained by throttling and external cooling both unsatisfactory expedients. Mechanical sealing devices being considered.

H-066

Hornschuck, H.
Why Mechanical Seals Improve Centrifugal Pump Operation
Power, V. 87, pp. 524-526, Aug. 1943

Lists how's and why's of putting stuffing box seals to good use on centrifugal pumps and lists the conditions that must be satisfied for successful operation.
Such information as seal construction balanced vs. unbalanced seals, lubrication, leakage, etc. are discussed.

H-067 T

Horsley, W. D.
Hydrogen Cooled Alternators
The Engineer, V. 186, pp. 587-589, Dec. 10, 1948

A buffered seal integral with bearing, a floating bushing liquid buffered seal, and a combination labyrinth and liquid buffered mechanical face seal, have been used with some success. The quantity of hydrogen makeup is not proportional to the leakage unless the leakage is excessive.

H-068

Hosang, E. (In German)
Recent Developments In Field Of Elastic Gaskets For Moving And Detachable Machine Parts
Werkstoffe u Korrosion, V. 2, pp. 191-4, May, 1951

Mechanical, thermal, electrical and chemical behavior; gaskets and stuffing boxes of synthetic rubber such as perbunan, silicones, thiokol and polyethylenes; suggestions and examples for design and installation of mechanically and chemically stressed packings and stuffing boxes.

H-069 B

Hough, A. T.
The Resistance Of Leathers To Hot Oil, And The Preparation Of Leather Suitable For Oil Seals
J. Society Leather Trades Chemist, V. 34, 4, pp. 138-49, April, 1950

H-070

Houghton, E. F.
Design And Application Of Leather "U" Packings
Modern Ind. "Press," V. 7, pp. 22, 24, 32, May, 1945

Original "U" packings had rounded corners. Improved reliable design packing with square bottom made possible with advent of more flexible mineral tanned leathers. Packing used most commonly in conventional gland, described in article "U" packing "Don'ts" listed.

H-071

House, P. A.
Self-sealing of Aerospace Ships By The Double Hollow Wall Concept
Aeronautical Systems Div., Wright-Patterson AFB, Technical Doc. Rept. No. ASD-TDR-62-248, Feb. 1962, 14 p.

To protect aerospace ships from loss of air when their walls are punctured by foreign objects such as micrometeoroids, a self-sealing wall structure was devised. The wall structure consists of an outer layer of aluminum sheet, a sheet of 3000 psi buna-N, a layer of fluid, another sheet of buna-N, a layer of a second fluid, and another sheet of buna-N comprising the inner wall. When the system is punctured, the two fluids mix, immediately solidify, and seal the puncture. Holes up to 0.075 inch in diameter can be sealed in this manner.

H-072

Housley, J. E.
Aluminum As A Gasket Material
Power, V. 65, p. 374, March 8, 1926

A description of events leading to the replacing of composition and lead gaskets with 2SO aluminum used (commercially pure and annealed dead soft).

H-073
Howard, J. H.
Designing With Metal Bellows
Machine Design, V. 26, pp. 137-48, Jan., 1954

A complete reference on the design of metal bellows, with tables on materials hysteresis, coefficients of thermal expansion and drawings showing various applications.

H-074
Howe, P. G.
Greaseless Vacuum Rotor Seal
Rev. Sci. Instr., V. 26, p. 625, 1955

Teflon ring gaskets which fit between the joints of a standard ground-glass joint or a hollow barreled vacuum stopcock and which are made vacuum-tight by Hg seal have been used in the construction of a vacuum rotor seal capable of holding a vacuum, while pumping, of about 10^{-5} MM Hg.

H-075 B
Hudson, R.A.
Some Recent Developments In Truck Wheel Seals
S.A.E. Paper 119, V. 3, p. 7, fig., October, 1959

A recent advance in lip seal design is described and should be attractive to all users insofar as they will no longer have to ensure that the shaft finish is up to any special standard. It would seem that the design will also have applications as a corrosive fluid seal since the metal sleeve can be made from a high corrosive resistant material.

H-076 T
Hughes, D. P.
Shaft Seal For High Gas Pressures
Int. Conf. on Fluid Sealing, Paper C1, 14 pages, British Hydromechanics Research Association, Harlow, Essex, England

A discussion of the design and development of a visco seal for a shaft which penetrated a nuclear reactor containment vessel wall.
Shaft 4.725" turning at 1500 rpm sealing against gas pressure 150 psi. Development work was done on a 1.5748" shaft turning at speeds up to 5000 rpm.
Complete description of test facility, experimental results, and final design configuration is presented.

H-077 T
Huhn, D.
Theory Of Fluid Sealing
Int. Conf. On Fluid Sealing, Paper D2, (8 pgs.), April 17-19, 1961

Basic laws for leakage, wear, and power requirements are deduced from the general equation for fluid motion, for radially and axially contacting seals. Pressure forces, and their influence, as well as improvement by hydraulic balance are discussed. Shaft eccentricity, vibrations, surface conditions to power consumption, and heat development by rotating seal parts were not considered.

H-078
Hull, J. W. (Boeing Aircraft)
The Development of Metal Piston Rod Seals For High Temperature Hydraulic Actuators
S.A.E., Technical Committee A-6 Meeting, May, 1958

H-079
Hull, J. W.
Ring-Spring Design For High Performance Metal Static Seals
Hydraulics & Pneumatics, V. 13, pp. 122-126, Sept., 1960

Eliminates necessity for a precise degree of axial compression for effective sealing that is a defect in face sealing metal seals. Ring-spring static seals give excellent results from -65 to 700°F, with cycling pressures of 0 to 4000 psi, and surges to 6000 psi. Operation and design are fully explained. Seals developed for aircraft hydraulic systems.

H-080 B
Hummer, H.
Material Selection For Mechanical Seals
American Society of Lubrication Engineers, Paper 58AM-A3, 5 pp., April, 1958

This paper presents the opinions of the Director of the Dura-Metallic Corp. on the most suitable face materials for mechanical seals. Fluids covered are water (hot and cold), petrol and other hydrocarbons, oil and oil solutions and acids. The merits and drawbacks of such materials as stellite, ceramics, and glass filled P.T.E.E. are discussed.

H-081
Hutchison, R. O.
Seal And Gasket Materials For Liquid Nitrogen
Carbide and Carbon Chemical Corp., Y-12 Plant Report Y-268, Sept. 22, 1948, 11 pp

Tests were conducted to determine the most suitable materials for a vacuum tight gasket and seal on equipment coming in contact with liquid nitrogen. Aluminum, copper and lead were used as metal gaskets. Duxseal, lubriseal, glyptal, bouncing putty, silicone rubber, fluorinated hydrocarbons and silicone high vacuum grease were used as seals. A lead gasket with silicone high vacuum grease gave superior results and is recommended for the application described herein.

H-082
Hyde, G. F., Puchsluger, J. H.
Materials For Elevated Temperature Piston Ring And Seal Ring Applications
Lubrication Engrg., V. 17, pp. 476-482, Oct., 1961

Explanation of basic functions of piston rings and seals and summary of properties required of materials for such parts necessitated by increasing temperatures and reduced lubrication.
Materials tested divided into following categories; carbons, carbides, nitrides, borides, and intermettalics, ceramics and cements, irons and steels, high temp. alloys, hard facing alloys, surface treatment and coatings.
Best materials standpoint of wear and oxidation resistance were found to be impregnated carbons, metallic carbides, and intermetallics, most of which unfortunately present problems of design and fabrication. Temperature range 600 to 1200°F. Rubbing speed 1, 600 sfpm.

H-083
Hyler, J. E.
Practical Points On Packing
Mill & Factory, V. 48, pp. 102-106, March, 1951

Gives fundamental "whats-and-whys" of packing construction and performance with particular emphasis on requirements in the hydraulic and pneumatic fields.
Some packing materials mentioned are leather O-rings, asbestos fabric, asbestos yarn. The last mentioned material is often employed in connection with soft-metal foil, which is wrapped around it, generally by helical wrapping. Such a wrapping is suitable for steam applications to 550°F. Copper-foil covered packings having a core of dry asbestos may be used against steam up to 1000°F.

H-084
Hyler, J. E.
Oil Seals On Modern Machinery
Canadian Machinery and Manufacturing News, V. 65, pp. 112-3, 140, 142, Aug., 1954. Sept. pp. 144, 146, 148, 152

Oil seals applied to rotary shafts on different types of machinery; factors important in design of seals; use of standard or stock oil seals recommended because of their lower cost. Application of single spring loaded seals employed in Gemmer hydraulic steering gear, rotary gear pumps, chain saw, in diesel engine governors, etc; manufacture and advantages of axial and radial seals.

I-001
Inall, E.K.
Shaft Seal For Pumps Handling Low Temperature Sodium-Potassium Alloy
Rev. Sci. Inst., V. 32, 1153-4, Oct. 1961

Details of the seal mounted in the housing, which can be filled with oil to a suitable level visible in the filler. Lubricates the lip seal and cools it by circulation induced by the spinning shaft.

I-002
Ingham, E.
Pump Packing
Colliery Guardian, V. 170, pp. 35-37, July 13, 1945

Suggestions as to selection, care, application, and periodical inspection of pump packings.

I-003 B
Iny, E.H., Cameron, A.
Rotary Oil Seals
British Hydromechanics Research Association, RR 579, 11 pp., 21 fig., February 1958

A large amount of experimental data obtained from the lip seal apparatus made by B.H.R.A. is presented and the author attempts to correlate this to current theories.

I-004 T
Iny, E.H., Cameron, A.
Synthetic Rubber Rotary Shaft Seals - The Fluctuation Of The Interfacial Gap At Moderate Shaft Speeds
British Hydromechanics Research Association, RR 683, Jan. 1961

I-005 T
Iny, E.H., Cameron, A.
The Load Carrying Capacity Of Rotary Shaft Seals
Int. Conf. on Fluid Sealing, Paper A2, (14 pages) April 17-19, 1961, British Hydromechanics Research Assn., Harlow, Essex, England

The effect of vibration on performance of a synthetic rubber rotary shaft seal is investigated. Under normal working conditions the interfacial gap fluctuates at twice the shaft speed, this mode of oscillation permits the generation of oil film between the faces which is capable of carrying considerable loads.

I-006 T
Isenbarger, R.O.
Theory And Practice In Design Of Hydraulic Seals
Applied Hydraulics, V. 3, pp. 14-16, 20, 40, Feb. 1950, pp. 22-24, March 1950; pp. 24-26, May 1950

Definitions, effects of operating conditions, properties of sealing materials, and desirable characteristics for various sealing conditions.
Radial and axial mechanical dynamic seals are described in May issue.

I-007
Isenbarger, R.O.
Surface Type Seal
Proc. of Eighth National Conference on Industrial Hydraulics, V. 6, pp. 220-228, Sept. 1952

Description and operation of the various types of flat-surfaced axial shaft seals.

I-008
Isenbarger, R.O.
How To Prevent Oil Seal Failures
Product Engng., V. 24, pp. 154-159, Jan. 1953

Reasons for shaft seal failure listed together with remedies for failure. Check list supplied as guide for proper seal selection.

I-009 B
Isenbarger, R.O.
Sealing Devices For The Exclusion Of Foreign Material From Hydraulic Mechanisms
Proc. National Conference Industrial Hydraulics, V. 14, pp. 207-19, disc. pp. 219-22, Oct. 1960 (Abstracted from Hydraulics and Pneumatics, Oct. 1960)

Special consideration to exclusion sealing operational conditions, motion, material to be excluded. Axial and radial exclusion seals for rotating shafts. Wipers, scrapers, boots and diaphragms for reciprocating shaft.

I-010 T
Ishiwata, H., Harabayashi, H.
Friction And Sealing Characteristics Of Mechanical Seals
Int. Conf. on Fluid Sealing, Paper D5, 22 pp., April 17-19, 1961, British Hydromechanics Assn., Harlow, Essex, England

Large number of radial face seal test in which the seal is considered to be analogous to a thrust bearing. It is shown that the coefficient of friction of a seal pair varies from 0.01 to 2.0, and suggests that this variation may correspond to the various regions of hydrodynamic lubrication. Experimental data correlation with nondimensional groups.

I-011
Islinger, J.S.
Development Of High-Low Temperature Aircraft Canopy Seals, Part 5: New Material Seals Of Current Design
P.B. 128235, Off. Tech. Serv., June 1956, 57 pp., also Contract AF-33(600) - 27588, ARF Proj. MO58, AF WADC TN 54441, Part I

Silicone rubber seals were evaluated in Part I of a program to develop high-low temperature cabin pressurization and rain seals to be used with present and future aircraft canopies. Three representative seal designs were evaluated. The seals fabricated from recently developed silicone elastomers, were to be capable of functional operation at extreme temperatures of -85°F and +200°F. The evaluation tests were conducted with each seal installed in a test fixture simulating a "half size" aircraft canopy.

I-012
Iwanami, S. Kato, H., and others
Oil Leakage From O-Ring Packing
JSME Bull., V. 2, pp. 638-643, Nov. 1959

Experiments were conducted to find out the behavior of oil leakage from an O-ring packing used for gaskets and reciprocating parts.
Conclusions:
1. For gasket: Seals can be made leakproof for oil pressure up to 280 Kg/cm^2, 4000 psi limit of this test pressure.
2. For reciprocating use: Quantity of oil leakage is proportional to the square of the reciprocating velocity of the rod, the n-th power of the kinematic viscosity of the oil.

I-013 T
Iwanami, S., Tikamori, N.
Oil Leakage From An O-Ring Packing
Int. Conf. on Fluid Sealing, Paper B2, 10 pp., April 17-19, 1961, British Hydromechanics Research Assn., Harlow, Essex, England

Report on experiments conducted to determine leakage characteristics of O-ring packings used for reciprocating shafts. Conclusions of experimental results.
1. The compressive force for squeeze is proportional to 1.2 power of the cross-sectional diameter of the O-ring, 1.3 power of the squeeze and 4.5 power of JIS hardness of the rubber.
2. Reciprocating use; the quantity of oil leakage is proportional to the square of reciprocating velocity of the rod and n-th power of fluid viscosity.
(N = 1.5)

J-001
Jackson, H.D.
Common Sense About Packings
Nat. Eng., V. 51, pp. 252-6, April 1947

Some reasons why packings fail; factors which affect packing service; problems encountered in centrifugal pumps; burnishing plungers to eliminate troubles; proper proportions of rods and stuffing boxes; shaft clearances and lanterns; stuffing boxes and packings for stop valves; packing failures from operating errors.

J-002
Jacobs, E.
How They're Sealing Cylinders
Hydraulics & Pneumatics, V. 15, pp. 67-76, 78, August 1962

Two static seals: the end seal and the piston-to-rod seal assembly. Two dynamic seals: the piston seal assembly and the rod bearing seal assembly. Covers lip seals, V-ring seals, cup seals, U-ring seals, flange seals, squeeze-type molded seals such as D, delta, T, square, X, and O-rings, rider or wear rings, back-up rings, and piston rings.
Materials are metals, polymers, plastics, leather and fabric-reinforced rubber.
Discusses a temperature range from -450F to 1500F.
For high pressures, ring-type seals and proper assembly are important.

J-003 T
Jacobson, M.J., Charnes, A., Saibel, E.
Studies In Lubrication: The Complete Journal Bearing With Circumferential Oil Inlet
Trans. Amer. Soc. Mech. Engrs., V. 77, pp. 1179-1183, 1955

An approximate pressure solution is developed for the Reynolds' equation for the case of the end-lubricated complete journal bearing.
Expressions are obtained for the load-carrying capacity, coefficient of friction, axial thrust, and lubrication rate flow. Applicable to the buffered bushing seal.

J-004
Jacquet, A.E.
Metal Gasket For Cylindrical Elements In Motion
LC or Spec. Lib. Assoc. Trans. #59-12919

A gasket is described which prevents passage of fluid between two cylindrical elements in reciprocal motion. The gasket is composed of a bell mounted metal sleeve, cut with a groove concentric to the axis and having a straight flank and an oblique flank. The sleeve is combined with an expansion ring of a metal harder than that of the sleeve, fitting the profile of the groove and tending to press the thinned lip of the groove against the element in motion.

J-005
Jaeckel, R., and Kammerer, E.
Suitability Of Buna Rubber For High Vacuum Packing (In German)
Zeit fuer Technische Physik, V. 23, pp. 85-8, 1942

Results of investigation demonstrate that buna rubber products can be produced which have no greater gas emission than natural rubber in high vacuum.

J-006
Jagger, E.T.
Rotary Shaft Seals
Scientific Lubrication, V. 9, pp. 24-27, April 1957

Mechanism of sealing is that the surface tension of the oil forms a meniscus which prevents oil flow through the gap between the seal and the shaft.

J-007
Jagger, E.T.
Study Of The Lubrication Of Synthetic Rubber Rotary Shaft Seals
IME Proc. of Conf. on Lubr. & Wear, pp. 409-415, Oct. 1, 1957

Subject of the paper is the investigation into the nature of the oil film which is present between the sealing lip of a conventional synthetic rubber oil seal and the shaft on which it operates.
Conclusion: Oil seals of the type under discussion have the sealing point separated from the shaft by a coherent film of oil with a thickness of 10^{-4} in.

J-008
Jagger, E.T.
Rotary Shaft Seals: The Sealing Mechanism Of Synthetic Rubber Seals Running At Atmospheric Pressure
I.M.E. Proc., V. 171, pp. 597-616, 1957

Very little was understood about the sealing mechanism. Article describes equipment and tests which were run to throw light on this subject. It was concluded that when the seal is functioning normally there is a coherent film of oil between the sealing lip and the rotating shaft. The mechanism of sealing is that the surface tension of the oil forms a meniscus which prevents oil flow through the gap between the seal and shaft.

J-009 B
Jagger, E.T.
The Sealing Of Anti-Friction Bearings
Sci. Lub., V. 10 (Special Extra Issue), pp. 39-40, 42, 44-47, Nov. 1958

This article describes many of the different ways in which lip seals may be used to reduce leakage from bearings. Several diagrams illustrate the various arrangements and the effects of working conditions.
Other factors on seal life are considered.

J-010 B
Jagger, E.T.
Effect Of Cylinder Surface Finish On The Performance Of Hydraulic Packing
Proc. 2nd European Fl. Power Conf., pp. 53-62, April 1960

Comprehensive study of the relation of cylinder surface and hydraulic packing for 500 different packings and 76 different tubes.

J-011 T
Jagger, E.T.
Rotary Shaft Seals - Some Effect Of Shaft Surface Finish
Int. Conf. on Fluid Sealing, Paper A1, 14 pp., April 17-19, 1961, British Hydromechanics Research Asm., Harlow, Essex, England

Results of experimental work on the effect of shaft surface finish on performance of lip seals.
Surface finish measurement. Wear of shaft and seal, extended running test.
Influence of dirt and grit. Effect of hardness of shaft, and initial shaft surface finish.

J-012
Jenner, A.W.
Metallic Gland Packing For High Pressure
Mech. World, V. 97, pp. 327-8, April 5, 1935

Principles and application of successful metallic packings.

J-013
Jerie, J.
Flow Through Straight-Through Labyrinth Seals
7th Int. Cong. Appl. Mechanics Proc., V. 2, pp. 70-82, 1948

Improved efficiency of steam or gas turbine and axial compressor blading, compressor impellers, etc. together with increasing pressure ratios enhance the importance of leakage losses through labyrinth seals. Better knowledge of flow through these seals is desirable. This covers such a study.

J-014
Johnson, A.E.
Pipe Flanges Research Committee, Third Report
Proc. Instn. Mech. Engng., London, V. 168, pp. 423-64, 1954

The third report is concerned with the results of a program of investigation into the plastic and plastic behavior of flanges, drilled and undrilled, by means of model flanges and certain bolt material.
An addendum to the report refers to matters which arise in connection with utilization of its results for engineering needs.
Test description, data, results, curves and photos presented.

J-015
Johnson, C., Butzin, P.E.
Air Powered Hydraulics
Proceedings of National Conference on Industrial Hydraulics, V. 6, pp. 111-20, 1952

Comment on special application of O-rings in pairs, contrary to usual procedure, and "matching" them with oil used.

J-016
Johnson, C.
Dynamic Sealing With O-Rings
Mach. Design, V. 27, pp. 183-188, Aug. 1955

As static seals, O-rings have been almost universally successful. This is not true of dynamic applications. Most unsatisfactory installations can be traced to misapplication, improper design and maintenance.
Such problems are discussed together with O-ring idiosyncrasies.

J-017
Johnson, D.P., Bowman, H.A., and others
A Versatile Closure For High Pressure Vessels Utilizing O-Rings For The Initial Seal
I.S.A. Journal, V. 3, pp. 241-242, July 1956

A method of simplifying fabrication and assembly of Bridgman unsupported area closures by use of commercially available rubber O-rings.
(150,000 psi and above) Drawings.

J-018
Johnson, H.
Packing Kinks
Power Plant Engineering, V. 25, pp. 925-6, Sept. 15, 1921

RR-1923 - Vol. 27
1929 - Vol. 33

J-019
Johnson, M.E.
Stuffing Box And Expansion Joint
U.S. Patent 2,528,436, Oct. 31, 1950, to U.S. Atomic Energy Commission

This patent discloses a stuffing box and expansion joint assembly for a retort containing a rotating shaft arranged so as to permit longitudinal expansion and misalignment of the retort without interfering with the rotation of the shaft or disturbing the gas-tightness of the stuffing box.

J-020
Johnson, R.L., and others
Wear Of Materials For High Temperature Dynamic Seals
SAE Paper, No. 686, p. 7, Jan 9-13, 1956; also NACA - Tech. Note 3595, p. 22, Feb. 1956

Wear experiments with carbon seal materials sliding against various metals gave following results: wear was accelerated rapidly by operating at higher temperature levels; adverse effect of temperature on wear of carbon was minimized by use of chromium plate as mating surface. Application in aircraft gas turbine engines.

J-021
Johnston, W.L.
Home-Made Metal Packing Gives Four Years' Service
Power, V. 68, pp. 26-7, July 3, 1928

Split babbitt rings, each pair making a ring of square cross-section were used as packing. Description of assembly and operation. Drawings included.

J-022
Johnstone, Taylor, F.
New Metallic Gland Packing Gives Service
Power Plant Engineering, V. 30, p. 225, Feb. 1, 1926

Simple metallic packing specially suited for the rods of high speed steam engines.
At the base of the gland is a gun metal ring, followed by a ring of packing, a cast iron lantern bushing, two more rings of packing, a ring of soft asbestos, finally followed by the gun metal gland.
A second type considered is a spring loaded seal.
Drawings and description.

J-023
Jones, D.H.
Design And Application Of Controlled-Volume Pumps For High Pressure In The Range Of 10,000 to 30,000 PSI
A.S.M.E. Trans., V. 75, pp. 361-67, 1953

Seal abstract: Sealing of a reciprocating plunger. The most successful plunger-stuffing-box combination found thus far, Fig. 8, includes the use of a hardened plunger superfinished to 3 to 5 micro inches, with a special arrangement consisting of a soft metallic packing combined with other packing materials. Such a combination has proved satisfactory up to 30,000 psi. Drawings included.

J-024 T
Jones, W.E.
Sealing With Present Day Methods
Design Engineering, V. 4, pp. 41-44, March 1958

A brief discussion is presented on static, rotary, and reciprocating seals.

J-025
Jordan, J.
Extreme Temperature Sealing
Missile Design and Development, pp. 24-26, Jan. 1959

New method uses O-ring and retainers of dissimilar metals.

J-026
Jordan, J.
Thermal Expansion Of Elastomers
Mach. Design, V. 32, p. 144, Jan. 7, 1960

New design for improved seals of synthetic rubber.

J-027
Jordan, J., and McCuistion, T. J.
The In-Place Seal
Product Engineering, V. 31, pp. 68-72, April 18, 1960

In-place seals are static seals in which rubber or an elastomeric compound is molded directly in a groove of the metal seal part. Thus it fits the groove better and responds better than O-rings to the various pressures. A comparison of gaskets, O-rings, and in-place seals is given. Figures are given for resistance of elastomers to fluids, thermal expansion, and life expectancy as affected by operating temperature.

J-028
Jordan, J.R.
Sealing Problems
Oil and Gas Equip. Reprint, Dec. 1959

O-rings; their use in solving important sealing problems.

J-029
Jordan, J.R.
Select O-Ring Replacements With Properties In Mind
Plant Engineering, V. 15, pp. 112-114, Jan. 1961

J-030
Jordan, J.R.
Seals For Hard Vacuums
Machine Design, V. 33, pp. 134-139, May 25, 1961

A study of seal materials for pressures below 1×10^{-6} mm Hg. Materials must have low permeability for low molecular leakage. They must not volatilize and sublimize in hard-vacuum environments. The properties of butyl and viton synthetic rubbers seem most suitable for these applications.

J-031
Jordan, J.R.
New Developments In Vacuum Sealing
Am. Vac. Soc. 1961 Trans., Eighth Nat. Vac. Symposium, Second International Congress, V. 2, pp. 1302-1308

New data concerning efficacy of elastomeric seals in vacuum are presented. Leak rate data and sublimation effects are used to assess the utility of these seals in "ultra high" vacuum systems and in space. Leak rates at several temperatures, permeability, sublimation effects, outgassing, and accessibility are discussed.
Silicones, Polyurethane, Vitons, Nitriles, Chloroprene, and Butyl compounds are used. Data tables.

J-032
Justice, R.W., and Plumley, E.
Leather Packings For Hydraulic Roll Jacks
Iron & Steel Engr., V. 23, pp. 82-4, Sept. 1946, pp. 103-5, Dec. 1946

The change to leather V. packings and improvements obtained; advantages of V. leather assemblies; leather versus composition packings.

K-001
Kalk, W. A.
Design Criterion Of Plenum To Reflector Block Seal For H.T.G.R. Reactor Vessel
(General Atomic Div., General Dynamics Corp., San Diego, California) (GAMD 1376)
Nuclear Science Abstract #17294-Contract AT(04-3) 314 (16 p.), April 20, 1960

A survey was made of sealing methods and their adaptability to the requirements of the plenum reflector seal on the HTGR reactor vessel. Lap joints of steel to graphite or graphite to graphite with a 100 RMS surface finish met the required leak rate (43.8 pounds/hr. helium) and two basic concepts of this type seal are illustrated.

K-002
Kalmus, P.I.P.
A Simple Flexible Vacuum Joint
Vacuum, V. 9, No. 2, p. 147, May, 1959

The new type of vacuum joint appears to have several advantages over the bellows coupling, the principal asset being ease of construction. The joint consists simply of a separate flange of special shape and a cylindrical tube. The vacuum seal between these is accomplished by means of a soft rubber O-ring which also gives the coupling its flexilibity. The O-ring fits around a polished section of the tube and is kept under compression in a groove which has an outside diameter 0.012 in. smaller than the outside diameter of the ring. The width of the groove is 0.005 in. less than the thickness of the ring, and its depth is sufficient to prevent the ring being pushed inwards along the tube by external atmospheric pressure.

K-003
Karassik, I.J.
Know Your B.F. Pump Stuffing Boxes
Power Engineering, V. 60, pp. 70-72, May, 1956

The operation of boiler feed pump stuffing boxes becomes more complex as pressures and temperatures go up.
Pressures of 400 psi and temperatures of 400°F are found.
Basic concepts are discussed.

K-004
Karassik, I. J., Carter, R.
Centrifugal Pumps: Mechanical Stuffing Box Seals
Industry and Power, V. 64, pp. 94-95, June, 1953

Presents mechanics of the seals. Diagrams.

K-005
Karassik, I. J., Carter, R.
Mechanical Seals For Centrifugal Pumps
Heating and Ventilating, V. 51, pp. 92-94, Dec., 1954

Packing types and gland inserts. Photographs, diagram.

K-006
Kavanagh, W.
Fitting The High-pressure Cylinder Of An American Ball Engine With A Special Metallic Packing
Power, No. 81460C, Jan., 1907

Illustrates and describes a difficult application of metallic packing.

K-007
Kaye, J., E. C. Elgar
Modes Of Adiabatic And Diabatic Fluid Flow In An Annulus With An Inner Rotating Cylinder
Trans. Am. Soc. Mech. Engr., V. 80, pp. 753-765, 1958

First phase of investigation of heat transfer in the air gap of a rotating electrical machine.
A review of laminar flow, laminate flow plus vortices, turbulent flow plus vortices, and turbulent flow.
Experimental results in graphical form. Preliminary heat transfer data.
Discussion.

K-008
Kearton, W. J.
The Flow Of Air Through Radial Labyrinth Glands
I.M.E. Proc., V. 169, pp. 539-550, 1955

Theory has been worked out for inward and outward flow with pressure distribution expressions being derived.
Single ring and multi-ring glands were investigated as were unstaggered radial glands.

K-009
Kearton, W. J. and Keh, T. H.
Leakage Of Air Through Labyrinth Glands Of A Staggered Type
IME Proc., V. 166, pp. 180-188, 1952

Theory presented for flow of air through single orifice. Extended to cover labyrinth gland of several constrictions of equal area. Experiments made over wide range of pressure ratios to determine coefficient of discharge for single annular construction having a small clearance.

K-010
Kegley, T. M. Jr.
Examination of HRT O-Ring Gaskets From Flanges A-117 And D-127.
Oak Ridge National Lab. Y-12, Tenn.
Metallography Report No. 40, Contract W-7405-Eng-26, Jan. 14, 1957, CR-57-1-50

The results of an examination of O-ring gaskets removed from the HRT following a discovery that water in the leak detector system contained 1000 PPM chloride are presented.

K-011
Keller, C.
Flow Tests On Labyrinth Glands For Steam Turbines
Escher Wyss News, V. 7, No. 1, pp. 9-13, Jan./Feb., 1934, also Power Plant Engineering, V. 41, No. 4, pp. 243-245, April, 1937

Development test made by Escher-Wyss Company on steam turbine labyrinth glands. Flow patterns were photographed using light metal particles which reflected light through glass observation panels.

K-012
Keller, G. R.
High Temperature Hydraulics
Mach. Design, V. 28, No. 16, pp. 121-24, August 19, 1956

Considerations when designing for high temperature hydraulics. Fluids, synthetics, compounded diesters, silicone fluids, Buna S base O-rings, neoprene seals, compression set, back-up rings, springs, structural metals, filtration. Discussion - no test data.

K-013
Keller, [illegible]
Optimizing Hydraulics For High Temperature Operation
Western Aviation, V. 37, p. 913, December, 1957

Discussion of the optimization of hydraulic systems to fit the increasingly high temperature conditions under which aircraft and missiles are being operated.

K-014 B
Keller, G. R.
Design Of Seals And Sealing Devices
Chapter 12 of book, "Aircraft Hydraulic Design," Publ. by Appl. Hyd., 1957

K-015
Keller, G. R.
Overcoming Some Of The New Problems In Hydraulic Control
Automatic Control, V. 7, pp. 38-41, July, 1957

Areas which will require special attention by the systems designer are outlined in this article. Temperatures, dynamics of hydraulic system, fluids, radiation, and some of the new seals available are discussed.

K-016 B
Keller, G. R., Mayhew, W. E., Cannon, C. H.
High Temperature Hydraulic And Pneumatic Equipment
I. Future Aircraft Power Systems, Gas Or Liquid
II. Hot Hydraulic Systems
III. Hot Pneumatic Systems
Control Engineering, V. 6, No. 4, pp. 89-104, April, 1959

I. Bases for choice, viscosity factor, materials and components
II. Fluids, components, servovalves, seals, summary
III. High temperature design problems, fillings, and seals, actuator, valves, spool valves, safety
Recommendations

K-017
Keller, G. R. and Stafford, P. H.
Metal Dynamic Hydraulic Seals
SAE Jour., V. 65, No. 6, p. 78, May, 1957

Testing and evaluation program undertaken at Autometics, Div. of North American Aviation, Inc. to study performance of metal dynamic seals for use in missiles or aircraft hydraulic equipment in lieu of elastomer seals, which are unsatisfactory at high speeds, high temperature; tests at room temperature of seals of various configurations, material finishes, etc. Shaft and piston seals tested at oil temperatures from 500-550°F and diameters ranging from 1/2 to 3-3/8 inches.

K-018
Kelley, S. G.
Gasket Materials: Their Properties And Uses
Materials & Methods, V 36, pp. 108-112, Aug., 1952

Available to meet the many different service conditions under which gasketed joints must function are a wide assortment of synthetic and natural materials. Among these are rubber, both natural and synthetic, cork, plant fibers, asbestos fibers, metals, plastic, leather, and felt. Properties of each are discussed.

K-019
Kenyon, R. L.
Pressure - Energised Seals
Rocketdyne, Division of N.A.A., July, 1960

Problems associated with static seals, and seal configuration.

K-020
Kephart, R. C.
High Pressure Steam Joints And Gaskets
Am. Soc. Nav. Engrs. Jour., V. 46, pp. 100-106, Feb., 1934

K-021
Kerkhof, W. P.
New Stress Calculations And Temperature Curves For Integral Flanges
Proc. 3rd World Petroleum Congress, 1951

K-022
Kiergan, N. B.
Aircraft Hydraulic Systems
P.B. 16819 - Off. Tech. Ser. (2 pp.), Oct., 1945 (Japanese) (AAF Am. Tech. Intelligence Group Report 18)

Interrogation of Takeo Doi, Chief Engineer of Kawasaki Aircraft Co. covered the problems of leakages and packing problems, power source, standard mineral fluids and equipments with which hydraulics were operated. Soy bean oil or rapeseed oil added to hydraulic control oil reduced the swelling rate of rubber parts to one half. Silk packings soaked in polyvinyl alcohol and metapecryl resin had good quality.

K-023
King, P.
Maintenance Of Metallic Packing On Locomotives
Am. Engr. & RR Jour., pp. 178-179, V. 78, May, 1904

Discusses the treatment packings should receive.

K-024
King, L.D.P.
High Vacuum Seal
Re. Ser. Inst., pp. 83-84, Feb., 1948

A sample two way valve and method of making a seal are described, both well adapted to large or small port openings in high vacuum work.

K-025
Kitts, F.G.
Development Of A Rubber For Service In Contact With Experimental Hydraulic Fluids At 400°F
PB 111598, Off. Tech. Sev. (15 pp.), Dec. 1954, CAA

A compound of neoprene WRT was developed which was marginally satisfactory after aging 168 hours in MLO-8200 at 400°F in absence of air. If air were not at least partially excluded the rubber would reach unsatisfactory conditions in less than 70 hours. O-rings have been fabricated and are undergoing evaluation tests. (AAF WADC-54-458)

K-026
Klein, Wilhelm
Centrifugal Pump With A Stuffing Box Relieved From Pressure
L. C. or S.L.A. Trans. 59-12619

K-027 T
Kluge, F. H.
Rotating Shaft Seal Means
U.S. Patent 2,621,087, Dec. 9, 1952

Relates to a buffered shaft seal using carbon rings. A slinger ring throws liquid away from the shaft near the housing and is backed up by three sets of carbon seal rings. Between the first and second set, a neutral gas is introduced at a pressure greater than the sealed pressure causing flow inward and outward. Between the second and third rings an outlet is provided for that portion of the neutral gas which flowed outward. This outlet may be kept at a pressure lower than atmospheric to prevent escape of the neutral gas to the atmosphere.

K-028 - T
Klupgrl, A.
Contactless Oil Film Shaft Seal For Hydrogen-Cooled Generators (in German)
German Patent 1,012,999, August 1, 1957

A bushing seal made of two axially contacting rings with a central supply groove for the buffer fluid. Clearance on process side of ring is made smaller than atmospheric side, thus reducing the flow of oil toward the process side. The seal ring incorporates discontinuous grooves which may be situated at an angle.

K-029 B
Knight, E.J.
Pneumatic Motors For Aircraft Actuation Systems
Aero/Space Engineering, V. 17, No. 12, pp. 44-7, Dec. 1958

Discusses the construction and performance of various pneumatic actuators. Special attention is given to materials and seals.

K-030
Knights, H. C.
Improvements In Or Relating To Sealing Devices For Tubular Members
British Patent 877,006

A sealing device is designed for effecting an endwise seal between two tubular members of different diameters.

K-031
Knudsen, A. W.
Metallic Vacuum Tight Gasket
Review of Scientific Instruments, V. 23, pp. 566-6, Oct., 1952

The unique properties of indium metal make it very well suited to the problem of a non-organic vacuum-tight gasket. Sheet stock 0.007 in. thick. Detail discussion of mounting procedure and recommendation.

K-032
Koch, H. G.
Engineering Studies In Packings
Paper Tr. J., V. 126, pp. 27-31, Jan. 1, 1948, Paper Ind., V. 29, p. 1494, Jan., 1948

K-033
Koch, R. H.
Packing And Gasket Materials
Machine Design, V. 28, pp. 117-120, May 31, 1956

Requirements for heat and cold resistance, chemical resistance, and life are tightening up a continuing search for more effective sealing materials. Recent attention has been focused on fluorocarbon plastics and new type elastomers which are discussed in this article.

K-034
Koch, R. H.
Improving Performance Of Packings And Gaskets
ASME Paper N57-PET-33, p. 15, Sept. 22-25, 1957

Practical aspects of maintenance problems and corrective actions that can be taken to minimize costs and unscheduled equipment shutdown; material, environment, and application which influence performance; how their effects are analyzed; particular reference to chemical and petroleum processing industry.

K-035
Koenig, T.H., Vargo, J. E.
Silicone Lubricant - Carbon Seal Interaction At Elevated Temperatures
WADD Tech. Report 60-757

Static immersion tests using 18 grades of carbon in combination with three silicone oils, one ester, and one silicone-ester blend at temperatures of 400°F, 500°F, and 550°F. A friction test rig has been constructed that can vary load, speed, and ambient temperature at the test section. Test run with QF6-7012 silicone oil at 3000 rpm, 45 psi face pressure, and ambient temperature.

K-036 B
Komotori, K.
Flow Observations In The Labyrinth Packing
Proc. Fujihara Memorial Fac. Engg. Keio Univ. (Japan) V. 9, No. 3, pp. 33-41, Oct. 1956 - in English

The flow patterns in the labyrinth packings were investigated, based on flow observations which were carried out by using a "smoke-tunnel." Moreover, these results are discussed in reference to my experimental data on the leakage loss through the labyrinth packing which was previously reported.

K-037
Kosting, P. R.
Neoprene Packings
P.B. 25428 Off. Tech. Sev. (6 pp.), Dec., 1941 (Watertown Arsenal Lab. Rept. 750/7)

Tests were made in order to determine the pressure which is exerted outwardly against the wall of a recoil cylinder when a backing ring made from neoprene is subjected to a given lateral pressure. The load, movement of the cylinder and diameter of the ring were observed. It was concluded that the same unit pressure is considered to be exerted outwardly against the cylinder wall as is applied laterly to packings of neoprene.
Diagrams.

K-038
Kotelewsky, G.P.
Gasket Reactions And Sealing Pressures In Vessels
Ind. Eng. Chem., V. 51, pp. 949-51, Aug. 1959

A method for calculating the maximum sealing pressure for gaskets in high-pressure vessels is developed by analyzing the forces among flanged connections, loaded bolts and gaskets.

K-039
Kraus, R.A.
Modern Sealing Devices For The Steel Industry
Iron & St. Engr., V. 26, pp. 53-64, Sept. 1949

Rotating shaft seals are discussed for a variety of application: pumps, gear reduction cases, machine tools, electric motors, etc.
Materials; flax, jute, cotton, rubber and several metallic materials.
Buna N is used for certain specialized applications.

K-040 B
Kronberger, H.
Vacuum Techniques In The Atomic Energy Industry
Proc. I. Mech. E., V. 172, pp. 113-24, disc., pp. 125-32, 1958, (abbreviated) Engineer, V. 204, 5312, pp. 702-5, Nov. 15, 1957

The development of atomic energy has stimulated a rapid development of industrial high-vacuum technology. Much attention to design details of welded and flanged joints. Joints and flanges - examples of good and bad of welding practice. Flanged joints for liquid metal, high pressure, high temperature.

K-041
Kruesi, A.H.
Fittings For Superheater Steam
West Electn., V. 41, p. 450, Dec. 7, 1907

The most satisfactory gaskets are of thin sheet steel not more than 5/8 in. in radial width, so that heavy pressure can be brought on it with bolts and flanges of ordinary weight, the steel being soft annealed so that it can be pressed into the rough surfaces of abutting flanges.

K-042
Krug, L.G.
Bearing Seals
Iron & St. Engr., V. 23, pp. 77-82, Nov. 1946

Steel mill bearings need complete protection against the entry of water and scale and loss of lubricant. This paper concerns these steel mill seals. 800 psi pressure and 240°F operating norms.

K-043
Kuchler, T.C.
Dynamic Sealing: Theory And Practice
Koppers Co., Inc., Baltimore, Md., Feb. 1961

K-044
Kuchler, T.C., and Pearson, W.
Clearance Seals
Machine Design, The Seals Book, pp. 20-23, Jan. 19, 1961

Clearance seals limit leakage by closely controlling the annular clearance between the rotating or reciprocating shaft and the relatively stationary housing.
Types:

1. Labyrinth - one or more thin strips or knives attached to either the stationary housing or rotating shaft.
2. Bushing or ring - close fitting stationary sleeve within which the shaft rotates.

Problems such as leakage and rubbing are discussed.

K-045
Kuech, W.
Recent Tests Of Glasses And Sealing Material For Pressurized Cabins
Lilenthal-Gesellschraft Fue Luftfahrtforschung Bericht-129, p. 30
Spec. Lib. Assn. Translation 3221, (13 p.), V. 3, No. 8, Aug. 1957

K-046
Kuhn, H.S.
Rotating Joints
Prod. Engng., V. 27, pp. 200-204, Aug. 1956

Rotating joints, originally designed around familiar packing gland, have become highly specialized for many applications in all industries. They now operate from - 300°F (liquid oxygen) to 700°F (dichlorobenzene) or high temperature superheated steam. They seal water, steam, brine, air, paint, lacquers, refrigerants, diphenol-diphenoloxides, dichlorobenzene, sea water, and air-mud mixtures. The most practical seal material is carbon with stainless steel, monel and high nickel steel mating surfaces.

K-047
Kunkel, E.V.
Selecting Gaskets To Limit Corrosion Of Stainless Steel Bolted Joints In A Chemical Plant
Corrosion, V. 10, pp. 260-266, Aug. 1954

Examples of corrosion of austenitic stainless steel joint surfaces attacked by liquid media penetrating gaskets are described. Author postulates liquids of high electrical conductivity, containing weak oxidizing and reducing agents are principally responsible for the pitting attack of the gasket bearing surfaces. Presence of chloride salts or organic chloride in the gasketing may contribute to stress corrosion cracking of the joint surface. Author concluded corrosion may be controlled by proper selection of gasket material, by proper joint design and by modification of corrosion characteristics of process fluid being handled.

K-048
Kupiec, H.P. (Aircraft Equip. Testing Co.)
Testing Of Metal Boss Seals
PB-119818, Off. Tech. Sev. (67 p.) photo, dia. dwg., April 1955, Order from L.C.

The metal boss seal was conceived by Wright Air Development Center to meet the requirements of hydraulic and pneumatic systems with operating pressures up to 5,000 psi, and temperatures as low as -100°F and as high as 600°F. The application of the new metal boss seal involves the use of a deformable metal ring in conjunction with standard AN hydraulic fittings. This seal is being considered as a replacement for the current standard ANG6290 synthetic rubber gasket. Proj. No. 1371 AF WADC Contract AF33 (600) - 26548.

K-049
Kunze, E.J.
Diesel Engine Upkeep
Power Plant Engineering, V. 36, pp. 601-2, August 1, 1932

Maintenance of packings, gaskets, and engine plant joints. Proper selection of packing materials. Joint types. Proper methods of using gaskets.
Common packing materials are asbestos, cotton, flax, fiber hemp, graphite, rubber lead, babbitt, antimony, aluminum, copper bronze, and cast iron. Common gasket materials are paper, leather, cotton fiber, cork, lead, aluminum and copper.

K-050
Kurek, J., and Simmons, J.
Development Of Secondary Seal For PWR Head Penetrations
Combustion Engineering, Inc., Nuclear Components Engineering Dept., Chattanooga, CENC-1006 CE
Contract Nos. 7954-25 and 7954-30, Aug. 1957, 66 p.

The results of the development program which was initiated to design a seal to back-up the primary seal of the pressurized water reactor are presented. This program was divided into three phases: a packing, a welded membrane, and a metallic gasket.

K-051 B
Kurie, F.N.D.
Vacuum Systems, Seals And Valves
Review of Scientific Instruments, V. 19, pp. 485-93, Aug. 1948

Design suggestions are given for valves, shaft seals, and gasketed joints intended for use in continuously pumped all-metal vacuum systems.
Vacuum valves, gaskets, O-rings.

K-052
Kuskevics, G., and Chappelle, A.
Low Temperature Vacuum Systems
R57AGT751, GEL Tech. Data Center, G.E.Co., Sch'dy., N.Y. (13 p.), Dec. 13, 1957

Vacuum systems operating below 90°K are reviewed. These include storage and transport equipment for liquified gases and cryogenic optical test equipment. A vacuum system for a liquid oxygen flow apparatus and a liquid nitrogen optical dewar are described in detail.
Among materials used for low temperature vacuum system construction, copper, brass, monel, and Kovar austenitic stainless steel are common. All are non-ferromagnetic. They are either soft or silver soldered, arc-welded or brazed. Rubber, grease, and sealing compounds cannot be used because they are hard at low temperatures. Hence only metal O-rings such as lead or solder can be used. Low temperature differential, contraction problems must be considered.

K-053
Kuss, E.
The Physics Of Extreme Pressures (In German)
Chem. Ing. Tech., V. 28, pp. 141-52, March 1956

A comprehensive treatise covering autoclave, gasketing and sealing, pressuring, measurements, equations of state and internal.

L-001
Labber
Cutting Copper Gaskets
Eng. & Mining Journal, V. 137, p. 189, April 1936

L-002
LaBour, H.E.
Improvements In Or Relating To Rotary Shaft Seals And Pumps Provided With Such Seals
British Patent 854,526, Nov. 23, 1960

A seal for the vertical rotary shafts of centrifugal pumps is designed so that the pump can be used for long periods without maintenance other than oiling the motor or shaft.

L-003
Labrow, S.
Design Of Flanged Joints
Institution of Mechanical Engineers Proceedings, V. 156, pp. 66-70, 1947, (discussion, pp. 70-73)

Basis of design adopted by one firm to insure consistency over a wide range of pressures and sizes. Table on gasket factors and yield values for various materials including metals, asbestos rubber, and fabrics, and one on effective gasket width.

L-004
Lange, W.J., Alpert, D.
Step-Type Demountable Vacuum Joint
Review of Scientific Instruments, V. 28, p. 726, Sept. 1957

Flanges are machined from stainless steel and are heliarc welded to vacuum system components. The gasket is a washer-shaped ring of O.F.H.C. copper about 0.040 in. thick. It is hydrogen annealed at 950°C after cutting. This flange has the advantages over knife edge type seals in that the steps are much more easily machined and the tolerances are less stringent. Advantages discussed, drawing included.

L-005
Lankester, J.A.
Bonded Seals
Appl. Hydraulics, V. 7, pp. 69-71, May 1954

Designers of hydraulic components, both aircraft and industrial, have sought a means of providing a reliable and durable static seal. The bonded seal is a result of this search. Applications discussed.

L-006
Larkin, R.G.
Evaluation Of High Temperature Hydraulic Seals To Temperatures Of 550°F. Part II Polymer Compounding And Evaluation
WADC Tech. Report 57-76, Feb. 1958

L-007
Larson, A.W.
High Pressure O-Ring Piston Packing
Res. Sci. Inst., V. 25, p.1136, Nov. 1954

The application of multiple O-rings to a piston to prevent leaks at 10,000 atmospheres for long periods of time.

L-008
Leach, L.L.
Seals For Shaft Housings And Bearings
Product Eng., V. 3, pp. 123-4, March 1932

Typical seal designs suitable for three general purposes: (1) to retain lubricant; (2) to keep out foreign matter and; (3) to prevent leakage of liquids or gases through clearance space between rotating shaft and housing.

L-009 T
Ledwith, W.A.
Bearing And Seal Assembly For Turbines
U.S. Patent 2,469,734, May 10, 1949

A bearing and labyrinth seal assembly designed to be installed as a unit.

L-010
Lee, J.
Metallic Seals For Extreme Temperature-Pressure Applications
SAE Journal, V. 70, No. 10, pp. 98-99, October 1962

Wedge seals fabricated from recently developed materials have been tested at Republic (Republic Aviation Corp.) to 1000°F and 400 psi for periods up to 17 hours with negligible leakage. Description of several types, with drawings, photos, and results discussion.

L-011
Lehmberg, W.H.
Plain And Laminated Felt Seals: Their Design And Application
Prod. Engng., V. 17, pp. 209-211, March 1946

Design properties of felt seals when used for retention of grease and oil and exclusion of dust fume or moisture are presented in tabular form. Selection of the correct grade in either plain or laminated felt seals is discussed as well as recommended designs for housings.

L-012 T
Lein, J.
Mechanical Test On Sealing Rings For Rotary Shafts (In German)
Konstruktion, V. 6, No. 10, pp. 384-389, Oct. 1954

A discussion of the results of an investigation into rubber and leather shaft seals conducted over a period of years. The paper covers a wide range of design considerations and presents a thorough discussion of factors affecting seal operation. The investigation was conducted at the Automotive Research Laboratory of the Karlsruhe Institute of Technology.

L-013
Lein, J.
Mechanical Tests On Rubber Sealing Rings For Rotating Shafts
Engrs. Digest, V. 16, pp. 59-61, Feb. 1955

Sealing rings made of synthetic rubber, for sealing the interior of a housing against leakage along a rotating shaft, have proved successful even at high rotational speeds, and with differences in inside and outside pressures.

L-014
Lem, D.R., Reynar, J.M.
Stress Relaxation Of Non-Metallic Gasket Material
Bull. Am. Soc. Test Mat. (ASTM), No. 207, pp. 81-85, 1955

A functional test fixture, a testing procedure, and a formula by which numeric values can be obtained for percent stress relaxation for all types of non-metallic gasket materials has been established.
Test data, results, and discussion.

L-015
Lenhart, W.B.
Ball Mill Gaskets
Rock Products, V. 45, p. 55, Sept. 1938

L-016
Leont'eva, V.P., and Slonimiskii, G.L.
A Study Of Stocks Used For Seals In Units Of Hydraulic Systems Of Airplanes Under Isothermal Compression From All Directions. (In Russian)
Moskov, Aviation Inst. im S. Ordzhonikidge, Sbornik Slater, V. 93, pp. 5-20, 1957, (LC or SLA-Translation No. 59-19477)

Compressibility of cured stocks of nitrile rubbers (natural rubber and SKBS and SKB) was studied under hydraulic pressures of 32-500 atm. at 18-20°C.

L-017
Le Pera, A.A.
Sealing Fluids In Propulsion Systems
SAE Jour., V. 67, pp. 118, 121, Aug. 1959,
Abstract of SAE paper 50U, "Fluid sealing in extreme environments," from meeting of March 31, 1959, 11 p.

Discusses fluids in which propulsion system static and dynamic seals must operate. Elastomer compatibility studies have been made with these fluids which include high energy fuels, extremely corrosive oxidizers, mono-propellants and Cryogenic fluids.

L-018
Le Pera, A.A.
Breakthrough Needed For Extreme Environment Fluid Sealing
Space/Aeronautics, V. 32, No. 2, pp. 50-53, Aug. 1959

Sealing problems on operation from - 400 to + 1000°F are discussed. A breakthrough may come from polymer technology or from heat-resistant elastomeric-inorganic composites.

L-019 T
Lewis, D.R.
Mechanical Seal Failures
Power, V. 89, p. 918, Mid-December 1945

This is an abstract of a paper presented at the ASME petroleum session.
A discussion of failures in mechanical seals, analysis of failures, and remedies, are included.

L-020
Lewis, D.R.
Mechanical Seals, Basic Considerations - Part I
Mach. Design, V. 18, pp. 125-128, Sept. 1946

Comparative merits of stuffing boxes and mechanical seals are discussed along with design considerations of each.

L-021 T
Lewis, D.R.
Mechanical Seals: Classification Of Types
Machine Design, V. 18, pp. 78-82, Oct. 1946

Basic definitions, and photographs of rotating joints, stationary joints, unidirectional forces related to inside assemblies and outside assemblies, single seal, and double seal. Types of seals: packing ring, gasket, helical spring; integral gasket, helical spring; bellows, integral gasket, helical spring, strained member bellows.
Speed, temperature and pressure ranges of each.

L-022
Lewis, D.R.
Mechanical Seals, Application Factors - Part IV
Mach. Design, V. 18, pp. 138-142, 182+, Nov. 1946

The three fundamental rules used in designing mechanical seals are discussed. They are: (1) Life of mechanical seal directly proportional to shaft rigidity. (2) Means should be provided to fasten stationary seal member to machine so face remains flat and normal to axis of rotation. (3) Allow for maximum ease of seal installation, inspection, and maintenance.

L-023
Lewis, D.R.
Mechanical Seals, Theoretical Design - Part IV
Mach. Design, V. 18, pp. 146-150, 184, Dec. 1946

Article relates the heat producing factors in a seal design to see how they can be controlled to minimize wear.

L-024
Lewis, J.W.
An Experimental Study Of The Motion Of A Viscous Liquid Contained Between Two Coaxial Cylinders
Proc. Royal Soc. London, Series (A), V. 117, pp. 388-407

A detailed report of an experimental investigation of the validity of G.T. Taylor's stability criterion for transition from laminar flow to vortex flow. Tests were run from kinematic viscosities varying from 0.006 to 0.018 CM^2/sec. with varying clearances to radius ratios. Photographs of flow patterns are included.

L-025
Lewis, O.L.
Flange, Gasket, And Bolting Materials For Refinery Service
Oil and Gas Journal, V. 49, p. 134, Aug. 24, 1950

L-026 T
Leyer, A.
Wedge Lubrication Or Pressure Lubrication (In German)
Schweizerische Bauzeitung, V. 71, pp. 66-69, Jan. 31, 1953

An analysis is made of the viscosity pump, and the maximum efficiency of such a pump is shown to be 33%. Hydrostatic lubrication is also considered briefly.

L-027
Li, Y.T., Lapp, P.A., Lee, S.
A Flowmeter For Measuring Mass Flow-Rate With A High Speed Of Response
Proc. National Conference on Industrial Hydraulics, V. VII, pp. 218-231, 1953

Includes a brief description of rotary seals used in the instrument.
Rigid perfect seal; flexible perfect seal; clearance seal, plate-type seal, labyrinth seal, mercury or water seal.

L-028
Liberto, R.R.
Research And Development On The Basic Design Of Storable High-Energy Propellant Systems And Components
Bell Aerosystems Company, pp. 1-281, May 19, 1961 (AFFTC TR 60-61)

Includes metals, elastomers, plastics, etc.
Effect of temp., time, pressure, etc.
Seals, materials.

L-029
Lighter, A.W.
Magnetic-Type Mechanical Seals
ASME preprint 59-PET-35, 2 pp., Sept. 1959

Dynamic shaft seal developed to eliminate springs in mechanical seals. Consists of an Alnico 5 magnet ring, rhodium plated for corrosion protection, two O-rings, one stationary and one rotating, and a carbon ring unit.
All types of fluids can be sealed except light, non-lubricating hydrocarbons at high pressures. These produce excessive wear in the carbon. Tests indicate that this carbon wear problem can be solved.

L-030
Likharev, K.K.
Constructional Features Of The Seal Between Parts Moving Relative To One Another In High Pressure Apparatus
Order from L.C. or SLA Translation No. 60-15806

L-031
Lillis, S.M.
Engine Oil Seals Improved To Meet Truck And Bus Needs
SAE Jour., V. 68, pp. 86-87, No. 3, March 1960; from SAE Paper N119T, 1959

New materials, methods, and machinery are being used to make oil seals for bus and truck engines. To illustrate two examples are cited; front and rear crankshaft oil seals.

L-032
Linderoth, L.S., Jr.
Selecting Hydraulic Seals
Mach. Design, V. 16, pp. 119-128, Sept. 1944

How well a given hydraulic packing installation achieves its main purpose of confining hydraulic fluid within the components of the system is a measure of its efficiency. Secondary considerations are friction, life, and ease of service or repair. Frequently it will be found that several different designs of hydraulic packings are equally effective. Article discusses briefly a number of generally accepted designs and related engineering factors.

L-033 T
Lindhardt, P.
The Hydrostatic Gas-Bearing As A Non Touching Gland For Rotating Shafts
Int. Conf. on Fluid Sealing, Paper C2, (8 pp.), April 17-19, 1961

The possibility of maintaining a uniform gap between the sealing faces of a radial face seal by the introduction of a series of pressurized pockets in the stationary face is examined. Expressions giving the gap thickness and condition for high axial stiffness in terms of the sealed pressure and seal geometry are derived, and preliminary results have shown that the seal is capable of operating satisfactorily.

L-034 T
Lindow, A.L., Graff, R.A.
Fluid Seal
U.S. Patent 2,960,332, Nov. 15, 1960

A packing assembly which is capable of self-sealing, automatically repairs damage to the sealing surface. The patent relates to seals for which relative motion is limited such as in fluid springs.

L-035
Linke, J.
Flange Seals For Low-Temperature Fluids
V.D.I. Zeitschrift, V. 98, p. 1614, September 1956 (In German)

L-036 T
Llewllyn, W.A., Mealing, J.E.
Development Of Carbon Sealing Ring Materials
Int.Conf. on Fluid Sealing, Paper F4 (14 pp.), April 17-19, 1961, British Hydromechanics Research Assn., Harlow, Essex, England

Discusses amorphous carbon and carbon graphite mixtures developed for sealing rings. Physical, chemical, and running characteristics, wear test results. Running characteristics of carbon graphite as face seals and counterface seals. Sintered metal-carbon-graphite seal face for operation under abrasive conditions.

L-037
Lobanoff, V.
Stuffing Boxes For Centrifugal Pumps
Petroleum Engr., V. 22, pp. C33-4, C36, March 1950

Function of stuffing box is to prevent leakage along shaft; usually, packing or mechanical seal minimizes leakage rather than prevents it completely; majority of high speed centrifugal pumps are still equipped with stuffing boxes fitted with pliable packing as sealing medium; types of stuffing box design.

L-038
Logan, S.E.
Static Seal For Low Temperature Fluids
Jet propulsion, V. 25, pp. 334-340, July 1955

Novel temperature sensitive seal developed by GE Co., consists of a neoprene gasket backed by an Invar insert and the gasket stress increases as the temperature drops and is independent of the original load used for assembly. This system is being investigated for static and dynamic cryogenic sealing down to -452°F.

L-039
Loggie, J.J.
Controlled Confinement Sealing
Hydraulics & Pneumatics, V. 15, pp. 90-91, March 1962

These seals are a valuable development for sealing exotic fluids at high temperatures in aircraft, missiles, and rockets.
Discussion of design factors; forces, time vs. temperature, and compatibility.
Silicone, Viton, buna N.

L-040
Longstrech, C.
Metallic Packings: Principles Of Design Essential For Successful Development
Mav. Engrs., May 1903

Discusses the purposes and requisites of a successful packing, importance of care in design, etc.

L-041
Longstreet, G.B.
Packing Centrifugal Pumps
Power, V. 74, p. 425, Sept. 22, 1931

A brief commentary on improper installation of proper size and material of packing, its effect and correction.

L-042
Lopata, S.L.
Practical Approach To Packing Of Chemical Pumps
Chem. & Met. Engr., V. 51, pp. 104-7, Dec. 1944

A discussion of the characteristics of the various yarns and lubricants employed in packings and how the packings should be installed and explains how to determine when it should be replaced.

L-043
Loricks, F.I., Jr.
Unique Sliding Seal For Vacuum Chamber
Rev. Scientific Instruments, V. 28, pp. 468-9, June 1957

A scattering chamber having a continuously variable angle of more than 180° between the entrance and exit beams, and having an exit port of sufficient diameter to accommodate a photomultiplier tube was required. The vacuum seal consists of a flexible strip sliding over an O-ring. A 5 mil strip of type 304 stainless steel, with angle stiffness of 0.010, and a 0.139 inch diameter O-ring was used. Details of assembly, and operation are given.

L-044
Loringren, H.E.
Flange Design Considerations: Flanges And Full Faced Gaskets
Petroleum Refiner, V. 27, pp. 117-19, Feb. 1948

L-045
Lucas, L.R., Hernandenz, H.P.
Inflatable Gasket For The 72 In. Bubble Chamber
Review of Scientific Instruments, V. 30, p. 941, Nov. 1959

A satisfactory glass-to-metal seal at liquid hydrogen temperatures has been developed for the large oval shaped optical window of the 72 in. liquid hydrogen bubble chamber. Indium wire is held in contact with the chamber glass by an inflatable stainless steel member capable of 0.160 in. useful deflection. Sealing technique described in detail. Pressures and temperatures discussed. Photograph included.

L-046
Lutz, M.W.
5 Kinds Of Stuffing Boxes For Barrel-Type B-F Pumps
Power Engng., V. 61, pp. 73-76, Sept. 1957

Design and maintenance of stuffing boxes, particularly in boiler feed pumps, has long been a problem for pump manufacturers, as well as for central station operators. Wide range of suction conditions up to 500 psi; speeds up to 9000 rpm or higher and temperatures above 400°F makes the choice of the correct type of stuffing box very important. Knowledge of merits or disadvantages of each type of stuffing box will permit a more intelligent selection of the proper type.

L-047 T
Lyle, C.A., Muller, R.H.
Sealing Means For Rotary Regenerative Heat Exchanger
U.S Patent 2,942,857, June 28, 1960

A radial type labyrinth seal is used to minimize leakage in a rotary heat exchanger.

L-048
Lymer, A.
Mechanical Sealing As It Stands Today
Engng., V. 191, pp. 572-573, April 1961

The several theories of the basic sealing mechanism are explained. Basic seals (balanced and unbalanced) are covered as are the pressure ranges of each (unbalanced up to 150 lbs. per sq. in. and balanced up to 1200 lbs. per sq. in.). Specific applications and future developments are discussed.

L-049 B
Lyon, G.T.
C.N. Metal To Metal Seals End Fluid Leakage In Hydraulic Systems
Proc. 13th National Conference of Industrial Hydraulics, V. 11, pp. 211-4, 1957

L-050
Lyubimov, M.L., Shakhov, K.P., Yukhvidin, Y.A.
Experience In Designing And Manufacturing All-Metal Vacuum Systems
Vacuum, V.9, No. 2, pp. 108-116, May 1959

Late years have seen a clear-cut tendency to replace glass vacuum systems by all metal systems for an obvious reason. We have been working in this direction for the last ten years and the main results of our work form the subject of this report. The subjects dealt with are: (a) the design of demountable, high-vacuum heated joints, (b) design of valves, (c) design of traps.
Types of seals are: (a) conical seals, (b) seals with groove-wedge profiles, (c) seals with a rectangular butt projection, (d) pyramidal seals for rectangular cross sections, (e) seals with a double profile, (f) diffusion seals. Materials: oxygen free copper, annealed in hydrogen; aluminum, gold plated copper gaskets with diffusion seals.
Pictures and drawings.

M-001
Mabbs, J.W.
Packing
American Water Works Assn. J., V. 12, pp. 295-6, Nov. 1924

M-002
MacDonald, K.
Carbon Packing Progress For Sealing Small Turbines
Power, V. 98, pp. 122-123, 200+, July 1954

Article discusses a variety of carbon ring packing glands for turbines together with installation and maintenance problems.

M-003 B
MacDonald, R.J.
Development Of Iron-Base Seal Materials For High Temperature Applications
A.S.L.E. Paper 60LC-11, (8 pp.), October 1960, (Also Trans. A.S.L.E., V. 4, No. 1, pp. 12-19, April 1961).

The results of a series of experiments on the rubbing properties of commercial, high speed, tool steels are described and were obtained for surface speeds between 16 and 150 ft./sec. with ambient temperatures between 75 and 1200°F.

M-004
Machen, J.R.
16 Unusual Applications For The O-Ring
Product Engineering, V. 30, pp. 90-91, Nov. 23, 1959

This handy little component finds a place in pumps, drives, glands, shock mounts, pivots, knobs, valves and seals.
Drawings of 16 examples, with appropriate legend.

M-005 T
Macks, E.F.
Seal
U.S. Patent 2,907,594, Oct. 6, 1959

A cartridge type mechanical face seal which is claimed to provide sealing under extremes of temperature and pressure. Orifices are provided in one face of the seal to permit a pressurized lubricant supply to the sealing film between the seal faces.

M-006
Main, M.M., and Stubenrauch, E.H.
Simple Compression Packings
Mechanical Design, The Seals Book, pp. 51-57, Jan. 19, 1961

Compression packings create a seal by being squeezed between the throat of the box and the gland. Under these compressive forces, the packings flow outward to seal against the bore of the box and inward to seal against the moving shaft or rod. Compression packings require frequent gland adjustments to compensate for wear and loss of volume.
Packing materials include vegetable fibers, mineral fibers, animal products, artificial fibers and metals. Packing lubricant criteria are discussed.

M-007 T
Maisch, O.
Rotary Shaft Seal
U.S. Patent 2,899,244, Aug. 11, 1959

A form of mechanical face seal in which leakage increases the force on the seal.

M-008 T
Malanoski, S.B., Fuller, D.D.
Survey Of Component Requirements And Availability For Gas-Cooled Nuclear Reactor Power Plants - Bearings & Seal
The Franklin Institute, Report I-A2392-8, USAEC Contract AT (30-1) 2512, June 1961

This report is the result of a survey and critical review of the bearings, seals, and bearing and seal systems designed for use in the primary loop components of four gas-cooled nuclear reactors under construction in the United States.

M-009
Manes, J.
Leakage Test On One Voi Shan Test Block N.V.S.D. 1000-8
Wyle Lab., July 1961

Seals, configuration, and testing leakage of material fila burner.

M-010
Mann, J.B.
Improved Metal Gasket For Vacuum Systems
Rev. Sec. Instr., V. 27, pp. 1083-4, Dec. 1956

A hydrogen-annealed copper ring with a 90° triangular bead which presses against the flat flange surface opposite a cylindrical bead on flange surface.
Short article and detail drawing.

M-011
Marker, R.C.
Vacuum Techniques And Components Used For A Continuously Pumped Linear Electron Accelerator
Vacuum, V. 9, No. 2, pp. 128-133, May 1959

A vacuum system has been designed to continuously pump a linear electron accelerator having an internal volume of approximately 30 ℓ and produce vacuum down to 1×10^{-8} mm. Hg. or better.
Double ridge flange design utilizing copper gaskets.

M-012
Mars, A., Lazav, N.M.
Metallurgical Factors In The Design Of Hydraulic Equipment For Elevated Temperature Application
A.S.M.E. Paper No. 58AV11, January 1958

The metallurgical factors controlling the use of metals for hydraulic application in the temperature range of 400 to 900°F are discussed. Materials available were appraised. Tool steel grade steels show promise. The cast irons show satisfactory structural stability in the foregoing temperature range.

M-013
Marshall, A.
Wide Temperature Range Pump Seal
Rev. Sci. Instr., V. 24, pp. 1151-2, Dec. 1953

A modification has been made to an Eastern Industries Pump, Model EL, enabling it to pump acetone at -78°C and water at 100°C without leakage. The rotor was soldered onto the pump shaft and its semi-conical end seating into the graphite carbon cup constitutes the liquid seal. The carbon cup and its container are a free fit on the shaft, and contact between the rotor and the carbon is maintained by tension in the bellows.

M-014
Marsh, M.G.
Developing Face Seals For Rotating Shafts
Nuclear Engrg., V. 7, pp. 149-151, April 1962

Descriptions, with experimental evaluation data, of progressive improvement of combined thrust bearing and annular face seal originally developed for hydrogen blower on 60 MW turbo-alternator and air-lubricated face seal used in Zenith reactor coolant circuit.

M-015
Marshall, J.
Felt As Bearing Seal
Product Eng., V. 2, pp. 508-9, Nov. 1931

List of factors governing selection of proper type of seal and bearing design; sealing materials and their characteristics; S.A.E. felt standards; characteristics of various types of felt bearing seals.

M-016
Martin, A.J.
Metal Seals In Vacuum Equipment
A.W.R.E. Report No. 0-40/54, August 1954.
(Atomic Weapons Research Establishment)

This report presents some measurements of leakage rates through metal gaskets under various conditions. The materials tested were aluminum, copper, indium, and solid mercury. Seamless wire rings of aluminum and copper tested from liquid air temp. to 400°C for aluminum, 800°C for copper. Limited work with rubber O-rings.

M-017
Martin, G.S.
O-Ring Vacuum Valve For Use With A Visible Flow Industrial Glass Apparatus
J. Sci. Instrum., V. 36, p. 141, March 1959

The body of the valve is attached to the pipe flange by a modified form of Q.V.F. fitting, in which a projecting metal flange on the body replaces the usual metal backing flange. An O-ring between the flat-ground end of the flange and the body of the valve ensures a vacuum tight seal. Descriptive drawing included.

M-018 T
Martin, H.M.
Labyrinth Packings
Engineering, V. 85, pp. 35-36, Jan. 10, 1908

The author used the total energy change of the steam to derive an equation for leakage past a labyrinth seal, assuming: -that when steam is throttled the product of pressure times specific volume is constant; subcritical flow and a discharge coefficient of unity; labyrinth area is constant for all stages. Corrections must be made if critical velocity is exceeded, necessitating trial and error method. Not convenient for design.
No test results.

M-019 T
Martin, H.M.
Steam Leakage In Dummies Of The Ljungstrom Type
Engineering, V. 107, pp. 1-3, Jan. 3, 1919

Accuracy of original formula, of 1908, established, a correction derived to extend its application to series of labyrinths of increasing area. Assumed the diameter of rings increase in arithmetic progression. No graphic data for design use.

M-020
Mason, C.J.
The Story Of Packing: Fibrous And Metallic
South Power Jl., V. 46, pp. 59-64, May 1928, V. 49, pp. 58-63, July 1928

May: Purpose of packing; kinds of packing; types of self-setting packings; superheat steam packing; hydraulic and pump packing. July: metallic packing; principle involved; flexible fing type; for ammonia; France vanadium metallic packing; description of Cooke seal ring; choosing metallic packing.

M-021
Mason, H.R., and Magina, A.S.
Additional Qualification Testing Of The Soft Seat "Circle Seal" Check Valve, Dwg. No. 733B205 R56AT24, GEL Tech. Data Center, G.E. Co., Sch'dy., N.Y., (37 pp.), March 29, 1956

Report presents the results of further testing required to demonstrate conformance with requirements as set forth in Boeing Airplane Co. letter 480-9-34804-461, dated Jan. 9, 1956.
Conclusion: The "Circle Seal" check valve 733B205 met all requirements and is recommended for introduction with the redesigned reservoir pressurization system into the 7T-AP-108 Turbo Hydraulic Pump.

M-022
Mason, J.C.
Mechanical Seals: Design, Operation, Selection, And Application
Plant Eng., V. 13, pp. 123-127, April 1959

Pump seals discussed include internal, external, single, double, unbalanced and balanced seals; gland cooling, lubrication, quenching, venting and draining. Material section includes: C, cast iron, bronze, Ni-Resist, stellite, glass, ceramics, and others.

M-023 T
Mataich, P.F.
Development Of Friction Seal Materials For High Temperature High Speed Operation
Machine Design, V. 29, p.186, Sept. 19, 1957, (PB-1219440.T.S.) From WADC Tech. Report No. 56-579 ASTIA - AD 118047

New seal materials are being developed for operation at 1000°F and 30,000 fpm, with life expectancies of 1000 hours without measurable deterioration. Reduction of friction, galling, and scuffing are discussed. Development centered around a soft silver phase infiltrated into a porous nickel matrix containing 5-10% of chromium boride, or tungsten carbide. Graphs of wear vs. temperature are included for a speed of 18,000 fpm with 11.5 psi face pressure.

M-024
Matheis, H.C.
Leather Packings And The Pitting Of Stainless Steel
Modern Industrial Press, V. 10, pp. 32, 36, Sept. 1938

It was found that salts present in the leather caused pitting. The same was true for other types of packing material. Use of special impregnating compounds prevented the corrosion.

M-025
Matheis, H.C.
Packings: Their Importance In The Life Of The Press
Modern Ind. Press, V. 5, pp. 48, 50, June 1943

Proper operation of hydraulic and pneumatic equipment predicated upon properly designed packings. Paper lists factors which must be anticipated in the design of a packing. Among these are operating pressure, temperature, operating medium, environment, size, material, etc.

M-026
Matheis, H.C.
Modern "V" Packing Design
Modern Ind. Press, V. 5, pp. 21-22, Aug. 1943

Modern "V" packings have found wide application and success on hydraulic press and aircraft installations. Unique properties contribute to long maintenance free service. Paper cites such properties and application norms to be considered.

M-027
Matheis, H.C.
High Pressure Hydraulic Leather Packings
Modern Ind. Press, V. 7, pp. 26, 30, Jan. 1945

Four standard types of packings used in hydraulic equipment each of which has its place and application. These are flange, cup, "U" and "V" packings. Pressure limitations, material, and space limitations of each is covered.

M-028
Matheis, H.C.
Design And Application Of O-Ring Packings
Modern Ind. Press, V. 7, pp. 24, 26, July 1945

Synthetic O-ring packings when carefully designed and applied to equipment where conditions are suitable are productive of satisfactory results.

M-029
Matheis, H.C.
Design And Application Of Leather Flange Packings
Modern Ind. Press, V. 7, pp. 38, 40, Sept. 1945

Flange packings usually used for small cross sectional widths, where a "U" packing cannot be used and on rods of small diameters where reasonably low pressures prevail. Ideal as oil or grease seals on rotating shafts at a maximum velocity of 1,000 feet per min. Design and application norms considered.

M-030
Matheis, H.C.
Design And Application Of Fabricated Synthetic Rubber "V" Packings
Modern Ind. Press., V. 8, pp. 22, 24, Jan. 1946

Paper describes a fabricated packing which gets its name from being fabricated from synthetic rubber and cotton duck or asbestos. By varying the cures and compounds it can be applied to steam pressure operating up to 700°F and as low as 0°F. It can be applied to pressures up to 10,000 lbs. per sq. in. Various-design configurations and their application are discussed.

M-031
Matheis, H.C.
Why Use Mechanical Packings
Modern Ind. Press, V. 10, pp. 24, 26, March 1946

The shape or design of these types is such that the fluid or gas pressure acts on the geometric contour of packing, causing the mechanical action to take place. It is not necessary to exert compression on the packing to cause the proper sealing action.

M-032
Matheis, H.C.
Application And Design Of Homogeneous Synthetic Rubber "V" Packings
Modern Ind. Press, V. 8, pp. 30, 32, 34, May 1946

Synthetic rubber "V" packings are readily adaptable to a rotating seal and can be applied successfully up to a surface speed of 200 feet per minute and a pressure of 750 lbs. per sq. in. Amount of lubrication, type of fluid and the temperature involved are often factors in determining the speed beyond that mentioned which can be used. Temperatures of 65°F below zero to 212°F are not uncommon.
Application tables are included.

M-033
Matheis, H.C.
Leather Packing Expanders; Their Use And Application
Modern Ind. Press, V. 9, pp. 30, 34, 44, Dec. 1947

Article explains difficulty of maintaining softness and flexibility in leather packings for low or built up pressures, after having impregnated the leather packings to render them impervious to the passage of air or the liquid medium used. Metal spring expanders solve this problem, making impregnation unnecessary.

M-034 B
Matheis, H.C.
How To Choose The Proper Packing
Appl. Hyd., V. 3, No. 8, pp. 22-4 and 53, Sept. 1950

Four types of packing: "V", "U", Cup, and O-rings are considered the most effective for high-pressure conditions, and are discussed on the basis of their merits for different applications. The author also briefly considers surface finish and clearance.

M-035
Matron, F.D.
Back Pressure Control Unit Uses Long-Stroke Rolling Seal
Instruments and Control Systems, V. 33, p. 984, June 1960

Drawings and discussion including the rolling diaphragm seal, its advantages.

M-036
Matt, R.J.
High Temperature Metal Bellows Seals For Aircraft And Missile Accessories
Transaction of A.S.M.E. Journal, Paper No. 62-WA-25, 7 pp., Nov. 1962

Seals operating over a wide temperature range require greater precision and design analysis than the standard cartridge seals used in most rotary equipment. Discussion of bellows construction, pressure limits, environmental limits, flatness, when and where to be flat, seal ring retention, mating rings, vibration, materials, relative fatigue rating, weldability, fatigue and failure. Drawings, photograph, and data.

M-037 T
May, E.M.
Rotary Joints With Fluid Seal
U.S. Patent 2,635,931, April 21, 1953

An invention which makes practical the use of O-rings as rotating shaft seals. The O-ring is forced by the sealed pressure against a stationary sleeve between shaft and housing. The friction forces of the stationary elements on the O-ring are greater than those of the shaft, and the O-ring then remains stationary as the shaft rotates.

M-038
May, E.M.
Pressure Drop Across A Packing Tells The Sealing Story
Applied Hydraulics, V. 10, pp. 110-112, 114, May 1957

Just how a packing works is a question often left to theorists. But here is how one company found out by using an original design of test equipment and showing that pressure drop readings along the pressure path are indicative of sealing characteristics.

M-039 T
Mayer, E.
Double-Acting Axial Slip Rings In The Chemical Industry (In German)
Chemical-Ingenieur-Technik, V. 32, pp. 285-288, 1960

The usual equation for the leakage rate of mechanical seals, which was derived assuming laminar flow, non-contacting walls, and no relative velocity, is discontinued. A new equation in which leakage is a function of the square of the gap height is developed. It is also noteworthy that the sealing width does not appear in the new equation, as it did in the laminar equation.
Practical experiences and test results with double seals are described with emphasis on material and construction.

M-040
Mayer, E.
Leakage And Wear In Mechanical Seals
Mach. Design, V. 32, pp. 106-113, March 3, 1960

How to reduce seal leakage and wear; their causes; leakage rate calculations; recommended design practices.

M-041
Mayer, E.
Resistance To Thermal Stress Cracking Of Materials, Used For Unbalanced Mechanical And Circumferential Seals (In German)
VDI Zeit, V. 102, pp. 728-32, June 21, 1960

Methods of computing resistance of individual materials and pairs of materials used in slide ring seals; tabulated data to facilitate computing "resistance factor" of plastics, synthetic and natural graphite, metals and alloys, and metallic oxides and carbides.

M-042 T
Mayer, E.
Loaded Axial Slip-Ring Seals For Liquids (In German)
Konstruktion, V. 12, No. 4, pp. 147-155, 1960, V. 12, No. 5, pp. 210-218, 1960. See also: Int. Conf. on Fluid Sealing, Paper E2, April 17-19-1961

Extensive investigation covering three fluids, five pairs of materials. More than 20,000 hours operation.

It is concluded that the flow and friction processes in mechanical face seals under normal operating conditions cannot be expressed by known formulas of hydrodynamics.
A new empirical equation is presented in which leakage varies with the square of the distance between faces and inversely with the square of the face pressure.

M-043 T
Mayer, E.
Leakage And Friction Of Mechanical Seals With Special Consideration Of Hydrodynamic Mechanical Seals
Int. Conf. on Fluid Sealing, Paper E3, 14 pp., April 17-19, 1961, British Hydromechanics Research Assn. Harlow, Essex, England

Standard balanced and unbalanced seals normally perform under boundary lubrication conditions. To increase the pressure - velocity limitation of present seals, it is necessary to reduce the coefficient of friction and increase cooling and heat dissipation. Investigations at pressures up to 3000 psi show relationship between friction and leakage and advantages of hydromechanical seals. New leakage and design formula and flow constants discussed.

M-044
Mayhew, W.E.
Design And Test Of A High Temperature Hydraulic System
National Conference On Hydraulic Testing, V. XV, pp. 17-27, Oct. 19-1961

A double stage dynamic seal is discussed. Pressure 1000 psi, at 1000°F fluid temperature.
Tests compare results after 5 1/2 hours run.

M-045 T
McArd, R.W.
Some Typical Metallic Packings
Power and Works Engineer, V. 39, No. 457, pp. 155-158, July 1944

The value of application of metallic packings for sealing reciprocating rods and rotating shafts is discussed, and some representative examples are illustrated.

M-046
McCarthy, P.B.
Air Leakage Through Labyrinth Seals
U.S. Atomic Energy Commission, April 18, 1955

M-047 B
McConnell, D.B.
Control Of Leakage In Taper Pipe Threads
Proc. 7th National Conference Industrial Hydraulics, V. 5, pp. 105-111, Nov. 1951

Analysis of the conditions that contribute to leaking of taper thread joints. Special taper joints are discussed as cures. Test results are discussed. No data.

M-048
McCray, C.R.
Radial Positive-Contact Seals
Machine Design, The Seals Book, pp. 9-14, Jan. 19, 1961

A radial positive contact seal is a device which applies a sealing pressure to a mating cylindrical surface to retain fluids and sometimes exclude foreign matter. Rotating shaft application most common but also applied to oscillatory or reciprocating motion.
Various types are discussed together with seal selection criterial and their significance.
Attendant problems - cause and cure - is also covered.

M-049
McCuistion, T.J.
How To Eliminate Problem Of O-Ring Leakage
Applied Hydraulics, V. 2, pp. 14, 20-1, Feb. 1949

Leakage problems encountered in application and installation of O-rings; remedies for control or elimination.

M-050 B
McCuistion, T.J.
The Correct Use Of O-Rings On Valve Seats
Appl. Hyd., V. 1, No. 11, p. 19, Dec. 1948

M-051
McCuistion, T.J.
Floating Gland Design For Long O-Ring Life
App. Hydraulics, V. 4, pp. 59-60, 90-91, Sept. 1951

Floating gland design result of study aimed toward increasing packing life. Design presents entirely new size relationship of piston grooves and cylinder bores to packing sizes. Previous theories of what was considered to be optimum "squeeze" do not apply.

M-052
McCuistion, T.J.
A New Rotary Seal For High Speed And High Pressure Applications
India Rubber World, V. 125, pp. 575-578, 581, Feb. 1952

Describes theory and experimental details in the development of the seal. Includes a consideration of compound characteristics. Referred to as R"O"TO seals. They are fabricated from a type of synthetic rubber.

M-053 B
McCuistion, T.J.
O-Rings In Rotary High Speed Applications
Applied Hydraulics, V. 5, No. 7, pp. 68-71, 106, July 1952

The author claims to have successfully used oversize rings under peripheral compression, and gives details of a number of severe condition tests. Also includes useful information concerning the seal materials, groove dimensions, and shaft surface most suitable for the use of O-rings for rotary motion.

M-054
McCuistion, T.J.
New Gland Design For High Temperature Aircraft Hydraulic And Pneumatic Seals
Committee A6, Aircraft Hydraulic and Pneumatic Eq. SAE-Conf., April 27-29, 1955

M-055
McCuistion, T.J.
O-Ring And Gland Design For High Temperature Seals
Prod. Engng., V. 27, pp. 151-155, Jan. 1956

Factors affecting the design of glands for sealing reciprocating or rotary motion under extremes of heat and pressure; requirements for seal material, gland design, lubrication, loading and surface finish are covered.

M-056
McCuistion, T.J., and Allen, R.E.
Factors Influencing O-Ring Operation
Applied Hydraulics, V. 3, pp. 12-4, June 1950

Design problems considered including dimensions, tolerances and clearances, stretching, stress cracking, etc; points to check in selection of O-ring compounds.

M-057 B
McCuistion, T.J., Schaube, E.M.
O-Rings Cure Data And Age Control
Applied Hydraulics, V. 10, No. 4, pp. 114-7, April 1957

This article provides ready reference charts with latest changes in the cure date life of O-rings for hydraulic applications, as specified by military services.

M-058
McCuistion, T.J., Teree, B.R., Pollard, F.H.
New Hydraulic Fluids And Seals
SAE Journal, V. 63, pp. 45-47, Sept. 1955

Review of hydraulic fluid problems; temperatures from -65°F to 400°F; silicone and neoprene rubber seals; seal design materials and techniques.

M-059
McCuistion, T.J., and others
Research On Aircraft Hydraulic Packings
SAE Journal, V. 2, pp. 227-233, 267, April 1948

Report covers survey of hydraulic packings, especially O-ring seals. Covers friction studies, torsional strength tests, plasticizer development, a study of rubbers, molding technique, and spiral failure.

M-060
McCutchan, A.
Gasketed Joints For High Pressure High Temperature Piping Service
Heating-Piping, V. 13, pp. 423-7, July 1941

The author points out that various types of gaskets can be used with good success provided flanges and bolting are sufficiently heavy. It is important to estimate the size of bending moments acting on flanged joints, because their magnitude has a pronounced effect on the ability to keep such joints tight. The effect of water, creep failure are discussed.
Drawings, tables, photos included.

M-061
McGeary, F.M.
Standardization Of Packing Materials For Naval And Marine Machinery
Am. Soc. Nav. Engrs. Jour., V. 41, pp. 563-588, Nov. 1929

M-062
McIlvraith, P.L.
Selection Of Axial Shaft Seals
Mat. in Design Engng., V. 2, pp. 90-94, Feb. 1959

Characteristics and factors for selection of seals whose basic components comprise stationary rubber ring, rotating seal ring, static seal, and pressure ring; advantages of use include elimination of wear on shaft, control of fluid leakage, long service life, reduced power absorption, and ability to compensate for shaft vibration, eccentricity and end play; application to handling of gases and liquids.

M-063 T
McLaughlin, R.W.
Bearings And Lubrication For High Speed Machinery
Product Engineering, V. 19, pp. 108-112, Dec. 1948

This article treats in a practical way the general requirements of the lubricating system of large high speed machines such as compressors, turbines, and generators. Components for adequate system are discussed. The shaft seal system and other requirements are noted.

M-064
McLean, D., Farmer, M.H.
Measurement Of Force Required To Pull Through A 1 Inch Dia. Wilson Vacuum Seal
J. Science Instrum., V. 30, p. 293, August 1953

The construction of a Wilson vacuum seal, as an alternative to the mercury seal, is described, and its frictional resistance to pulling determined. For an adequately greased seal this was found to be about 1/4 kg, which could be ignored in tensile creep test in vacuum (loads 100 to 500 kg) on copper and other materials liable to hydrogen contamination.

M-065 T
McNair, D.G.
Pump Glands In Operation
Power Engineering, V. 29, No. 345, pp. 453-456, Dec. 1934

This article discusses practical problems arising from centrifugal pump service conditions. Special liquids are considered.

M-066
McSwiney, A.
Steam-Engine Packing
Mech. Eng., V. 9, pp. 739-744, May 31, 1902

A description is given of various forms of metallic packings and stuffing-boxes.

M-067
Mehnert, K.
The Consideration Of Thermoplastic Properties Of Fluoro-Polymers As Gasket And Packing Materials (In German)
Chem. Ing. Tech., V. 27, pp. 284-6, 1955

Deformation behavior of Teflon and Kel-F polymers under static loading at 20 to 140°C and pressures of 25-150 Kg/sq. cm. is compared with behavior of rubber.

M-068
Merkel, E.
Fluorine-Containing Ethylene Polymers As Gasket And Packing Materials (In German)
Chem. Ing. Tech., V. 27, pp. 279-83, 1955

A discussion of the various applications of Teflon and Hostaflon with many diagrams and figures.

M-069
Meyer, R.
Gasketing Of Flanged Connections For Ultra High-Pressure Service
Mach. Design, V. 28, pp. 109-112, Nov. 1, 1956

Article is concerned chiefly with flanged connections for pressures above 10,000 psi, an area that may be labeled as ultra-high pressure.

M-070 T
Michel, R.G.
Flooded Bearing Seal For Gas-Filled Machines
U.S. Patent 2,899,245, August 11, 1959

A sealing system for bearings on gas filled machines. The bearing is used as the seal, with a forced oil supply entering between inner and outer portions of the bearing with sufficient pressure and flow to prevent gas from flowing counter to the oil.

M-071
Middleton, R.E.
Hydraulic And Pneumatic Problems In High-Performance Aircraft
Applied Hydraulics, V. 9, pp. 98-101, Dec. 1956

General discussion of problems of temperatures of 400 to 600F. Suggests metal seals will be required.

M-072
Middleton, R.E.
O-Ring Packing Leakage
Lockheed Field Serv. Dig., pp. 10-15, Jan.-Feb. 1957

Discussion of the development of seals and back-up rings for aircraft hydraulic systems, and methods of evaluating leaks around O-rings.

M-073
Middleton, R.E.
O-Ring Packing Leakage
Flight Safety Found., Av. Mech. Bul., pp-4-8, 18, 19, March-April, 1957

M-074
Milleron, N.
Utilization Of The Surface Tension Of Liquid Metals In Making High Vacuum Seals
UCRL-4938, U/CAL Radiation Laboratory, Aug. 30, 1957

M-075 T
Miner, J.A.
Rotary Shaft Seal
U.S. Patent 2,265,951, Dec. 9, 1941

A seal designed for track rollers on track laying vehicles. A floating ring mounted with resilient gaskets on O-rings in the roller, presses against a stationary ring joined to the main shaft and housing by a tube of resilient material which fits into the grooves on the side of the ring and housing, and is concentric to the main shaft.

M-076
Minter, V.A.R.
Getting The Right Pump Packing
Power, V. 84, pp. 256, 257, April, 1940

M-077
Mitchell, C.J.B.
Recent Solutions To Pumping Problems
Chem. Engng. Prog., V. 50, pp. 431-435, Sept. 1954

Reviews several sealing devices which provide long life, uninterrupted service, and inexpensive replacement of a minimum of parts.
Problems pertinent to chemical industry.

M-078 T
Mitchell, T.
Refrigeration Note Book: Tips On Stuffing Boxes And Special Shaft Seals
Power, V. 88, p. 397, June 1944

Maintenance hints on compressor stuffing boxes and shaft seals.

M-079 T
Miyadzu, A.
Theory Of The Westco Type Rotary Pump
Trans. Japan Soc. Mech. Engrs., V. 5, pp. 109-115, Feb. 1939

This article not available for review as of March 1, 1962.

M-080 T
Miyadzu, A.
Theory Of The Viscosity Pump
Trans. Japan Soc. Mech. Engrs., V. 9, pp. 85-96, pt. 3, Nov. 1943

This article not available for review as of March 1, 1962.

M-081 T
Mochel, N.L.
Carbon Gland Rings
Power Plant Engineering, V. 42, p. 624, Oct. 1938

The use of renewable sleeves of hard stainless steel under the carbon or heavy chromium plating of the shaft to prevent pitting and thus ring wear, is suggested.

M-082
Moffett, J.A., and Panagrossi, A.
Silicone Rubbers: New Gasketing Materials For High-Temperature Uses
Mach. Design, V. 18, pp. 109-113, Sept. 1946

Comparison drawn between silicone and natural rubbers for heat resistance, dielectric strength, moisture and chemical resistance and volatility. Applications cited in X-ray, aircraft, steam generator, gas cylinder and oil burner devices.

M-083 B
Molyneaux, F.
A Simple Mechanical Seal For Pumps Or Agitators
British Chem. Eng., V. 5, No. 9, p. 663, Sept. 1960

M-084
Moncher, F.L.
Hydraulic Components And System Considerations For High Mach. Flight
A.S.M.E. Paper 57-A-223, Dec. 1957

High Mach. flight leads to increased use of automatic controls. Hydraulics is particularly adapted to this application because of the inherently fast response. Severe development problems exist, resulting from high temperatures induced by aerodynamic heating.

M-085
Moncher, F.L., Taylor, L.D.
Designing And Testing For High-Temperature Hydraulics
A.S.M.E. Paper 58-A-V-6, December 1957

Development of hydraulic equipment for high temperature applications involves considerable modification in present designs and specialized testing techniques. The test procedures and facilities for evaluation and qualification of hydraulic equipment suitable for use at temperatures in excess of 400°F are complicated, difficult to control, and expensive.

M-086
Monich, M.T., and Bragdon, C.T.
Properties Of Integral Seal Ball Bearings
Gen. Motors Eng.-Jour., V. 4, pp. 8-15, July, Aug., Sept. 1957

Contact and clearance type seals; properties of felt or synthetic rubber seal materials, relative seal efficiency according to grease retention, wet contaminant exclusion, and dry contaminant exclusion tabulated; how design factors of final device affect sealed bearing use.

M-087
Monroe, P.H.
Description And Operational Aspects Of Main Helium Circulators For The 40-MW(e) H.T.G.R. Plant
Nuc. Sci. Abst. 28386 (V. 16, No. 20, p. 3713, Oct. 31, 1962), (Gen. Atomic Div., Gen. Dynamics Corp., San Diego, Calif.)

A general description including operational aspects of the main gas coolant circulators is presented for 40-MW(e) HTGR plant. Included are discussions of the main coolant system, circulators, shaft seal system, drivers, lube system, and operation.

M-088
Montgomery, W.J.
Chromate Gasketing
Sheet metal worker, V. 36, pp. 41-42, Feb. 1945,
Same-Marine Eng., V. 50, p. 200, April 1945,
Same-Iron Age, V. 155, p. 61, Feb. 5, 1945 (cond.)
Same-Chem. Inc., V. 56, p. 86, Jan. 1945 (cond.)

East of cutting, resistance to fire and water make the chromate gasket more than a substitute for war-scarce rubber gasketing.

M-089 B
Moore, C.F.
Compression Packings - Using The Right Ones
Plant Engineering, V. 13, 6, pp. 105-7, June 1959

M-090
Moore, H.R.
Demountable Vacuum Seal For Attaching An End-Plate To A Glass Tube
Rev. Sci. Instrum., V. 29, No. 8, p. 737, Aug. 1958

Although designed originally for attaching glass and quartz windows onto glass discharge tubes, the seal should be effective with other combinations of material. Rotation of a knurled lock ring causes a 45° bevel on the collar to press against an O-ring, the collar sliding freely on the tube. Equal pressures are then transmitted by the O-ring to the surfaces to be sealed, the outer tube surface and the inner surface of the end plate.

M-091 T
Moores, J., Marsh, M.C.
The Effects Of Eccentricity And Flatness Of The Sealing Face Of A Carbon Mechanical Contact Face Seal
Int. Conf. of Fluid Sealing, Paper H1(9 pp.), April 17-19, 1961, British Hydromechanics Research Assn., Harlow, Essex, England

Test results confirmed that variations in performance of nominally identical seals might be due to small differences in eccentricity, the eccentricity encouraging a film of lubricant between the faces.

M-092
Morfey, D.S.
Improvements Relating To Seals For Pressure Vessels
British Patent 856,952, Dec. 1960

A seal of the same type as the ordinary ring seal but without a tendency to leak is designed comprising a T-shaped groove in the closure member and a resilient ring or gasket formed with lateral ears to retain it in the groove.

M-093
Morgan, H.E.
Review Of Turbine Sealing Methods
Power Engng., V. 61, No. 7, pp. 78-79, July 1957

Features of labyrinth seal, designed for modern turbine operation, which consists of thin strips of metal with spaces between them that project to shaft; use of carbon rings in combination with labyrinth seal acting as pressure breakdown; complete gland seal for shaft ends; exhauster system; use of brushing and flexible metal packing to prevent leakage at valve stems, multiple bushing or floating ring seal assembly for range beyond 850 psig and 1000°F.

M-094
Morgan, P.G.
Radial Compression: The Effect Of Low Temperatures On Rubber Packing Rings
Rubber Journal and International Plastics, V. 136, No. 22, p. 872, June 6, 1959

Cooling of rubber seals brings about a decrease in radial force, the amount depending upon the initial compression.

M-095 T
Morgan, R.E., Hummer, H.B.
Mechanical Shaft Seals-Their Pressure-Velocity Limitations
Petroleum Engineer, V. 28, pp. 34-38, Nov. 1955

Description of balanced and unbalanced seals. Test results on face materials. The authors assigned a P.V. rating to each face material and fluid combination. This P.V. rating, when divided by peripheral speed in ft/min. gives the maximum contact pressure limitation in psi.

M-096
Morris, A.E.
Temperature Control In Hydraulic Systems
Applied Hydraulics, V. 10, pp. 68-85, August 1957

Report on methods for determining how much heat will be generated, whether auxiliary oil coolers will be required, and how to select these and other controls. Also, heat generation, temperature effect on fluid packings, natural heat dissipation, system design to reduce heating, oil cooling, immersion heaters, automatic temperature control.

M-097
Morris, R.E., Hollister, J.W.
Plasticizer For G-R-S Gasket Stock To Be Used At Low Temperatures
Rubber Age, V. 70, pp. 195-203, Nov. 1951

Plasticizers were tested in G-R-S stock for extractability by water and volatility. Several G-R-S stocks containing different plasticizers were checked for compression set at -60°, - 30°, 0°, and + 30°F after various recovery times. Data are graphed.

M-098
Morris, R.E., Hollister, J.W., Shew, F.L.
Butadiene Polymers For Low Temperature Service
Ind. & Eng. Chem., V. 43, pp. 2496-2500, Nov. 1951

The purpose of this investigation was to determine whether or not Butadiene-styrene co-polymers with different ratios constitutes a different polymerization temperature than standard GR-S and/or are better rubbers for the manufacture of gaskets for low temperature service than standard GR-S.

The Butadiene-styrene co-polymers tested totaled 34. The cold compression set test was performed at -35°F for evaluation. Test results on experimental stocks are presented in table and graphical form.

M-099
Morris, T.R.
Metal Gaskets Should Not Be So Large As To Extend To The Flange Bolts
Coal Age, V. 18, p. 171, July 22, 1920

M-100
Morrison, J.B.
O-Rings And Interference Seals For Static Applications
Machine Design, V. 29, pp. 91-94, Feb. 7, 1957

Interference seals such as the familiar O-ring, are widely employed for static sealing in nearly all pressure ranges. Mechanics of such seals is little known. Article gives mathematical analysis of interference seals, design formulas, and describes a recently developed application.

M-101
Morrison, W.M.
New Approach To Hydraulic Seal Design
Compressed Air & Hydraulics, V. 27, No. 316, pp. 277-9, July 1962

Polytetrafluoroethylene working surface allied to synthetic rubber support in specific dimensional relationship is used as combination of materials for seals; tests of sealing in variety of applications throughout temperature range of minus 80°C to 220°C at pressures from 0 to 4000 psig showed low friction coefficient compared with conventional types; seals are suitable for variety of hydraulic fluids including inflammable oils.

M-102
Morrow, J.
Theory Of Labyrinth Packing
University of Durham-Philosoph. Soo. Proc. 3.S, pp. 281-285, 1909-1910, Engineering, V. 90, p. 136, July 22, 1910

Although the clearance between each strip and collar is reduced to the lowest practicable limit, steam passes through each clearance space at high velocity, but this velocity is destroyed by eddying in the spaces between the strips. A part of the drop pressure is thus consumed at each clearance. The mathematical equations required in this method of calculation are similar to those used for the flow of steam through orifices, those for the energy; for those portions throughout which the flow is assumed to be adiabatic. From these are derived equations for velocity and weight of steam. The application of this theory to the design of labyrinth packings and to the calculations of the pressures at the various clearances follows.

M-103
Moslander, K.P.
New Designs Broaden Scope Of Shaft Seals
Mach. Design, V. 12, pp. 44-7, Nov. 1940

Various types of face seals are discussed; seal noses impregnated with different metals depending upon operating conditions of temperature and load; metallic bellows seals; rubber or synthetic diaphragm type of seal; lubricating problems; illustrations of the various seals.

M-104
Motz, W.H.
Ammonia Compressor Rod Packing
Power, V. 53, pp. 543-545, April 5, 1921

Describes the principle of packing:
(a) the use of separating oil films,
(b) requirements of a good packing,
(c) types of packing,
(d) the merits of packing
Metallic packings vs. fibrous packings.
No data.

M-105
Motz, W.H.
Packing For Refrigerating Machines
Power Pl. Engng., V. 31, pp. 363-366, March 15, 1927

Describes representative types and discusses their operating characteristics.

M-106
Mowers, R.E.
How The New Propellants Affect Plastics And Elastomers
Mat. in Design Engrg., V. 50, pp. 89-91, Sept. 1959

Graphs show quantitatively the effects of the new storable missile propellants on properties of usable elastomers and plastics. Reference is made to CFE fluorocarbon resin as being the most desirable material for lip seals. Despite this, many lip seals removed from disassembled valves after actual service in hydrazine fluids have shown radial cracks. Such failures are believed to have been caused by the combination of internal stresses in the molded seal and exposure to the fuel. A possible solution to this problem is to fabricate seals by postforming from sheet stock. Parts then stress relieved prior to final dimension.

M-107
Moyer, W.H., and Piotter, E.C.
Flexitallic Gaskets For Mechanical Closures
Babcock and Wilcox Co. Res. and Devel. Dept.
BW-5409, 39 pp., Jan. 2, 1956

Leakage rates have been determined for two types of flexitallic gaskets. Test fluids were water and helium. Test conditions included temperatures up to 500°F and pressures up to 2500 psi.

M-108
Mueller, M.H.
Vacuum Seal For Thin Metal Windows
Rev. Sci. Instr., V. 27, p. 411, June 1956

A method is presented for vacuum sealing of thin metal windows such as the BE windows in high-temperature X-ray cameras; a modification of the O-ring type seal was developed for curved windows.

M-109
Muller, E.
Compressed Oil Operation Of Tube And Bar Extrusion Presses For The Processing Of Heavy And Light Metals
Zeitschrift fur Metallkunde, V. 46, No. 1, pp. 11-16, Jan. 1955 (In German)

Design and operation of high-power oil pumps whose advantages are reliability, simple packing design, and absence of corrosion.
Photographs, drawings.

M-110
Muller, W.
High-Pressure High-Stroke Diaphragm Mechanism
Engineer, V. 198, pp. 602-3, Oct. 29, 1954

Flexible diaphragms enable a complete seal to be provided between two chambers and at the same time allow axial movement (e.g. for control purposes) to be transmitted from one chamber into the other. Can be used with a combination of liquids and gases.

M-111 T
Mullner, F.
Seal For Rotary Shafts Of Gas-Cooled Machines
U.S. Patent 2,743,948, May 1, 1956

A carbon ring seal. Two rings of rectangular carbon blocks are used, with individual blocks of each ring positioned by separating wedges on an adjacent radial plate. The outer plate contains a set of springs which cause the two sets of carbon blocks to bear against each other and the two rings are positioned so that the blocks on each cover the spaces between the blocks on the other. A rubber ring with a garter spring around it causes the blocks to bear against the shaft and effects a static seal between the two plates holding the carbon blocks.

M-112
Murray, E.A.
Molding Gaskets For Feed Valves
Railway Mechanical Eng., V. 90, p. 320, June 1916

M-113
Murray, J.
Effect Of New E-P Lubes On Elastomers For Oil Seals
Lubrication Eng., V. 15, pp. 140-143, April 1959

Discussion of chemical effects of oil and chemical additives on elastomer seals.

N-001
Nakagawa
Ramsbottom Packing Rings
Engineering, V. 98, p. 130, July 24, 1914

Packing rings made under various conditions are allowed to press small blocks outward against the inside of a cylinder, and the force necessary to slide the blocks is taken as a measure of the normal pressure exerted by the ring. The polar curves of pressure are all symmetrical about a line and have five pronounced lobes. The author suggests a new design of a ring, the inner profile of which consists of two semicircles of unequal diameter and unequal eccentricity with regard to the outer circle.

N-002 T
Naka, H., Kawasaki, H.
On the Experimental Results Of A Viscosity Pump
Jnl. Japan Soc. Mech. Engrs., V. 46, No. 310, pp. 11-13, 1943

This article not available for review as of March 1, 1962.

N-003 T
Nau, B.S., Turnbull, D.E.
Some Effects Of Elastic Deformation On The Characteristics Of Balanced Radial Face Seals
Int. Conf. on Fluid Sealing, Paper D3 (8 pages), April 17-19, 1961
British Hydromechanics Research Assn., Harlow, Essex, England

When a relatively small amount of pressure balancing is used in design of radial face seals, they are able to run with a very thin fluid film separating the faces. Test results ~~show~~ that with local deformations on the face, the film is capable of supporting a greater load than predicted when no deformation occurs.

N-004
Navarre, N.L., Stevens, R.W.
High Pressure, High Temperature Steam Flange Type Joint Sealed With A Flexitallic Gasket
ESE Report 6A66781, U.S. Naval Engineering Station

N-005 B
Naw, B.S.
A Reconsideration of Pressure Generation in Radial Face Seals
British Hydromechanics Research Association, RR699 (244 pp), Sept. 1961

First, a non-mathematical account of the present situation regarding the phenomenon of pressure generation and inward pumping as they affect radial face seals. Second, the formal presentation of mathematical analysis of radial face seals and expressions for the distribution of averaged pressure and the load carrying capacity are derived. Discussed also is coefficient of friction of misaligned seal.

N-006 T
Naylor, H.
The Use Of Mechanical Seals
Steam Engineer, V. 26, pp. 17-19, Oct. 1956

The design and application of mechanical seals are briefly discussed.

N-007 T
Nazio, C.H.
Liquid Seal
U.S. Patent 2,288,638, July 7, 1942

A seal for a vertical shaft in which mercury is used as the sealing means.

N-008

Neaf, W.
Stuffing Box Packing (In German)
Gummi-Ztg., V.51, pp. 1213-4, Dec.17, 1937

Selection of suitable packing; problem of price; directions for use of self-lubricating packing; failure of packing.

N-009
Neer, N.G.
Gland and Stuffing-Box Practice
Mech. Wld., Dec. 25, 1914

Illustrates and describes types.

N-010
Neild, A.B., Jr.
Development Of Cylinder Water Seal to Prevent Diesel Engine Crankcase Cavitation Erosion
SAE Jour., V.68, pp. 64-65, July, 1960

Failure of several crankcase designs studied at U.S. Naval Eng. Experiment Station; test procedure involving use of soft test plugs for rapid evaluation of possible corrective measures required for 600-hp aluminum crankcase; development of special cylinder liner water seal with seal bands of different Paracril BLT compounds; seal strength was adequate for at least 3000-hr. engine operational periods.

N-011 B
Neou, C.Y.
Pressure Actuated Cylindrical Diaphragm Seals
A.S.M.E. Paper 59-SA-1, (9 pp), June, 1958

Analytical formulas and charts are developed for the rational designing of a thin-walled metallic cylindrical shell or band welded peripherally to the inside of a valve body. The flexible sealing band can be inflated to provide a positive contact seal when the valve is in a fully-closed position and deflated to allow for contact-free seating and unseating of the valve disc.

N-012
Nester, R.G.
High Speed Flexible Blade Stirrer and Stirrer Seals for High Vacuum Use
Rev. Sci. Instr., V.27, pp. 1080-1, Dec., 1956

Operation of the stirrer under high vacuum is made possible by a simple, highly effective combination seal and suspension bearing seat that is made of a plug of self-lubricating plastic such as nylon, Teflon, or polyethylene. The passage through the plug, wide at the top, narrows down to form an inverted conical bearing seat. The bearing is an inverted conical, hardened stainless steel bead on the shaft of the stirrer and gives a very effective rotary seal that adjusts itself for slight wear or misalignment.

N-013
Newell, G.C.
Preliminary Results of All Metal Boss Evaluation and Development
Boeing Aircraft, Transportation Division, pp. 1-11, Sept., 1961

Drawings showing different manufacturers' seals.

N-014
Nielsen, A.T.
Centrifugal Pumps for Process Use
Chemical and Metallurgical Engineering, V. 49, No. 3, pp. 90-2, March, 1942

A number of extremely useful ideas for increasing the life of pumps operating under severe conditions, and for facilitating inspection and maintenance.

N-015
Niemeier, B.A.
Seals to Minimize Leakage At High Pressure
ASME Trans., V.75, pp.369-379, April, 1953

Fixed and moving seals used in super-pressure range of 10,000-100,000 psi are studied and evaluated. Discussion of theoretical problems in confining pressures of great magnitude precedes methods of designing seals. A method and device for evaluating seals developed. Practical seal problems and applications are shown.

N-016
Niessen, P.F.
The German Gasket Industry
PB 48421, Off. Tev. Serv. (13 pp.), October, 1946

Summarizes the processes, materials, and equipments used in production and development. Outstanding developments consist of cylinder head gaskets made from Buna or synthetic rubber and steel, and the manufacture of three meter wide sheets of compressed asbestos packing and curing or vulcanization of oil seal diaphragms at high temperatures. The complete absence of graphite in the oil seal compound and the method and equipment used for determining the running or sealing characteristics of oil seals is of special interest.

N-017
Niessen, P.F.
Amendment To Fiat Final Report #951
PB L-48421-S, Off. Tech. Serv. (Supplement to PB 48421), June, 1947

This report contains two drawings by the Victor Manufacturing and Gasket Co. One on an oil seal diaphragm manufactured by Carl Freudenberg, of Weinheim, Germany. The other illustrates various oil-seals manufactured by Carl Freudenberg. Under the trade-name of "Summering," all embody the same characteristics. An experimental mold was made and tested and test results are given.

N-018
Noble, J.E.
Prepared Tin Foil For Packing Purposes
Power Plant Engineering, V. 32, p.1205, Nov. 15, 1928

A discussion of preparation of tin foil for packings for more satisfactory use. Alternate sheets of tin foil and cotton cloth coated with powdered graphite mixed with oil, rolled, and cut to length for application.

N-019 B
Nolt, T.G.
Techniques for Evaluating Gasket Loads in Flanged Joints
Machine Design, V.33, pp. 128-34, September 28, 1961

This article describes the new technique, and the three other methods, for flange design gasket selection and leakage analysis.
Solid plug compression test; gasket load from bolt torque; gasket load from bolt elongation; gasket load from washer-type force gage.

N-020
Noonan, J.W.
Materials In The Design Of Seals For Extreme Environments
Machine Design, V.34, pp. 186, 188, etc., August 16, 1962
Abstracted from SAE paper 523G, April, 1962

Study of materials for both static and dynamic seals for vacuum effects, radiation effects, and temperature extremes.
Vacuum effects: (1) evaporation or sublimation of volatile components from polymers up to 530°C; (2) partial or complete removal of the adsorbed layer of gas which covers all materials in the atmosphere. Fatigue life of metals is improved in vacuum especially at high stresses, sometimes poorer at low stresses than in air. Creep strength of metals is better in air at high temperature and low strain rates and better in vacuum at low temperature (1100°F) and high strain rates.
Dry film lubricants are best in a hard vacuum.
Radiation effects: results of gamma and ultraviolet radiation on elastomeric materials.
Temperature effects: fatigue and creep properties of metals at low temperatures improved. Discusses high and low temperature properties of fluoroethylene polymers.

N-021
Norbom, R.
Water Cooled Stuffing Box For Oil Pump Designed For Arc Welding
Modern Power and Eng., V.37, pp. 58-9, 80, Dec. 1943

Illustrated description of long throat, water-cooled stuffing box for hot oil pump

N-022
Normand, L.E., Rathkamp, W.R.
Evaluation of Vacuum Gasket Materials
Oak Ridge National Laboratory, 7-12 Area, Oak Ridge

The evaluation of the vacuum properties of materials attempted in the investigation reported here is restricted to the determination of the relative order of values for the outgassing rates and vapor pressures of the materials. Test operation, description, and data given. Teflon, silicone SR 1060, Mycaseal, silicone SR 1080, koroseal, Hycar OR15, neoprene, Tygon, Garlock 7986.

N-023 B
Norton, R.D.
13 Ways To Solve Packing Problems
Chemical Engineering, V.62, pp. 183-6, March, 1955

Presented are 13 ways to solve the frequent problem of shaft and packing wear on centrifugal pumps for abrasive liquids.

N-024
Norton, R.D.
Mechanical Seals For Handling Abrasive Materials
Chem. Engng., V.63, pp.199-210, Sept., 1956

Centrifugal pump problems involving abrasive materials are troublesome. Article describes mechanical seals in general and means of keeping seal faces free of solids.
Many ideas described had very limited field testing.

N-025 T
Norton, R.D.
Current Uses and Developments of Mechanical Seals for Abrasive Liquids in Centrifugal Pumps
Int. Conf. on Fluid Sealing, Paper H3 (24 pp.), April 17-19, 1961.
British Hydromechanics Research Assn., Harlow, Essex, England

When abrasives are present, it is possible to design the seal to operate either in the presence of the abrasive bearing liquid or in a clean environment. Various practical examples of both are discussed. Special attention is given to crystal formation and high-temperature operation.

N-026
Nosov, Y.A., Farberova, T.
Methods of Evaluating Rubber Used In The Manufacture of Sealing Units
Soviet Rubber Technology, V.18, No. 4, p. 36, April, 1959

The purpose of this article is to review the methods of evaluating and control-testing rubber used in the manufacture of seals. The authors note the importance of stress relaxation measurements as an index to seal material usefulness. They take care to emphasize that stress relaxation and compression set are not the same. Physical properties characterize only to a very small extent the working ability of a seal. Doubts are cast upon the method of heat aging using tensile strength as a yardstick to measure aging effects. The authors prefer ultimate elongation as a more reliable indicator. An apparatus for measuring the radial pressure exerted by O-ring seals is described. Designed at the Scientific Research Institute of the Rubber Industry, the instrument utilizes a piston holding the O-ring pushed through a thin-walled cylinder instrumented with strain gages.

N-027
Novozhilov, S.F.
Packing Devices For Gas-Turbine Regenerators
Technical Translations, Off.Tech.Serv., V.1, p. 373, June 5, 1959. #59-11280

The effectiveness of a variety of packing structures was studied to control the dispersion of air from a disk-type gyrating regenerator for gas turbine engines. Distortions in packing shapes were caused by union with the revolving rotor, high temperatures, and pressure changes. Contact-type packing work out the friction surfaces, resulting in a curvilinear shape. To prevent leakage of air, packing surfaces had antifriction and stretch properties. Leakage was further reduced with heated gas and a pneumatic accordion device (sil'fon). Experiments were conducted with various packings, including a ribbed band with triangular channels; radial, depth, and circumferential structures; a ball and roller; a piston; a contoured-hollow of the shoe; and a spring. The contoured-hollow and spring types proved most effective.

O-001
O'Donnell, J.P.
Club Told O-Rings Need Standard
Oil and Gas Journal, V. 60, pp.128-30, May 7, 1962

Proponents of O-rings applying for adoption of code. Test results of O-rings, Tables and curves. Compares pressure and stress of 2-in. ASA300 flanges equipped with O-ring and the conventional gasket. Drawings and photographs.

O-002
Okochi, M., and Ebihara, K.
Researches On The Perfect Tightness Of The Stuffing Box (In Japanese)
Soc. Mech. Engrs. of Japan, Jl., V. 31, pp. 515-9, Aug. 1928

New metallic packing for ensuring tightness of stuffing box was invented, and some properties of packing were investigated.

O-003 T
Oldroyd, J.G.
Non-Newtonian Effects In Steady Motion Of Some Idealized Elastico-Viscous Liquids
Proc. Royal Soc. of London, Series A, V. 245, pp. 278-297, 1958

Normal stress effects and the variation of apparent viscosity with rate of shear in simple types of steady flow of certain idealized elastico-viscous liquids are discussed. Normal and shear stresses are investigated as is the Weissenberg climbing effect. The behavior of these idealized liquids when sheared in a narrow gap between a rotating wide angled cone and a flat plate is compared with Roberts observations.

O-004
Opheim, M.H., and Pritchard, J.E.
Oil Resistant Rubbers From 2-Methyl 5-Vinylpyridine
Ind. Eng. Chem., V. 46, pp. 2242-5, 1954

A co-polymer is described, which shows improvement in resistance to swelling and other damage by various organic solvents and synthetic lubricants.

O-005 B
Orcutt, F.K., Bell, J.C., and others
Rotating-Shaft Helium Seal Investigation
Battelle Memorial Inst. Proj. No. 8-97-65-001C - Status Report, 44 pp., 6 Fig., 3 Tab., Aug. 1959

Second stage report of investigation concerning the sealing of Helium at 600 lbs/sq. in. Experiments to determine suitable face seal material for 3" shaft rotating at 25,000 rpm.

O-006 T
Orcutt, F.K., Bell, J.C., and others
Rotating Shaft Helium Seal
Status report on contract DA-44-009-eng. 3375-Project No. 8-97-65-001C, to U.S. Army Engineer Research and Development Laboratories, Aug. 25, 1959, Supp. Rept., Nov. 9, 1959

Evaluation and study of positive rotating shaft seals for application to high speed turbo machinery using high pressure high temperature helium as working fluid. Target specifications: Turbine speed 25,000 rpm, with helium at inlet 1000 psi and 1350 F, seal life 8000 hours.

O-007 B
Osola, V.J.
Some Experiences In The Application Of Mechanical Seals To Pumps And Stirred Pressure Vessels
Chem. Engr., 152, pp. A41-53 (Incorporated in Trans. I. Chem. Engrs., V. 38, December 1960

Description of the general operating principles of mechanical seals used in water circulating pump, a process pump and overhanging shaft.
The sealing of various designs of pressure vessels, capacities up to 600 gallons, 100°C temperature, to 250 lbs/sq. in. pressure.

O-008
Ossefort, Z.T.
Evaluation Of Polymers For Use As Ammunition Box Gaskets
U.S. Arsenal, Rock Island, Ill., 33 pp., photos, graph, tables, July 1952, PB 115301 - Lib. Cong. Publication Board Project - Wash. 25, D.C.

Project No. TB4-521C Report No. 2
1. Gaskets, Synthetic Rubber-Test
2. Boxes - Gaskets
3. Rubber-Synthetic Test
4. R1A6 R52-1972.

O-009
Osten, D. von den
Packing Ring
L.C. or SLA - 60-13286 (Trans. of Swedish pat. 111,729, C147F22-11, 9 May 1942)

A packing ring, designed for packing parts which are exposed to high temperatures, is described which consists of a sheet metal jacket and a mineral filling substance. The filling substance is composed of a fine powder of a substance pressed into the jacket which, in the vicinity of the operating temperature, through physical or chemical change, experiences an extra increase in volume which is greater than that which corresponds to the simple expansion by heating. The filling substance consists of talcum.

P-001
Page, R.L.
Testing Of O-Ring Seals For Reactor Pressure Circuits
GEC Atomic Energy Rev., V. 2, pp. 210-215, Autumn, 1960

Statis seals of silicone rubber are satisfactory for sealing carbon dioxide to 150°C and 150 psig. Any loss of carbon dioxide is due to diffusion through the silicon rubber rather than to leakage.

P-002 T
Pai, S.I.
On Turbulent Flow Between Parallel Plates
ASME Trans., V. 74, pp. 109-11, 1953

Reynolds' equations of motion are derived and solved for two special cases, Poiseuille flow, and Couette flow. The mean velocity distribution and the correlation u/u (turbulent velocity fluctuations) can be expressed in a form of polynomial of the coordinate in the direction perpendicular to the plates, with the ratio of shearing stress on the plate to corresponding laminar flow of the same maximum velocity as a parameter. These expressions hold true all the way across the plate, i.e. both turbulent region and viscous layer including the laminar sublayer. It also shows that the logarithmic mean velocity distribution is not a rigorous solution of Reynolds' equations.

P-003
Pall, D.B.
Development Of Filters For 400°F And 600°F Aircraft Hydraulic Systems
Library of Congress, PB 128057 ($7.80/photostat copy) May 1956

Report on development of ten micron hydraulic oil filters operating in the range from -65°F to +600°F. Housings of aluminum construction and suitable seals were developed for range -65°F to 400°F. Steel construction -65°F to +600°F service.

P-004
Palmer, A.B.
Pressure Regulator Seals
Missile Des. & Devel., pp. 38, 50, April 1959

Development, requirements, and operation of pressure regulator seals.

P-005
Palsulich, J., and Piedel, R.E.
Dynamic Seals For Aircraft Gas Turbine Engines
SAE Paper No. 685, Jan. 9-13, 1956

Problems encountered in sealing various media in main shaft and accessory drive applications; major types of seals; sealing problems grouped into three categories; main shaft bearing lubrication systems, containing or controlling air flow within proper channels, sealing accessory drives.

P-006
Panek, J.R.
Elastomeric Joint Sealants
A.S.M.E. Paper No. 62-WA-328, 4 pp., Oct. 1962

The author discusses the need for sealants, types of sealants, recommended uses for sealants, recommended sealing practices, and specifications for elastomeric joint sealants. Thiokol polysulfide liquid polymer.
Urethane sealants
Silicone sealants
Chlorosulfanated polyethylene sealants
Acrylic-acid base sealants
Tests procedures.

P-007
Paree, D.
Illustrated Description Of Utilization Of Felt In Connection W... Ball Bearings In Order To Prevent Oil From Leaki... (In French)
Annales des Travaux Publics de Belgique, V. 46, pp. 536-40, Aug. 1945

(None given)

P-008
Parker, E.J.
How O-Rings Seal Air And Oil Components
Applied Hydraulics, V. 10, pp. 121-124, May 1957

This paper covers the theory, advantages and limitations of O-rings together with certain application problems: surface, temperature, contamination, etc. O-rings have been used up to 50,000 psi.
A glossary of O-ring terms is included.

P-009
Parker, E.J.
How To Design For O-Ring Seals
Applied Hydraulics, V. 10, pp. 84-87, 134-135, June 1957

Most published data for O-rings follows accepted aircraft standards based on satisfactory sealing at -65°F and using breathing cylinders. This imposes cost penalties on equipment without these same operational requirements.
This article provides details of gland design to assure proper sealing action on specified jobs.

P-010
Parker, E.J.
Prevent O-Ring Failure
Applied Hydraulics, V. 10, pp. 152-154, 156-158, Oct. 1957

O-rings may be used for many and varied applications as hundreds of O-ring compounds are manufactured to handle specific service environments.
A chart of 19 such compounds, their major uses as O-rings and their temp. range is covered.
Possible reasons for failure are also included.

P-011 B
Parker, S.M.
Seals
Chapter 1 of Book "Aircraft Hydraulics, V. II - Components Design," pp. 1-19, Conway, H. (Ed.)
Publ. by Chapman & Hall, 1957

P-012
Parrish, W.A.
Synthetics Hold Aromatic Fuels
Aero. Digest, V. 52, pp. 62, 155, Feb. 1946

Gaskets and seals made of plastics exhibit resistance to oil and aviation gasoline.

P-013
Parsons, C.A.
Labyrinth Packings
Engrg., Jan. 10, 1908

Describes a packing adopted by C.A. Parsons to diminish leakage losses in his steam turbine, and gives formula for calculating the discharge through the packing.

P-014
Pate, M.T.
Reducing Leaks In Pipeline Flanges
Power, V. 82, p. 113, February 1938

Flanges on 8 inch main line. Normal working pressure of 675 lbs. per sq. in. produced a total endwise load on flange bolts of 19,000 lbs.-29,000 lbs. As a result of investigation, metallic gaskets and joint rings were removed from all points subjected to pulsating pressures, and replaced with plastic materials. A further improvement was made by cutting the inside diameter of gasket material from 1/8 to 1/4 in. smaller than the inside diameter of the flanges.

P-015
Paterson, M.S.
O-Ring Piston Seals For High Pressure
Journal Scientific Instruments, V. 39, No. 4, pp. 173-4, April 1962

Describes three types of piston seals, and gives advantages of each. Discusses several other types. Drawings included.
O-ring seals; 10,000 atmospheres pressure, bevel rings of bronze and beryllium copper.

P-016
Paxton, R.R., Shobert, W.R.
Testing Carbon For Seals And Bearings
Lubrication Engineering, V. 17, pp. 27-33, Jan. 1961

A test method which has been of practical value in screening carbon-graphite materials for seals and bearings is described. Details of relatively simple tests are given. Typical experimental data are presented illustrating the use of this test stand to obtain performance data on one carbon base material running non-lubricated.

P-017
Payne, F.E.
Roll Neck Seals - Their Development And Application
Steel, V. 123, pp. 92, 94, 98, 102, Nov. 29, 1948

Steel mill roll bearing sealing problems increased with advent of high speed mills. Early attempts at solution included conventional stuffing boxes, piston rings, lip seals (single and double) and finally vertical plane and face seal or the "John Crane" roll neck seal, described in text.
Materials included felt, flax, semi-metallic packings, treated leathers, etc.

P-018
Payne, G.F.
Rubber And Asbestos In Engineers Jointings And Packings
Chem. Age (Lond.), V. 38, pp. 63-4, Jan. 22, 1938

P-019
Pearl, D.R.
O-Ring Seals In The Design Of Hydraulic Mechanisms
Mach. Design, V. 19, pp. 97-102, May 1947

Practical information on their design, characteristics, and application in hydraulic mechanisms is given as are special uses.

P-020 T
Pearl, D.R.
O-Ring Seals In Design Of Hydraulic Mechanisms
Trans. SAE, V. 1, No. 4, pp. 602-611, Oct. 1947

A thorough work on O-ring seals covering characteristics, theoretical aspects, moving O-ring seals, rotary shaft seals, types of seal failures, design procedures. Sketches and data curves included.

P-021 T
Pearson, J.D.
Turbine Glands
Trans. Inst. Marine Engrs., V. 44, Part 5, pp. 219-223, 1932

A review of the water filled gland seal and the labyrinth seal. The discussion of the labyrinth seal is basically that due to Stodola, and a number of seal geometries presently used in steam turbines are shown.

P-022
Pennington, J.W., and others
High Speed - High Temperature Shaft Seals
SAE Paper No. 687, p. 12, Jan. 9-13, 1956

Problem of selecting and designing fluid flow barriers against rotating shafts in gas turbines; basic designs of several seal types discussed have provided sealing ability required in most turbine applications.

P-023
Perry, J.H.
Choosing Gaskets For Use In Chemical Plants
Chem. & Met. Engng., V. 41, pp. 194-198, April 1934

The importance of the gasket in a flanged joint, unit pressure and the bolt load; twenty commonly used types of formed and sized commercial gaskets; proper application and installation of gaskets.

P-024
Perry, J.T.
Mechanically Sealed Beryllium Windows For Permanently Evacuated X-Ray Tubes
Review of Scientific Instruments, V. 27, pp. 759-762, Sept. 1956

In the tubes described, the Beryllium window, which may be as thin as 0.25 mm., is clamped with a soft metal gasket to a steel window frame. The frame is actually a flange, copper-brazed to a nickel-iron sleeve which is sealed to the side of the glass envelope of the tube, and the window and gasket are clamped down by a second steel flange with 12 fixing screws. Some data are presented on the performance of the tube.

P-025 T
Peters, A., and Sciulli, E.B.
A Bibliography On Gas-Lubricated Bearings
The Franklin Institute Laboratories, Interim Report I-A-2049-16, Oct., 1961, OTS No. AD 264965

The results of an extensive survey was part of a government sponsored program of research on gas-lubricated bearings. The survey included all available references, many foreign, on the subject of gas-lubricated bearings.

P-026 T
Peters, C.W.
Word About Turbine Packing
Power Plant Engineering, V. 39, pp. 586-7, Oct. 1935

A brief descriptive article of a shaft packing failure of a steam turbine generator which almost threw a unit from its base.

P-027
Peters, J.L.
Mass Spectrometer Tests Tightness Of Seals
Electronics, V. 33, pp. 74-76, April 1, 1960

Production-line leak-detecting mass spectrometer is usable in electron tube manufacturing. Unit achieves high precision with two magnetic analyzers in series. It determines leak size by measuring quantity of helium escaping. Description of equipment, operation, performance and industrial use. Drawings, curves, and data.

P-028
Peters, R.L.
Lubrication Of Rocket Engine Pumps Utilizing High Energy Propellants
Sci. Lub., V. 14, pp. 26-29, April 1962

General problems pertinent to the design of a hydrogen pump are initially discussed. Specific oxidizer problems as presented in design of a hydrogen pump are next discussed and finally the fluorine pump problem is treated. Seal isolation in the fluorine pump is accomplished by use of a secondary oxidizer which provides a barrier between the fluorine (extremely corrosive) and the seal.

P-029
Peters, V.
Notes On Hydraulic Packings And Stuffing-Box Design
Iron Age, Dec. 15, 1904

Describes several constructions of hydraulic cylinder and plunger packings and their practical applications, and gives hints for selecting and designing satisfactory packings for parts moving under hydraulic pressure.

P-030
Petersen, A.B.
Development Tests Of Pneumatic Equilibrators With Experimental Neoprene Packings (O Type) For 155MM Carriage M1, 2nd Report; 37th Report On Ord. Pro. 5084
PB 33949, Off. Tech. Sev., 76 pp., Jan. 1945

The results of tests to determine if the experimental "A" type neoprene packings are suitable and satisfactory for the prevention of nitrogen leakage, and whether they are sufficiently superior to its "T" type neoprene packing to warrant adoption as a standard in the pneumatic equilibrators, are contained herein.
It was found that nitrogen and oil seal leakage were not prevented. Photographs, drawings, and tabulated test results are presented.

P-031
Peterson, L.M.
Sealing Rings From German Engine
PB 2943 - Off. Tech. Sev., 3 pp., 1945, (Army Air Forces, Eng. Div., T.S.E.P.L. 5-503-3080)

Memorandum report on the physical and chemical properties of synthetic rubber cylinder sleeve sealing rings from Junker Juno 213 aircraft engine.

P-032
Petrie, E.C.
Ring Joint; Its Relative Merit And Application
Heating and Piping, V. 9, pp. 213-220, April 1937

Data on bolt, gasket, and pressure are presented in connection with the application of the ring joint to high pressure and high temperature service.
General characteristics of ring joint connection.
Effective pressure area of ring joints.
Results of comparative tests of ring and flag gasket joints.
Determining width of ring sections for ring joints.
Application of ring joints.
Data and Photographs.

P-033
Pfefferle, G.H.
Distribution Of Gasket Pressures In Pipe Joints And Clamps
Gas Age, V. 73, pp. 379-82, 386-8 +, April 21, 1934

P-034
Pheil, G.D.
Air Curtain Shaft Seal
Prod. Engng., V. 31, p. 24, Dec. 26, 1960

An "air curtain" shaft seal similar to those used to keep cold air out of doorless supermarkets and public buildings is proposed. Air would be pumped through the center of the shaft and out of a grid or porous bronze section, in the shaft. It would provide an effective seal between enclosed areas of different temperature, pressure or humidity and would not have to be lubricated or replaced because of wear. At high rpm, radial holes could be angles so that centrifugal force would automatically pump the air and contribute to the effectiveness of the seal.

P-035
Phillips, M.J.
Design Of Piston Type Accumulator Uses Positive Type Pressurized "O" Seals
App. Hydraulics, V. 4, pp. 26-27, Feb. 1951

A simple design provides a well lubricated seal with a higher sealing pressure than the existing pressures on both the gas and oil sides of the piston.

P-036
Pieckii, V.L., and Christensen, D.A.
How To Choose A Dynamic Seal
Product Engineering, V. 32, pp. 57-68, March 20, 1961

Describes and shows operation of 5 basic types: shaft seals, face seals, compression packing, molded packings, and labyrinth or positive-clearance seals.
Analyzes each type for basic selection factors, materials, and design recommendations.

P-037
Pieckii, V.L.
Selection Factors For Radial-Type Oil Seals
Machine Design, V. 34, pp. 264, 266, 268, 270, April 26, 1962, Abstract from SAE Paper 473B, Jan. 1962, 13 pp., "Oil Seal Performance: a Key to Reliability," see also SAE Journ., V. 70, June 1962, pp. 57-63

Selection factors for dynamic elastomer seals for sealing in lubricants. Factors are seal materials and section shapes. Effect of various cross-sectional shapes for sealing elements is discussed.
Material considerations for increasing seal life:
1. Choice of elastomer. This is determined by type of lubricant and mean temperature.
2. Curing agent used in vulcanization of elastomer.
3. Reinforcing agent for elastomer to improve abrasion resistance, dynamic stability and moldability.
4. Additives to lubricants, especially antioxidants and detergents.

P-038 B
Pillai, G.
Elastomers For Hydraulic Packings
Hydraulics and Pneumatic, V. 14, pp. 80, 82-3, March 1961

Reviews the manufacturing processes, compounding moulding and finishing of elastomeric materials. Attention is drawn to the importance of seal fluid compatibility with water, polyglycols, organic phosphates, silicones, chlorinated hydrocarbons and petroleum hydrocarbons.

P-039 B
Pillsbury, R.D.
Plastic Piston Rings
Machine Design, V. 33, pp. 147-9, September 28, 1961

Use a seal material capable of operating without lubrication to solve the problem of product contamination from lubricants used in reciprocating sealing devices.
Configurations, Material properties - Piston design.

P-040
Pirani, M., Yarwood, J.
Principles Of Vacuum Engineering
Reinhold Publishing Co., N.Y., 1961

Vacuum Plumbing: Seals and Sealing Chapter 4, pp. 146-213. Demountable Seals:
4.15 Vacuum Greases
4.16 Elastomer Seals
4.17 Rubber Gaskets and O-rings
4.18 Vacuum Pipe Connectors
4.19 End Seal Plates
4.20 Lead in Seals
4.21 Rotary Vacuum Seals
4.22 Rectangular Rubber Gaskets
4.23 Gaskets Supported by Metal Inserts
4.24 The Wilson Seal & the Gaco Seal
4.25 Silicone Rubber Gaskets
4.26 Teflon Gaskets
4.27 Metal Gaskets

P-041
Platt, H.
You Can Pack Centrifugal Pumps To Hold High Pressure And Temperature
Power, V. 88, pp. 8-10, 62a., Jan. 1944

Paper explains methods and materials used to pack centrifugal pumps for subject conditions. Materials are braided cotton, braided asbestos, molded plastic asbestos with rubber or neoprene binder, lead foil with asbestos core, aluminum, copper or phosphor-bronze foil. For high pressure or high temperature control of stuffing box, a sealing liquid is circulated through lantern rings to absorb heat and provide lubrication of the packing.

P-042 T
Ploeger, C.E.
Compressor Shaft Seal
U.S. Patent 2,290,813 July 21, 1942

A mechanical seal design which is self-aligning and which can be easily demounted for replacement or repair.

P-043 B
Plutalova, L.A.
Graphite Seals For Rotating Shafts
Vestnik Mashin (U.S.S.R.) 2, pp. 3-8, 1957 (In Russian) (United Kingdom-Atomic Energy Authority translation No. LB Information series 64(RD/R) 1959, also LC or SLA No. 61-13618

Russian progress in the field of split face seals with apparent success with pressures up to 150 lb/sq. in. Shaft diameters 4 and 6 cm and speeds between 600 and 3000 rpm were used during experimental work.

P-044
Pohl, E.
Damages To Steam Turbine Stuffing Boxes (In German)
Maschinenschaden, V. 15, pp. 117-123, 1938

Illustrated account of various instances of failures of labyrinth packings.

P-045
Pohl, E.
Operating Experiences With Packings And Equalizing Discs Of The Ljungstrom Turbine
Maschinenschaden, V. 17, pp. 81-86, 1940

Economical and safe operation of the Ljungström turbine depends to a considerable extent on the ability of the packings to stand up under various service conditions. Paper discusses the design of the elements in the packings and equalizing discs, and the various causes of mechanical wear and corrosion.

P-046 B
Pollard, F.H.
The Development Of High Temperature Hydraulic Systems For Aircraft
Proc. 13th National Conference Industrial Hydraulics, V. 11, pp. 57-65, 1957

Considers temperature categories, packing development, back up rings, and static seals. Test results on Buna N, Neoprene, acrylics, and Viton A. (Vinylidene Fluoride Hexafluoropropylene Copolymer) temperatures -65 to 500°F. Pressures to 3000 psi.

P-047
Pollard, F.H.
Design Tips For Using Viton
SAE Jour., V. 67, pp. 80-81, May 1959, Abstract of SAE Paper 50R, "Designing for use of Viton O-rings at elevated temperatures," from meeting of March 31, 1959, 10 pp.

Dupont Viton A, vinylidene fluoride hexafluoropropylene copolymer, has a usable thermal range of -65 to 600F. Its characteristics of hardness, mold shrinkage, fluid effects and extrusion gap are assessed for O-rings for seals in hydraulic cylinders and other components of supersonic aircraft.

P-048 T
Popper, B., Reiner, M.
The Application Of The Centripetal Effect In Air To The Design Of A Pump
British Jnl. App. Physics, V. 7, pp. 452-3, Dec. 1956, (correction) V. 8, pp. 493-4, Dec. 1957

The authors describe a device to make use of the centripetal effect in air in an air pump.

P-049 B
Porges, F.B.
Mechanical Seals In The Petroleum Industry
Proc. 3rd World Petrol. Conf. (The Hague), pp. 222-35, 1951

After discussing the principle of this type seal, the author illustrates and describes in detail several leading contemporary designs and compares them. Description of two cases in which a face seal became more economical than a stuffing box in spite of its high initial cost.

P-050 T
Poetler, J.W.
Pressure Compensating Seal
Chemical Engineering, V. 54, pp. 118-19, June 1947

Suggest the use of a double mechanical face seal for higher pressures, where a pressure of half the sealed pressure is introduced between the seals.

P-051 T
Pratt, W.E.
Selection Of Pumps For Chemical Service
Ind. and Eng. Chemistry, V. 31, No. 4, pp. 408-15, April 1939

A general discussion of the importance of pump selection procedure, special attention to material and alloy of component parts. Seal commentary limited to stuffing box with water as a lubricant.

P-052
Pratt, W.E.
Unusual Stuffing Box Problems Encountered In Acid Pumping
Chem. and Met. Eng., V. 50, pp. 109-13, April 1943

The main concern of this article is to analyze some of the troubles that occur on certain acid pump installations where all operating conditions appear to be favorable, and yet recurrent serious difficulty persists. Illustrations of shaft seals.

P-053
Pratt, W.E.
How Carbon Rings Saved $114 Per Year In A Troublesome Centrifugal Pump Problem
Chem. and Met. Eng., V. 52, pp. 116-17, July 1945

Pumping hot acid with abrasive solids in suspension is a severe service. The problem was to bring about a condition which would prevent any contact between the bottom rings of packing and the shaft shroud, which was an integral part of the impeller. This could be accomplished by the use of packing rings in which the solid abrasive particles could not imbed themselves. Details of a case in point and its solution. Photographs included.

P-054 T
Primeau, G.H
Seal
U.S. Patent 2,977,139 March 28, 1961

Seal for automatic transmissions. A tetrafluoroethylene sealing member is maintained in sealing engagement with a rotating surface by a garter-spring. The sealing member is held stationary by a two-coil helix retaining ring.

P-055
Provaznik, F.
Gas Seal Of Hydrogen-Cooled Electric Machines
Elektrotech, Obzor, 44, No. 5, 221-4, 1955 (In Czech).

The tightness of the gas seals of hydrogen-cooled electric machines is investigated quantitatively by a method based on the Van der Waals' equation. The permeability of the seal, expressed in M^3 gas escaped during 24 hours, is determined from the initial and final parameters of the gas compressed in the seal to about 1-2 atm. above gage pressure. The effect of the kind of gas and its pressure are also investigated.

P-056
Puchstein, A.F.
Sealing Components For Shafts And Bearings
Elec. Mfg., V. 31, pp. 106, 108, 110 etc., March 1943

Reviews types of seals suitable for the sealing of rods, shafts and bearings.
Materials: fibrous, leathers, natural and synthetic rubbers, cork, plastic, asbestos, carbon, graphite, metallic, etc.
Materials suitable to hold or exclude low medium or high pressure at low medium or high temperatures up to 1500°F. Compatible with a variety of materials v.g. oils, gasoline, acids, etc.
A variety of seal types are discussed, labyrinth, cup, piston, rings, etc.

P-057
Purdy, R.B.
Get The Most Out Of Your Packing
Power, V. 77, pp. 624-5, Dec. 1933

Proper packings to use; what causes a packing to leak; how to install a packing on pump or engine rod.

P-058
Purves, T., Jr.
Packing For Piston Rods And Valve Stems
Ry. Review - 1200, May 16, 1896

A review of a paper before the New England Railroad Club. The conclusions are favorable to metallic packings in preference to hemp, and some figures of maintenance cost are given.

P-059 T
Quinn, N.R.L.
Sealing Device
U.S. Patent 2,814,512 Nov. 26, 1957

In a thin shaft seal, which is related to the face seal, the rotating seal surface is a radial surface perpendicular to the shaft. The stationary seal ring then forms a converging passage in direction of pressure decrease. The result of the passage configuration is a force tending to open the gap. With this device, by regulation of the intermediate pressure, any desired normal operating gap may be maintained.

R-001
Rabinow, J.
Developing Seals For Abrasive Mixtures
Mach. Design, V.23, pp. 128-131, June, 1951

Describes problems attendant to the development of a shaft seal for a magnetic-fluid clutch.
Various types of seals were tested, such as standard rubber packing, O-rings, felt seals, and special Teflon seals, but none were successful.
A successful seal was finally designed based on a well-known magnetic separator principle. Rubber seal rings were used in this seal.

R-002
Rademaekers, W.A.
Seal-Ups Fitting Reliability
Aircraft and Missiles, V.4, pp. 42-4, Feb., 1961

Description of a double seal-type fitting for aircraft and missile fluid systems. The new fitting gives reliability at high temperatures and pressure. Factors affecting design of such fitting are discussed briefly.

R-003
Raether, H.
Note on "Simmer Rings," Vacuum-Proof Packings, Used in Laboratories for Actuating Movements Within Vacuum Apparatus (In German)
Zeit.für Technische Physik, V.23, pp. 266-6, 1942

Sealing arrangement consists of cylindrical jacket and internal sleeve, both of rubber-like material called "Simrit"; jacket is stiffened with metal ring; sealing of sleeves is effected with an aid of "packing lip" likewise of "Simrit;" because of its flexibility, this lip clings to spindle and prevents air leakage.

R-004 T
Rainey, R.S.
Recent Developments in Fluid Sealing
Automotive Industries, V.100, pp.283-4, March 15, 1949

Description of new improvements in standard leather and rubber lip-type shaft seals, which could be used in many instances where more expensive sealing methods are now used.

R-005
Rainey, R.S.
Engineering Oil Seals to the Product
Tool Engr., V.23, pp. 29-31, Oct., 1949

Relative values of leather and synthetic rubbers as oil seals are discussed. Application norms are given. Maximum temperature considered 300°F.

R-006
Rainey, R.S.
Which Shaft Seal?
Prod. Engng., V.21, pp. 142-147, May, 1950

Tabulates the characteristics of the various types of seals such as springless, spring-loaded, dual, and face. Materials: leather, rubber, synthetics. Application problems are discussed.

R-007
Raisbeck, L.R.
Investigation of Hydropneumatic Recoil Mechanism Packing Spring Loads
PB 174730, Off. Tech. Serv.

Conventional packings for Howitzer Hydropneumatic Recoil Mechanism floating pistons were treated to determine the relationship between axial packing load and radial packing pressure at sub-zero, normal, and elevated temperature. Tests were conducted with leather - silver - rubber and Teflon-aluminum-rubber packings. Both types were tested in a 105 mm Howitzer Hyd. pneumatic Recoil Mechanism to determine their sealing effectiveness.

R-008 B
Ramadanoff, D. and Sherlock, J.J.
Mechanical Carbon Graphite: A Unique Material for Rubbing Surface Applications
A.S.L.E. Paper 61 AM, 1B-2 (7 pp., 1 fig.), April, 1961

R-009
Rand, W.M., Jr.
New Flowed-In Gaskets
Materials and Methods, V. 37, pp. 78-79, Jan., 1953

Called the "flowed-in" gasket, it is applied as a liquid synthetic rubber or resin compound forced through a nozzle onto a spinning component part and then baked to form a solid rubbery gasket.

R-010
Rand, W.M., Jr.
Mechanical Applications of Flowed-In Gaskets
Mechanical Engineering, V.75, pp. 541-544, July, 1953

Describes applicable materials and uses. Diagrams.

R-011
Rand, W.M., Jr.
The Latest Development in Flowed-In Gaskets
Product Engng., Annual Handbook, pp. J.30-J.31, 1954

New flowed-in gaskets produced by pouring liquid gasket material into place, and baking the compounds to solid films, are described.
Requirements of material and equipment covered.

R-012
Rankin, D.R.
Using Mechanical Seals for Centrifugal Process Pumps
Heat, Piping and Air Cond., V.23, pp.102-105, Oct., 1951
Petroleum Engng., V.23, p. C.13+, Sept., 1951

Discusses in some detail the application of pump seals, and gives practical advice on their proper use in industrial processes. Seals are limited to temperatures below 250°F with standard materials and 450°F with special ones. Operating norms for balanced and unbalanced seals are listed.

R-013
Rankin, D.R.
Why, When, Where to Use Mechanical Seals for Centrifugal Process Pumps
Industry and Power, V. 61, pp. 76-78, 117, Dec., 1951

Surveys the problems and characteristics of the above. Illustrations and diagrams.

R-014 T
Rankin, D.R
Gives Mechanical Seals A Chance
Chemical Eng., V. 59, pp. 152-3, January, 1952

Lists the advantages and limitations of mechanical seals. Temperature and pressure ranges presently sealable are given. Also an explanation of the difference between balanced and unbalanced seals.

R-015
Rasmussen, R.
Method of Sealing Flared Tube Fitting Used With Permeative Fluids
Parker Appliance Co., pp. 1-4, March 21, 1957

To compare the performance of Teflon-coated fittingnoses to standard machined noses.

R-016
Ratner, A.V. and Zelenskii, V.G.
Erosion of Seals in Water and Steam Fittings
Teploenergetika, No.12, pp. 28-32, 1957.
Main Library Translation TT-4755, General Electric Co.

Data are given on the erosion of various metals by water and superheated steam. Temperature 480°C, at a velocity of 150 m/sec.

R-017
Ratner, A.V., Kagan, D. Ya.
Investigating Corrosive Action in Packings
Trans. of Teploenergetika, USSR, V. 8, No. 5, pp. 35-39, 1961
(Translation No. FT612, available from Faraday Translations, New York, N.Y., $11.75 ea.)

Description of gaskets, hydraulic seals, pneumatic packing, corrosion research.

R-018
Raymond, E.T., and Hull, J.W.
Aircraft Reliability Demands Leakproof Hydraulics
Applied Hydraulics and Pneumatics, V. 11, pp. 74-77, Oct., 1958

In designing aircraft hydraulic systems for high reliability, the prevention of fluid leakage is one of the most important factors.
Minimizing the number of leakage points is one of the principal steps in designing a leakproof system.
Elastomeric O-rings are still considered best for most seal applications, although metal seals are contemplated for future aircraft and missile applications.
Bellows-faced seals are used for many applications.
Two-stage seals improve reliability when applied to O-ring and metal seal configurations and used in conjunction with a pressure drop bushing.
Ring seal tolerates angular misalignment between the rod and seal gland. The choke seal designed for low friction has worked very satisfactorily in laboratory tests. Both are metal seals and show significant advantages over elastomeric O-rings.

R-019 T
Raymond, E.T.
High Temperature Hydraulic Seal Developments for Future Flight Vehicles
Int. Conf. on Fluid Sealing, Paper G1 (26 pp.), April 17-19, 1961
British Hydromechanics Research Assn., Harlow, Essex, England

Problems involving static and dynamic seals due to high temperatures encountered in high-speed flight are discussed. Promising high-temperature sealing methods are discussed.

R-020
Reeder, A.
Hydraulic Packings
Am. Mach., V. 29, pp. 322-323, Sept. 6, 1906

Illustrated examples of packings and types of connections, giving explanations.

R-021 T
Reiner, M.
The Centripetal-Pump Effect in Air
Proc.Royal Soc. of London, Series A, V.240, pp. 173-188, 1957

An instrumental arrangement is described which constitutes a centripetal air pump. It can be seen that air is pumped from the outer atmosphere into the pump cylinder. Air is in a state of stress including elastic cross stresses. Definition in Hencky's logarithmic measure. Confirms Maxwell's theory that air is elastoviscous. A rheological equation for air is proposed.

R-022 T
Reiner, M.
The Centripetal-Pump Effect in a Vacuum Pump
Proc.Royal Soc. of London, Series A, V.247, pp. 152-167, 1958

An instrument is described which works as a centripetal vacuum pump. The rotor can be freely displaced along its axis of rotation. By centripetal pumping action it is kept floating on an air cushion of thickness "D". The relation between "D" and speed of rotation was determined for different rotor weights supported. The results are interpreted on the basis of Maxwell's theory of the elastoviscosity of air.

R-023 T
Reissenger, H.
Two and Three-Dimensional Flow of Viscous Incompressible Fluids, Without Inertia, Between Relative Rotating Eccentric Cylindrical Surfaces (In German)
Zeit. für Angew. Math. und Mech., Report I, V. 15, pp. 81-87, 1935;
Report II, V. 16, pp.275-286, 384, 1936

The author derives a general equation for pressure function and flow function in the annull, which be solves for the case of a narrow annulus. For evaluating bearing load force and frictional moment, the basic stress relationships according to Stokes are presented. Report I, two-dimensional flow. Report II, three-dimensional flow.

R-024
Resek, R.B.
Hydraulic Actuator Rod Seal for High Temperature Operation
PB 135219, Off.Tech.Serv., USAF-WADC-TR-56-268

A rod seal was required for hydraulic actuators which would operate with fluid and ambient temperature as high as 700°F. A test actuator and test stand were designed and built for testing sample seals from -65°F to 400°F. Both static and cycling tests were run with metallic seals, tubular metal seals, and several types of asbestos seals.

R-025
Reuss, H.
Problem of Sealing of Ships Shafts (In German)
Werft-Reederei-Hafen, V.22, pp. 273-5, Sept.15, 1941

Selection of seals in relation to type of bearings; illustrated description of various types of seals which result in considerable savings in costs and valuable metals.

R-026
Reynolds, F.L.
Use of Indium in High-Vacuum Equipment
Atomic Energy Commission Report V.C.R.L. 2989, May, 1955

Three methods of using indium for gaskets are described: 1) indium wire, 0.050" diameter, is placed in a groove between the flanged joints to be joined; 2) indium is soldered to the flange surfaces and pressure sealed, 3) indium is coated onto a copper ring gasket and a knife edge structure is employed for the metal-to-metal contact.

R-027 B
Rheingans, W.J.
Recent Developments in Francis Turbines
A.S.M.E. Paper 51-A-101 (15 pp., 15 fig.), Nov., 1951
Mech. Engng., V.74, No.3, pp. 189-96, March, 1952
Disc. Mech. Engng., V.74, No.6, pp.510-2

A carbon ring seal is illustrated which can replace the stuffing box used in the past. Advantages are indicated.

R-028 T
Rhodes, A.F., Eichenberg, R.
Pressure-Energized High Pressure Gaskets
Int.Conf.on Fluid Sealing, Paper B4 (16 pp.), April 17-19, 1961
British Hydromechanics Research Assn., Harlow, Essex, England

Describes the pressure-energized gasket design arising from further development of the AWHEN-API gasket and flange design previously presented to the American Society of Mechanical Engineers in ASME Paper 57-DET-23, by Robert Eichenberg.

R-029
Rice, C.W.
Liquid Film Seal for Hydrogen-Cooled Machines
General Electric Review, V.30, pp.516-530, Nov., 1927

The type of seal studied by the author consists, in its rudimentary form, of a metal ring surrounding the shaft with a small clearance between them; a groove is turned in the median plane of the ring which is provided with an opening through which oil under pressure is forced into the groove. It flows axially through the annular clearance spaces on each side of the groove, one toward the hydrogen and one toward the air. Data, drawings, and test results.

R-030 T
Rice, C.W., Sneil, D.S.
Liquid Film Seal
U.S. Patent 2,236,274, March 25, 1941

A shaft seal on a gas-cooled machine which utilizes the bearing as part of the seal. Lubricant and identical sealing fluid are pumped under pressure higher than gas pressure to annular chamber formed between the bearing and an additional bushing. The sealing fluid which contacts the sealed gas is drained off separately.

R-031
Richards, A.E.
Packing
Modern Power & Eng., V.44, pp.79-80, Dec., 1950

Packing defined as material used to lessen or prevent leakage of fluid or gases between moving parts; suggestions regarding selection of packing to fit particular process and equipment; problems of high-speed packings; packing pointers.

R-032
Richards, C.B., Smith, J.R.W.
Demountable Vacuum Seal
Journal Scientific Instruments, V.31, pp.431-2, Nov., 1954

Drawing of the simplest version of the seal as used to close the end of a glass tube. Basically, it consists of a metal plate and retaining ring holding an O-ring, the section of which is constrained on three sides by the metal and presents the fourth side to the square cut, flame polished end of the tube. The force exerted on the plate by the pressure of the atmosphere is sufficient to compress the O-ring against its constraining surfaces and make the seal vacuum tight.

R-033
Riddle, J. and Durrett, R.D.
Face Type Rotating Shaft Seals
Mill and Factory, V.63, pp.87-90, Sept., 1958

Mechanical axial seals are divided into two types, depending on whether the spring-loaded seal ring is stationary or rotary.
The most common applications of both types are discussed, together with their advantages and disadvantages.
A fluid compatibility chart of several elastomers is included.

R-034 T
Riesing, E.F., Klein, H.H.
Radial Shaft Seals
Mechanical Engng., V.73, pp.820-1, Oct., 1951

Commentary on unitized radial contact shaft seals operating on shafts or in boxes, rotating from 1 rpm to 10,000 rpm. Peripheral speeds from less than 1 fpm to 5000 fpm; temp. -60°F to 400°F; 300 psi to negative 14 psi.

R-035
Riley, M.W.
Treated Felts
Materials and Methods, V.44, No.6, pp. 90-93, Dec., 1956

Description of felt treatments consisting of chemical modification, impregnation, lamination, surface coating and combinations of these treatments; their use to change basic physical properties of felt; to protect felt; or improve forming characteristics and simplify fabrication.

R-036
Robbins, C.G.
Metallic Packings
Power, N.Y., V. 22, pp.15-24, 39 fig. April, 1902

Descriptions are given of the principal packings now in use in the United States. 15 different designs.
United States Metallic Packing: babbitt, bronze, metal rings
Swain metallic packing, spring loaded and inclined surfaces
Jerome metallic packing, spring loaded, cast iron
France metallic packing, spring loaded on outer circumference
Walker metallic packing, encircling springs
Paragon metallic packing
Harthaus metallic packing
Deeds metallic packing, 4 blocks, antifriction, beveled joints
Holmes metallic packing
Katzenstein metallic packing
Faultless metallic packing
Tripp metallic packing
Sterling metallic packing
Donning metallic packing
Udstad metallic packing

R-037
Robbins, C.G.
Observations on Metallic Packings
Marine Engng., V.7, pp.337-9, July, 1902

A comparative study of the theories of fibrous and metallic packings and the reasons for the efficiency of the latter. Ill.

R-038
Roberts, F.
Increasing the Efficiency of Piston Rod and Throttle Packing
Railway Mech. Eng., V.95, p. 520, Aug. 1921

The use of wedge rings will effectively compress packing rings long after they have lost their resilient properties. The wedging action forces the packing ring against the rod, effectively preventing the escape of steam.
Drawing included.

R-039
Roberts, I.
Gaskets and Bolted Joints
Jour.App.Mech., V.17, pp.169-179, June, 1950

Consists of a study of loading requirements of gaskets in bolted joints with the object of developing a rational basis for design of such joints.

R-040
Roberts, R.W.
Rotary Vacuum Seal
Brown Univ., M-6845, Contract AT (30-1)-2101 (4 pp.), Jan. 13, 1959

A brass rotary vacuum seal is described for use in a molecular beam experiment.

R-041
Roberts, R.W.
Rotary Vacuum Seal
Review Scientific Instruments, V.32, pp. 750-1, June, 1961

It was necessary to rotate a 40-lb. helical velocity selector at speeds up to 4200 rpm in a metal high-vacuum chamber. This was achieved by a rotating shaft going through a vacuum seal in the end flange of the vacuum chamber. The seal was divided into three sections by two No. 63X3 silicone rubber Garlock "Klosure" seals, and two silicone rubber O-rings.
The center section is exhausted to a pressure of 1 x 10^{-3} torr by a mechanical pump. The high vacuum chamber pressure is 1 x 10^{-7} torr. At 1000 rpm, the pressure is 7 x 10^{-7} torr. Details and drawing included.

R-042
Roberts, V.
Coolable Vacuum-Tight Window Seals for Optical Use
J. Sci. Instrum, V.36, p.99, February, 1959

A flange is permanently soldered to the specimen chamber. An annular polyethylene gasket 0.005" thick is clamped between the flange and the window diaphragm by a retaining ring. The assembly is held together by sixteen screws. Seals of this type have been frequently used at liquid hydrogen and helium temperatures.

R-043
Robertson, D.D., Colvin, R.H.
Why Valve Packing?
Bus Transportation, V.15, pp.568-9, Dec., 1936

R-044 T
Robinson, C.S.L.
Flow of a Compressible Fluid Through a Series of Identical Orifices
ASME Trans., V.70, pp. 308-310, 1948

The author has used perfect gas laws and the assumption of constant initial enthalpy at each orifice to provide a straightforward method of flow calculation for air, saturated and superheated steam. Permissible errors in specific heat ratio K produces only small error. Method graphical and useful in design. (No experimental results included.)

R-045
Robinson, J.J.
Evaluation of Dyna-Seals for Use as Sealing Gaskets on Aircraft Gear Motors
DF61MD314 (5 pp.), GEL Tech.Data Center, G.E. Co., Schenectady, N.Y., April 26, 1961

Standard "Dyna-seal" gaskets were evaluated for use as a sealing gasket in conjunction with AN-type oil fill and drain plugs on aircraft-type gear motors utilizing MIL-L-7808 oil as a lubricant. It was concluded that the dyna-seal may be used for the stated purpose. A standard 110-5/16 Dyna-seal (product of Precision Rubber Products Corp., Dayton, Ohio) was evaluated.
Tests have indicated that a more positive seal for MIL-L-7808 oil was obtained when this dyna-seal was substituted for the present AN-900-5 copper asbestos gasket.

R-046
Robinson, J.N., Lundin, M.I., and others
Development of Ring Joint Flanges for Use in the H-RE-2
ORNL-TM-68 (51 pp., 53 ref.), Oak Ridge National Lab.

Report on Contract W-7405-eng-26.

R-047
Robinson, N.W.
Bakeable High Vacuum Seals
J. Scientific Instruments, V.34, p.121, March, 1957

Plane flanges can be joined and baked repeatedly to $450^{\circ}C$ using an all-metal seal which has been developed for use with high vacuum demountable apparatus. Two flat copper washers are separated by a steel ring which has annular knife edges formed on the upper and lower surfaces and a peripheral groove to allow a cantilever action of the knife edges when under pressure. Suitable dimensions are given.

R-048 T
Rodes, T.W.
Mechanical Seals for High Speed, Temperature and Pressure Applications
Industrial and Engineering Chemistry, V.43, Supp. pp 117A-118A, June, 1951

Increased operating ranges are noted for mechanical seals and credit is given to development of the balanced seal, and improvements in material.

R-049 T
Roeder, K.
Die Abdichtungsaufgabe im Dampfturbinenbau (In German)
Archiv. für Waermewirtschaft, V.18, No.15, pp. 147-150, May, 1937

This article discusses design principles for steam turbine packings.

R-050
Roger, V.
Rotary and Sliding Seals (In French)
Technique Moderne, V.42, pp. 149-53, May 1-15, 1950

Illustrated description of different types: references apply largely to American and British practice.

R-051
Rogers, D.T.
Mercury Seal for Stirrers
J. Am. Chem. Soc., V.5, p. 419, 1933

A Hg seal made of Allegheny metal No. 22 or mild steel coated with paint is good for speeds of 1000 rpm.

R-052
Rogers, P.R.
Pressure Seals for Reactor Mechanical Drives
Nuclear Eng., V.7, pp. 110-113, March, 1962

Details are given of the testing and development carried out on seals for rotating and reciprocating drives for various items of equipment associated with reactor gas circuits, charging machines, and service machines.

R-053
Rooke, N.F.
Aircraft Power Plant Seals
SAE Jour., V.61, pp. 23-29, July, 1953

Secretary's report of round-table discussion on aircraft power plant seals and their application held at SAE meeting.
Problems attendant to sealing against fuel leakage. Temperature range -65°F to 250°F, or higher, are covered.
O-rings, metallic seals, etc., are discussed.

R-054
Roper, R.S., Tipton, F.W. and Trepus, G.E.
Utilization of Plastics in Seals for Extreme Environments
SPE-Annual Tech. Conf. Paper 26-1, 1961

Study relevant to materials for space vehicles; experimental study of high temperatures, ionizing radiation and extremely low temperatures on sealing efficiency of O-rings; filled back-up rings; Mylar film for low temperatures.

R-055
Ross, W.S.
Design for 60 MW Steam Turbine to Take Steam at 1500 PSIG and 1050°F
Instn. Certificated Engrs. (S.Africa), V.32, No.8, pp. 249-259, August, 1959

Design and general construction of turbine; novel features included; methods of overcoming problems in use of available materials and in steam sealing for h-p cylinder and control gear; special reference made to use of austenitic materials where differential expansion is involved; gland sealing and leak-off system; oil systems and control gear.

R-056
Rossheim, D.B. and Markl, G.R.C.
Gasket Loading Constants
Mech. Engng., V.65, pp. 647-648, Sept., 1943

A review of the gasket-loading constants suggested in Table UA-7 of the A.S.M.E. unfired pressure vessel code.

R-057
Roth, A.
Calculation of "Leaky" Vacuum Installations
British Chemical Engng., V.6, pp.40-44, Jan., 1961

Methods of calculating the dimensions of leaky vacuum systems presented in this article include short-cut graphical procedures for estimating the permeabilities of seals, characteristics of vacuum pumps and of the pumping system in the presence of leaks. Also presented is a nomogram for determining the most economical operating conditions.

R-058 T
Rowell, H.S., Finlayson, D.
Screw Viscosity Pumps
Engineering, V.114, pp.606-607, Nov. 17, 1922

Pioneer paper to explain the theoretical basis of the screw pump performance. Assuming laminar flow, neglecting leakage flow past screw-bushing clearance and screw helix angle, the authors present an equation for maximum pressure which can be developed, and the discharge rate for simple screw viscosity pumps.

R-059 T
Rowell, H.S., Finlayson, D.
Screw Viscosity Pumps
Engineering, V.126, pp. 249-250, Aug.31, 1928 pp. 385-387, Sept. 28; Discussion, pp. 659-660, Nov. 23, 1928

This paper deals with the theoretical and experimental aspects of the screw viscosity pump. In an earlier paper by the authors, theoretical equations for pressure rise and flow were derived. Experimental results are presented to confirm the conclusion that pressure rise and discharge rate per unit cross sectional area of the groove increase as the ratio of the groove width to groove depth increased.

R-060
Royston, D.
Hydraulics for High Performance Aircraft - Why and How
Can. Aeronautical, V.5, No.6, pp. 233-242, June, 1959

Steps involved in choosing optimum power supply for flight concepts and utility services of high performance, supersonic interceptor aircraft. Reasons for choice of hydraulics and problems involved such as: choice of fluid, tube connections, reservoir design and temperature problem. Future trends relating to fluids, seals, etc.

R-061
Rudis, M.A.
Strength and Rigidity of Seal Rings
Russian Engrg.Jour., V.41, No.6, pp. 28-31, 1961

Static seals for joints in machines. Establishes the connection between the force of tightening acting on a seal of predetermined dimensions and the pressure, and gives formulas for determining the stresses.

R-062
Ruhl, F.F
High Temperature Face Seals
Mach. Design, V.20, pp.127-132, Jan., 1948

How hydraulic coupling and torque converter seals were developed. Steps leading to the design of successful high-temperature face seals.
Following seals investigated:
1. Metal diaphragm; 2. Face-type seals (single and double); 3. Bellows; 4. Lip-type and others.
Ten years of development in making seal units for hydraulic transmissions provided proof that a carbon-graphite material, "Graphitar," has best qualifications for seal rubbing member.

R-063
Rushing, F.C.
Centrifugal Pump and Shaft Sealing Means
U.S. Patent 2,951,448, Sept. 6, 1960

A description is given of sealing means between a hollow rotatable shaft and a stationary member surrounding the cross-section, comprising a plurality of axially-spaced rings held against seats by ring springs which serve to subdivide the sealing space into a plurality of zones.

R-064
Ruthberg, S., Creedon, J.E.
Aluminum Foil High Vacuum Gaskets
Rev.Scientific Instruments, V.26, p.1208, Dec., 1955

An aluminum gasket clamped between two Monel flanges serves very well as a simple, easily manipulated high vacuum seal which can be baked at $400^{\circ}C$. A thin gasket of this type avoids the problems associated with gasket expansion during bake-out. Flange design and drawings included. Performance record discussed. Temperatures and pressures down to 8×10^{-8} mm Hg.

R-065
Rutledge, F.S.
Lubrication Methods for Rod Packings
Power Plant Eng., V.38, pp.535-6, Nov., 1934

Shutdowns and many other engine and pump troubles can be avoided by proper attention to packing and its lubricants.

S-001 B
Sabanas, M.
Designing Seals With New Materials For Use At High Temperatures
Industrial Laboratories, V.10, pp.16-24, October, 1959

S-002
Sabanas, M.
High-Temperature Static Seals From Metal Fiber Composites
Product Engineering, V. 31, pp. 57-61, May 30, 1960

Seals made by impregnating hard metal-fiber felts impregnated with compatible soft metals or polymers. The most common skeletal metals so far are stainless steel and molybdenum. Some soft metals used are tin, indium, silver and magnesium Air at 5000 psi and 1000°F is sealed by these seals. Mechanical properties and sealing characteristics of various types are given. Discusses the compatibility of materials, fabricating the seals, and the method of impregnation.

S-003
Sadowski, M.A.
Analytic Development of Desirable Stress Distribution Characteristics in Sealing Rings Leading To Recommendation of Ring No.5 for Application to Closed Breech Launchers
PB-146215 Order from L.C. (WVT RR6006) (98 pp.), March, 1960

Laboratory testing of closed breech sealing rings, Nos. 1 and 4, and the mathematical analysis of those rings showed that they are prone to elliptic buckling and plastic flow due to stress concentration at the lips. Ring No. 5 was then designed on the basis of mathematical shell theory. It is believed that it will be as nearly free from those faults as possible. It is suggested that Ring No. 5 be produced and evaluated experimentally.

S-004 T
Saibel, E., Lyman, F.A.
A Theoretical Investigation of Leakage Through Rotary Shaft Seals
Report #850489, U.S. Naval Eng. Experiment Sta., Annapolis, Nov. 28, 1961

It was shown that the effect of face waviness, non-parallelism of faces, or a notch in the seal face could be sufficient to generate a fluid film between the faces. Approximate solutions to the Reynolds' equations are derived by using perturbation methods, and comparison with previously published results shows fair agreement.

S-005
Salzmann, F. and Pravl, P.
Valve Spindle Leakage
Escher Wyss News, V. 10, pp. 88-92, July-Sept., 1937

Methods of estimating leakage at the spindle of steam turbine valves. This article consists entirely of formulae and numerical examples.

S-006
Salzman, P.K. and Schell, F.N.
S2G Valve Sodium Freeze Seal Evaluation
Knolls Atomic Power Lab., Schenectady, N.Y.
KAPL-M-EDL-125 Contract W-31-109-eng-52 (50 pp.), Nov. 20, 1956

The design of the S2G 8-inch sodium stop valve freeze seal was evaluated under established conditions by cycling before and after a simulated bellows failure.

S-007
Samuel, A.J.
Grooveless Flange-Gaskets and Their Installation
U.S. Atomic Energy Commission, Oak Ridge, July, 1944, 13 drawings, AECD 2685, PB-100704, Off. Tech. Serv.

The gaskets are shown in drawings and were designed to be used with 2", 3", 4", and 6" standard ASA 150# forged steel welding neck flanges respectively, having 1/16 inch between flanges. Method of installation, mechanical details of the mold.

S-008
Sancier, K.M.
Vacuum Tight, Flexible Metal Glass Seal
Rev. Sci. Instruments, V.20, p. 958, December, 1949

A sleeve of Saran tubing can be used to form a vacuum tight, flexible seal between metal or glass tubing.

S-009
Sanders, W.E.
The Development of Non-Metallic Packings
Power, V. 29, Aug. 25, 1908

A review of early forms of packing, giving classification, and stating objections to monkey-wrench packings.

S-010
Sanders, W.E.
Moisture and Expansion Packings
Power, V. 29, Sept. 1, 1908

Describes the packings, especially the manufacture of rubber packings Ills.

S-011
Sanders, W.E.
Automatic Steam and Ammonia Packings
Power, V. 29, Sept.8, 1908

A discussion of the advantages, cost, and application of diagonal or automatic packings, and of packings for high temperatures and ammonia.

S-012 T
Sanderson, H.C.
Back Diffusion in Labyrinth Seals
Report KL-1188, Oak Ridge Gaseous Diffusion Plant, Union Carbide Nuclear Co., Oct. 23, 1961

In the seal system proposed for the experimental gas cooled reactor, labyrinth seals were employed to prevent the back diffusion of the buffer fluid (water) into the cycle fluid (helium). There was no evidence of back diffusion under any of the test conditions which were:

System temperature	-	80°F to 500°F
Steam pressure	-	25 psi to 250 psi
Δp across labyrinth	-	0.1 psi to 10 psi
System gas	-	air or helium

S-013
Sandstrom, C.O.
Gaskets For Pressure Vessels And Heat Exchangers
Chem. & Met. Engng., V.41, pp. 130-134, March, 1934

Flange-type sealing using both metal and non-metal materials; gasketless joints; crushing resistance of gaskets; special type gaskets such as wire, corrugated, and ring; temperature differences and effects on flanges.

S-014
Sass, F.
A New Stern Tube Packing for Seagoing Ships and River Boats
Konstruktion, V.2, No.11, pp. 321-325, 1950
Spec. Lib. Assn. Trans. 58-713 (18 pp.), V. 4, No.5, May, 1958

S-015
Saunders, R.D.
Quality Control in the Use of O-Rings
Approach, V. 7, pp. 34-38, 40, 41, June, 1962

Discussion of the use of O-ring seals, covering: (1) the sealing principle; (2) the selection of compounds; (3) causes of failure; (4) rules for proper installation; (5) O-ring gaskets; and (6) the evaluation of leakage.

S-016
Saunier, W.P.
Keeping Steel Gaskets Tight At 1350 PSI
Power, V.72, pp. 22-23, July 1, 1930

The use of copper-plated gaskets of dead-soft steel on high-pressure steam lines; specifications and sizes used.

S-017 B
Sawyer, D.W. and Paull, D.A.
Teflon Insert Compression Rings in Radial Gas Engines
A.S.L.E. Paper 60 AM 4C-1 (9 pp., 4 fig.), April, 1960

This paper gives the results of tests to determine the possibilities of Teflon inserts in piston rings which were performed at a plant operating approximately 250 gas burning Nordberg engines. Results sufficiently important to warrant further work in this field.

S-018
Sawyer, J.W. and Crawford, L.
A Mercury Shaft Seal
Am.Soc.Nav.Engrs.Jour., V.62, pp.349-363, May, 1950

This paper discusses a seal which opposes fluid flow by means of an internal pressure field. This is accomplished through the use of a centrifugal mercury seal supplemented by a positive contact seal. This paper describes attendant problems and a test model.

S-019
Saxon, A.F.
Multistage Sealing
Machine Design, V.25, pp. 170-172, March, 1953

Sealing a rotating shaft against pressures to 5000 psi at temperatures to 800°F demands more than ordinary means of sealing. How multistage sealing can be effectively applied under such service conditions is demonstrated by an autoclave design. Experimental models of the stuffing box have been successfully operated at pressures of 10,000 psi and higher.

S-020
Scepaniak, A.
New Seal Reduces Shaft Wear
Plant Engineering, V.13, pp.98-99, Nov., 1959

S-021 T
Schaffer, R.
Mechanical Seals For Centrifugal Pumps In The Chemical Industry (In German)
Chemie-Ing-Techn, V. 29, No.4, pp.241-249, 1957

This paper presents a basic practical study of mechanical seals for centrifugal pumps used in the chemical industry. Attention is given to materials of construction for corrosive media, and permissible temperature and pressure limits are discussed. A limited number of experimental observations are presented.

S-022
Schaphorst, W.F.
Blowproof Gasket
Textile World, V.69, p. 3047, May 1, 1926

S-023
Schmidt, E.G.
Flanged Joints, Their Development and Trend
Proceedings of National Conference on Industrial Hydraulics, V.II, pp. 33-42, 1948

Present briefly a picture of the development of flanged joints, the factors influencing their development, and work done to create joints which are simple, positive action, and easier to use. Test results, and curves included.

S-024
Schmidt, J.D.
Turbine Sealing Glands
Power Plant Engrg., V.46, pp. 86-87, Feb., 1942

A seal is necessary to prevent leakage along the shaft from the high-pressure end and the leakage of air inward on the vacuum end.
Stuffing boxes are impractical.
Types used are labyrinth, the carbon ring and the water seal, used alone or in combination. These are discussed pro and con.

S-025
Schmitz, C.E.
Mechanical Seals
Lubrication Eng., V.2, pp. 162-5, Dec., 1946

Problems encountered in packing or sealing rotating shafts; factors involved; influence of operating conditions such as temperature, pressure, nature of material being sealed, space requirements, and lubrication of rubbing surfaces of seal, in design of mechanical seals; construction requirements, application example in mechanical field are cited.

S-026
Schmitz, C.E.
A Look Inside the Mechanical Seal and Design Consideration
Lubrication Engng., V.4, pp. 14-17, Feb., 1948

Considerations in the choice and design of shaft seals: 1. Flexibility, 2. Surface finish of mating parts, 3. Laboratory testing, 4. Metallurgy, 5. Engineering. These plus application problems are discussed.

S-027
Schmitz, C.E.
Mechanical Seal: Its Construction, Application, and Utility
Paper Trade J., V.127, pp.21-5, Dec. 16, 1948

Outline of important design features; factors of speed; pressure, temperature, and viscosity; advantages; when special construction is desirable; utility of mechanical seal in refrigeration compressors, pumps, engines, etc.

S-028
Schmitz, C.E.
Hydraulic Packing and Seals
Applied Hydraulics, V.2, pp. 8-10, Feb., 1949, pp. 14-16, April, 1949

Survey of compression and self-sealing types of packings, including new materials, available for hydraulic applications; important factors concerning packing problems are discussed; angularity; number of rings; spring loading; following construction factors of typical mechanical seals, positive driving means, metallurgy, and engineering, laboratory testing and research application problems.

S-029
Schmitz, C.E.
The Mechanical Seal: Its Construction, Application And Utility
ASME Trans., V.71, pp. 635-641, Aug., 1949

Paper outlines important construction features of the mechanical seal; discusses speed, pressure, temperature, and viscosity; application considerations, advantages, and when special construction is desirable; utility of the mechanical seal; and suggested services.

S-030
Schmohl, L.H.
Sealing and Holding Separate in New O-Ring Standard
SAE Journal, V.64, pp.92-93, April, 1956

Sealing is accomplished by a synthetic rubber ring being compressed in a cavity between the fittings and the boss in the pumps, valves, or other components. Recommended usage discussed. Temp. 65°F to 212°F. New standard developed. Drawings included.

S-031
Schneckenberg, E.
Flow Through Narrow Concentric and Eccentric Circular Slits
Zeits f. Tech. Physik, V.11, No.9, pp.354-7, 1930

The author gives an insight into the present-day demands with respect to throttle packings for pumps and describes a practical process for making very narrow circular slits of width which can be accurately ascertained. Flow experiments are described for various types of packings with diagrams and illustrations.

S-032 T
Schneckenberg, E.
Flow of Water Through Concentric and Eccentric Cylinder Packing Glands With and Without Annular Grooves (In German)
Zeitschrift fur Angewandte Mathematic und Mechanik, V. 11, pp. 27-40, 1931

This paper is an extremely detailed report of experiments performed by the author. Calculations are based on equations derived by Becker. The experimental flow rates, which are presented graphically, provide a measure of the effectiveness of cutting annular grooves. Effects of eccentricity are included.

S-033
Schrader, E.W.
Face Type Seals For Rotating Shafts
Design News, V.15, pp. 6-7, Aug. 15, 1960

Face-type seals previously used almost exclusively on rotary pumps and primarily on centrifugal pumps. Recently, the seal has been used for special application with high speeds and pressures. Increased use of metal bellows has expanded their use to exotic fluids and much wider temperature ranges. Standard and stock seals available for temperatures from -450°F to 1000°F. Pressure limit 500 psi.

S-034
Schrader, E.W.
O-Ring Squeeze For Dynamic Sealing
Design News, V. 16, pp. 4-6 . Sept. 1, 1961

Factors affecting O-ring seal for reciprocating motion. Data curves presented.

S-035
Schuster, E.C. and Rupe, V.L.
Save Money With O-Rings
Petroleum Refiner, V.34, No.11, pp. 201-204, Nov., 1955

Testing synthetic rubber O-ring flanges; it is concluded that testing 300 ASA flange with O-ring can give cold service pressure ratings equal to 600 ASA ring joint for sizes up to 12 inches.

S-036 T
Schwaigerer, S., Seuffert, W.
Investigations on the Sealing Capacity of Sealing Strips (In German)
Brennstoff-Waerme-Kraft, V.3, No.5, pp. 144-148, May, 1951

This paper presents the results of sealing tests of flat and profiled sealing strips which were investigated with various pressurized fluids and various surface finishes. On test bases, relations can be formulated for the calculation of the critical sealing pressure which is required for alignment of the sealing surfaces in addition to that sealing pressure which is required during operation for a permanent seal.
(Included here since the gland of a mechanical face seal or buffered housing must be sealed statically with the housing of the compressor or turbine.)

S-037
Schweickhart, O.
Balanced Packing Rings (In German)
Waerme, V. 60, pp. 19-22, Jan. 9, 1937

Discussion of metal packings for use between parts in relative motion (piston rings); constructional features and pressure conditions of a number of different designs of partially and fully balanced rings for single and double-acting pressure.

S-038
Schweiger, F.A.
Dynamic Seals
DF59FPD659C - Sm. Av. Eng. Dept., G.E. Co., Lynn, Mass.

Discussion of dynamic seals including: types and selection, circumferential seal, segmented carbon ring seals, controlled gap seal, labyrinth seals; their advantages and limitations.

S-039
Scott, P.A.
PRTR Mechanical Seal Pump Operating Experience.
Sept. 1958-Aug. 1960
HW-65724 Rev., GEL Tech.Data Center, G.E. Co., Schdy, N.Y.(68 pp.), Sept. 16, 1960

The plutonium recycle test reactor (PRTR) is the first large high temperature-high pressure reactor for which mechanical seals will be used in the primary pumps. The faces run on a lubricating film of the liquid being pumped. In this case, the liquid, heavy water, is recovered by equipping the pump with two mechanical seals in series on the shaft in such a way that leakage is returned to the system.
Report relates experience gained from operation of the spare PRTR Primary Process Pump, and of a small pump with prototype mechanical seals and shaft assembly. Operated at temperature and pressure conditions prototypical of the PRTR.

S-040 T
Secord, J.R., Chapin, E.A.
Rotary Shaft Seal
U.S. Patent 2,916,314, Dec. 8, 1959

A seal designed to withstand translatory gyrations and axial vibrations which occur at the rear main bearing of an automobile engine crankshaft. Sealing action is by a piston type ring, either split or solid, which is maintained in contact with a shoulder on the rotating member by a corrugated annular spring ring.

S-041
Seki, H.
Simple Demountable Indium O-Ring Seal Tight To He II
Rev. Sci. Instr., V.30, pp. 943-4, Oct., 1959

An indium O-ring was used to seal an isolated chamber to He II at 10^{-6} mm Hg, and no appreciable change in pressure was detected as the λ point was passed.

S-042
Seufert, W.
Investigations of Packing Efficiency of Joint Surfaces (In German)
Brennstoff-Waerme-Kraft, V.3, pp. 144-8, May, 1951

Tests on different types; results show that no packing is absolutely impervious to gases and not entirely impervious even to liquids; complete impermeability can be achieved only when packing surfaces are plastically deformed.

S-043
Sharapov, G.A.
New Type of Gland Packing
KhimicheskayaProm, V.8, pp. 18-19, 1944

Glass-fiber waste was twisted into a cord and successfully used as a gland packing for pumps handling corrosive liquids.

S-044
Sharapov, G.A.
Gland Packing
S. Khimicheskaya Prom, V.2, p. 21, 1945

The packing withstood 65-75% H_2SO_4 containing 0.01-0.2% of HNO_3 at 35-170°C.

S-045 B
Shatford, P.A.
O-Ring Sealed Vacuum Valves
Journal of Scientific Instruments, V.29, No.10, pp. 336-7, Oct., 1952

Half-page description and sketches of a two-way, and an isolation valve.

S-046
Shaw, C.J.
Laboratory Test and Evaluation of Seals Used in the Propellant Line Systems
Douglas Aircraft (3227), Mar. 30, 1960

Determine sealing capacities of gaskets used with liquid oxygen and fluid systems.

S-047
Shaw, G.V.
Mechanical Shaft Seal for Centrifugal Pumps
Oil & Gas J., V.42, p. 251, April 13, 1944

Light liquids, high pressures and speeds have necessitated development of more reliable and economical means of sealing rotating shaft than can be obtained when using conventional stuffing box and packing; with drawing showing essential design details.

S-048 T
Shaw, M.C., Nussdorfer, T.J.N.
An Analysis of the Floating Journal Bearing
NACA Report No. 866 (13 pp.), 1947

An analysis of the operating characteristics of a full-floating journal in which a floating sleeve is located between the journal and bearing surface. The whirl of the floating bushing is briefly considered.

S-049 T
Shenton, F.
Outside Type Seal
U.S. Patent 2,252,526, Aug. 12, 1941

A shaft seal designed for use in a refrigeration compressor or similar device where the sealed pressure may fluctuate above and below atmospheric pressure. The seal is designed for use external to the compressor. The sealing mechanism centers around two "C"-shaped rings placed back-to-back in opposite directions which are forced into contact with the shaft by a pressure unbalance in either direction.

S-050 T

Shenton, F.
Shaft Seal
U.S. Patent 2,323,730, July 6, 1943

A shaft seal assembly such as might be used on a compressor crankshaft where pressure would fluctuate from positive to negative. A pressure differential across two flexible "C"-shaped annular seal rings, between a sliding collar and rotating fixed collar, aids springs in holding wear ring against the seal nose. An oil circulation system for oil retention is also included.

S-051

Shepherd, H.F.
Seals Perfect Lubrication System
Machine Design, V. 5, p. 23, Nov., 1933

Advantages of using unit oil seals embodying specially treated "hot leathers" in pressed steel case; if shaft is not required to penetrate housing wall, single leather seal confines oil admitted through feed tube, forcing it to take path through shaft bore, whence it may be directed as needed.

S-052

Shepler, P.R. and Noren, O.
Split-Ring Seals
Mechanical Design, THE SEALS BOOK, pp. 24-31, Jan. 19, 1961

Split rings are used for a large number of seal applications. Expanding split rings (piston rings) are used in compressors, pumps, and internal combustion engines.
Use of contracting split rings or rod seals in linear actuators is increasing where high pressure, high temperatures, radiation, thermal fatigue, and need for reliability prohibit the use of more conventional seals or make them less desirable.
A table, "Key Principles of Sealing," is included as is a table, "Corrosion-Resistant Material," covering medium-temperature seal material and seal application.

S-053 B

Sibley, L.B. and Nace, A.E.
Characteristics Governing the Friction and Wear Behavior of Refractory Materials for High Temperature Seals and Bearings
Battelle Memorial Inst., WADD Tech.Report 60-54 (54 pp.) (PB 171010), May, 1960

Various ceramic and cermet materials have been evaluated for unlubricated wear resistance at high sliding speed (100-200 fps) and low unit load (5 and 50 psi) in 100 and 1800°F air. A statistical correlation was obtained between the measured wear rates under these conditions and the coefficient of friction, thermal stress-resistance, and the thermal diffusivity of the mated materials on which wear predominated.

S-054

Siebel, E. and others
Tests on the Performance of Gaskets (In German)
Forsch. Gebiete de Ingen., V.5, pp. 298-305, Nov.-Dec., 1934

The purpose of the above investigation was to obtain numerical data concerning the forces necessary for various shapes of packing and materials in order to obtain a tight joint at given pressures. Data of this kind is indispensable if useful methods of calculation are to be obtained for flange connection. Tests were carried out on the behavior of packing materials at room temperature with water as a pressure medium as well as at 350°F with steam.

S-055 T

Sikorski, J., Woods, H.J.
Rotary Vacuum Seal
Jour. of Scientific Instruments, V. 30, p. 43, Nov., 1953

A mechanical face seal was tested for its performance under vacuum. A pressure of 2×10^{-4} mm Hg was sustained for 220 hours in a 1.5 liter container with 1" shaft rotating at 600 - 900 rpm. Vacuum was obtained by a Metropolitan-Vickers 03B oil diffusion pump in series with a two-stage rotary pump. Apiezon oil "B" was used for the seal. Improvement could be accomplished by improving the seal.

S-056

Sinclair, J.B.
How and Why Of Seals And Their Installation
Paper Trade Journal, 141:40, July 8, 1957
Coal Age, 62:98+, Sept., 1957
Brick and Clay Rec., 131:67, Oct., 1957

S-057

Sinclair, J.B.
Correctly Installed Seals Prevent Damage and Down Time
Petroleum Engr., V.29, No.13, pp. D23-24, Dec., 1957

Installation of leather and rubber seals to protect vital parts of earth moving machinery; installation of lip seals.

S-058 T

Sitney, L.R.
High Speed Rotary Vacuum Seal
Review of Scientific Instruments, V.23, pp. 505-6, 1952

This seal has operated satisfactorily for speeds up to 18,000 rpm at pressures to 8×10^{-6} mm Hg. It is believed suitable down to 1×10^{-6} mm Hg. Two Garlock Klozures with a guard vacuum between them are employed in the seal.
(Garlock Klozures are rubber lip seals.)

S-059

Skarstrom, C.W.
Sealing Means For Relatively Rotatable Members
U.S. Patent 2,957,709

A sealing means is offered for maintaining a seal between a pair of relatively rotatable members, particularly between a rotating shaft and a stationary member surrounding the shaft.

S-060

Skarstrom, C.W.
Seal for High Speed Centrifuge
U.S. Patent 2,816,704, to US A.E.C.

A seal is described for a high-speed cen... wherein the centrifugal force of rotation acts on the gasket to form a tight seal.

S-061

Skinner, E.T.
Graphite-Impregnated Rubber Packing for Reciprocating Pumps And Oil-Well Stuffing Boxes
ASME Preprint No. 51 (4 pp.), PET-17, 1951

Paper explores the use of graphite-impregnated rubber as a possible material for making mechanical rubber goods subjected to abrasive wear, e.g., gland packings for slush pumps.

S-062

Skow, N.A.
Plastic Laminates Bonded To Other Materials
Matls. & Methods, V. 45, No. 4, pp. 118-121, April, 1957

Design advantages offered by thermosetting laminates when combined with other materials such as steel, aluminum or rubber; material and forms available; use of combination laminates as structural materials for plating rolls, aircraft shock strut piston heads and precision bearing retainers, as electric materials for printed circuits, etc., and as sealing materials.

S-063 T

Smaardyk, A.
Summary of Shaft Seal Tests for High-Temperature, High-Pressure Water Application
Argonne Nat'l. Lab. ANL-5220 (33 pp.), April, 1954

Summary of program directed toward the development of an all-metallic seal assembly for use in connection with the design of external control rod drive activating mechanisms for pressurized, water-cooled reactor systems. Temperatures up to 450°F, pressures up to 2000 psig.
Specifications, drawings and test data included.

S-064

Smith, A.M.
Radioisotopes as Design Tools: Gasket Leakage
Product Engineering, V.26, p. 134, May, 1955

A technique to find the origin of leaks and determine whether design changes were advisable. Use of a soluble radio isotope in the cooling water offered promise on the assumption that activity would be deposited at any point where liquid evaporated to dryness. After running a test, the gasket could be removed and placed in contact with X-Ray film to show the leakage pattern. Test procedure and results presented.

S-065

Smith, J.N.
How To Apply V-Leather Packings
Product Engng., V. 7, pp. 141-142, April, 1936

Advantages of V-leather, the factors that determine the thickness, proportions and number to use and how to support them in the stuffing box; with tables and illustrations.

S-066

Smith, J.N.
Design and Materials For Hydraulic Packings
Aero Digest, V.40, pp. 119-120, 122, April, 1942

Hydraulically-operated units in the aircraft industry subjected to extreme service conditions. Packing must withstand all of following conditions. Pressure 0-3000 psi, temperature -40°F to 158°F. Material problems attendant to these operating norms are discussed. Leather is cited as most nearly perfect material available. Its advantages and limitations are discussed as are certain design problems.

S-067

Smith, J.N.
Hydraulic Packings For Industrial Equipment
Mach. Design, V.21, pp. 155-160, Dec., 1949

Standardization of hydraulic packings by sizes and types to simplify selection and reduce sizes and inventories worked out. Article presents information towards this end.

S-068

Smith, J.N.
Theory and Design of Large Diameter Packings
Proceedings National Conference on Industrial Hydraulics, Vol. V, pp. 121-130, 1951

Discusses packings up to 100" dia., up to 140,000 psi. Machining tolerances, surface finish, eccentric loading. Outside vs. inside packings, and their respective advantages.

S-069 B

Smith, J.N.
Adaptors for "V" Ring Packings
Machine Design, V. 32, 7, pp. 138-40, March 31, 1960

Metal, leather, hard homogeneous rubber, and phenolic adaptors are compared, and test results and data presented. Qualitative adaptor comparisons and recommendations are made.

S-070

Smith, J.N.
Molded Packings
Mechanical Design, THE SEALS BOOK, pp.58-67, Jan. 19, 1961

Molded packings are often called automatic, hydraulic, or mechanical packings. As a group, these packings usually do not require any gland adjustment after installation. The fluid being sealed supplies the pressure needed to produce the force for seating the packings against the wearing surface.
Two categories are lip and squeeze types. The lip-type includes flange, cup, "U" cup, and "V" ring packings.
The squeeze-type includes the O-ring and other similar forms which rely on the interference built into the ring for effective sealing.
Leather, rubber and Teflon are used.
Leather, -65°F to +200°F.
Rubber, -65°F to +500°F.
Teflon, upper limit +500°F- suffers from lack of resilience.

S-071

Smith, J.N.
Compatibility of Seals With Modern Hydraulic Fluids
Lubrication Engrg., V.18, pp. 320-323, July, 1962

Operating compatibility (not chemical) of dynamic seals with hydraulic fluids. Importance of knowing a mineral oil's aniline point and the anticipated operating temperatures of the fluid before selecting the packing material. Tabulates properties of synthetic rubber packing materials.

S-072
Smith, J.R.W.
Demountable Vacuum Seals
J. Sci. Instrum., V.29, p. 131, April, 1952

Three types of seals are described. One is a Wilson-type seal suitable for use with a glass end plate on a vacuum system. An O-ring rests on the highest part of the flange. On the inside, the O-ring is supported by a metal ring. Up to eight seals of the latter kind have been used with success in vacuum systems operating at pressures of 3 x 10^{-6} mm Hg.

S-073
Smith, L.H., Jr.
T-58 Guide Vane Pneumatic Actuator Seal
Tech. Proposal, Gen. Eng. Lab., G.E. Co., Schehectady, April 15, 1960

Recommendations are made for a low friction piston seal to be incorporated in a small pneumatic actuator. The program includes design and fabrication of a suitable seal; procurement of a commercial seal; design and fabrication of test equipment, and test evaluation of the seals. C arbon piston rings and a spring-loaded "U" type seal are considered.

S-074
Smith, L.L.
Composite Inorganic Resilient Seal Materials
WADC Tech. Report 59-838, Part III, May, 1959

The principal objectives of this research program is to investigate and develop composite materials suitable for use as static and dynamic seals at temperatures ranging from cryogenics to 1200°F, and pressures up to 5000 psi.
1. Fibrous composite with pure metal impregnants.
2. Fibrous composites with other impregnants.
3. Non-fibrous composites (talc or glass).
Studies of resilience, fiber orientation, friction and wear, static seals, reciprocating and rotating shaft seals.

S-075 T
Smith, M.I., Fuller, D.D.
Journal-Bearing Operation at Superlaminar Speeds
Trans. Amer. Soc. Mech Engrs., V. 78, pp. 469-474, April, 1956

Results of investigation of load-carrying capacity and frictional characteristics of a journal bearing which was operated at speeds up to five times the critical (laminar breakdown of film) speed. The frictional behavior of the unloaded bearing was such that regions of laminar, transition, and turbulent flow could be defined.

S-076
Smith, W.R.
A Quick-Change Seal Opener
Rev. Sci. Instrum., V.24, pp.886-887, Sept., 1953

In case of the ORNL 86-inch cyclotron, the target assembly is water cooled and designed to be rotated or displaced in a linear direction. The stem of the assembly terminates in a water header which is fitted with three Wilson-type seals made from koroseal gaskets. The seal opener described in this note is designed to simplify insertion of the stem. It consists of a nut inserted into the seal body in a position concentric with the cylindrical bore accommodating the target stem. The new arrangement reduced the time required for changing the target assemblies to 5 minutes, equal to 1/5 the total time required previously.

S-077
Smoley, E.M.
How To Check Compression Characteristics Of Gasketed Joints
Mach. Design, V.28, pp. 76-78, May 17, 1956

This article describes a quantitative method of test called the "solder plug test," that can be used for approximating gasket compressions in an application. Also shows how the test is used for evaluating flange bending or bowing. Also points out some general procedures that can be used for analyzing leak problems and salvaging faulty flange designs.

S-078
Smoley, E.M.
Flange Pressure In Gasketed Joints
Machine Design, V.30, pp.133-137, June 12, 1958

Flange pressure is a critical factor in gasketed joints. Little information is available to specify the minimum sealing load for a given gasket material or to relate this load to flange-bolt torque, at best an uncertain criteria.
Procedures for handling such problems are discussed.

S-079
Smoley, E.M.
How Gaskets Behave
Prod. Engng., V.30, pp.40-42, May 25, 1959

Sparked by more aluminum in auto engines, gasket design techniques are being overhauled to fight high-temperature loosening. Presented is the latest research about the way typical gasket materials behave in aluminum-steel joints cycled to 300°F.

S-080
Smoley, E.M.
Nonmetallic Gaskets
Mechanical Design, THE SEALS BOOK, pp.83-89, Jan. 19, 1961

Joint and gasket design must be considered together. A joint is only as good as its gasket and the gasket may succeed or fail according to whether or not the joint is designed and constructed to make the best use of the properties of the gasket material. Therefore, the joint components must be thought of as a unit or as a system for effecting a seal.
The mechanical factors of joint design for nonmetallic gaskets are defined, requirements are set up, and an empirical solution to the problem is given. The various nonmetallic gasket materials are discussed, and recommendations are provided for their proper selection.

S-081
Smoley, E.M. and Frazier, E.C.
Nonmetallic Gasket Materials And Forms
Mechanical Design, THE SEALS BOOK, pp.90-99, Jan. 19, 1961

Material must meet following requirements: 1. Impermeability, 2. Ability to flow into joint imperfections when compressed, 3. Maintain seal in spite of age, temperature, and pressure variations, 4. Resist environmental deterioration.
Gasket materials, tables, form, seating stresses.

S-082
Smoley, E.M.
Retaining Tension in Gasketed Joints, Part I
Assembly and Fastener Engineering, pp.30-34, Sept., 1961

Describes some reasons why bolt tension is reduced in ball flanged joints. Shows effect of gasket materials, thickness, bolt length, flange materials, etc.

S-083 T
Snell, L.N.
Theory of Viscosity Thrust Bearing Based On Circular Geometry
United Kingdom Atomic Energy Authority, IPDC (14 pp.), p. 137, 1959

The Whipple Theory of the viscosity plate thrust bearing is based on the assumption that the radius of the plate is large compared with the width of the pumping portion. The paper contains a modified theory which takes full account of the circular nature of the plates. It is shown that such a modification will not reduce the discrepancy between theory and experiment.

S-084
Sniffen, T.J.
How To Select And Install Pump Seals
Petroleum Refiner, V.35, No.3, pp.207-210, March, 1956.
Pipe Line Industry, V.5, No.1, pp. 34, 39-40, 42, July, 1956

Principles of mechanical seal performance, maintenance and method for preventing its failure; seal selection, material, and installation precautions.

S-085 T
Sniffen, T.J.
Mechanical Seals From A to Z
Power and Fluids, pp. 20-23, Winter, 1958
"Basic Principles of Mechanical Seals," Product Engineering, V.29, pp.12-14, Mid-Sept., 1958

This article describes mechanical seals with respect to basic seal elements, operation, selection, installation, and materials.

S-086
Soderholm, L.G.
Fiber-Metal Seals Allow High Temperature-Pressure Operation
Design News, V.16, 90-91, June 19, 1961

Fabrication of high-temperature, high-pressure static seals using sintered and brazed skeletons of Mo and stainless steel fibers impregnated with melted I_n and A_g by compression under vacuum and in an Argon atmosphere. Data are given for compression and recovery measurements, thermal cycling and hydraulic leak tests at various temperatures to 1000°F and pressures to 5000 psi at varying impulse cycles.

S-087
Soderholm, L.G.
Integral Cross Seals Tighten Half Seal Sections
Design News, V.16, No.21, pp.28-29, October 13, 1961

A new one-piece half-seal has molded projections extended radially from the end of the semi-circular half seal section. These projections fit into a groove in either the lower or upper support members.

The cross-seal projections do not fill the groove entirely when they are placed in position, but project slightly above the face of the receiving support member. When the top surface is placed in position and assembled, the cross seal projections are compressed and flow downward and outward into their grooved slots.

S-088
Somerling, H.
Contribution to Calculation of Labyrinth Packings
Revue M de la Mecanique, V.3, pp. 47-59, April, 1957

Critical examination of existing calculating methods; causes for inaccuracy of Stodola formula; new more exact method developed is based on theoretical considerations and on laboratory results obtained on experimental installation.

S-089
Sondermann, H.
Design of Packings and Gaskets For Hydraulic Pressures (In German)
Maschinen Konstrukteur, V.64, pp.231-4, Dec.10, 1931; V.65, pp. 8-9, Jan. 10, 1932

Dec.10: Design of packings and gaskets for high hydraulic pressure.
Jan. 10: Dimensions of packing of convex or concave type suitable particularly for double-acting pistons.

S-090
Sondermann, H.
High-Pressure Hydraulic Gland Packing
Engrg., V.136, pp.325-6, Sept. 22, 1933

A study of the design of high-pressure glands used in hydraulics. The design and use of "U" and "V" leathers in hydraulic rams.

S-091
Spain, R.G.
Compounds For High Temperature Fuel Seals And Their Testing Under Simulated Use Conditions
Included in WADC Tech. Report 57-478, Elastomeric for Air Weapons; WADC, Univ. of Dayton Conference, Oct. 1957

S-092
Spain, R.G., Lajiness, W.G. and Deck, E.
Research and Development On High Temperature Fuel Resistant Rubber Compounds
Wyandotte Chemicals Corp., Wyandotte, Mich., WADC-TR-55-492 (Rt.4); Contract AF33(616)-5544, Project Nos. 7340 and 3048 (90 pp.), Feb.18, 1959

The development of elastomeric compounds resistant to high energy fuels (HEF) at elevated temperatures is described. Description of test methods used.

S-093 T
Spaulding, D.C., Jr.
Non-Rubbing and Rubbing Seals For Oil Retention
Product Engineering, V.26, pp.42-5, Mid-October, 1955

These two articles present a pictorial description of methods of retaining lubricant in bearings. Included are various types of labyrinths, lip seals (felt and rubber), and mechanical face seals.
Applications are given.

S-094
Spees, A.H. and others
Vacuum Gaskets at Low Temperatures
Review of Scientific Instruments, V.28, p. 1090, Dec., 1957

Soft aluminum wire has been used successfully as a gasket material for vacuum systems. It is of particular importance in low-temperature work and has proved to form a satisfactory seal in liquid helium I and II.

S-095
Spence, E.L.
Increasing Packing Life
Chem. Engng., V.59, pp. 200-201, Feb., 1952

A little extra care in storing, installing, and lubricating packings can greatly increase their life. Suggestions for accomplishing this are given.

S-096
Spitsyn, N.A.
Basic Designs of Seals For High Speed Roller Bearings (In Russian)
Vestnik Mashinostroeniya, V.39, pp.3-8, Sept., 1959

Survey of various types of seals to avoid passage through journal into vacuum chamber of air, gas, oil or other materials; selection of insulation is made in relation to rotation velocity.

S-097
Sprigg, E.A.
Bonded Gaskets Solve Sealing Problems in Die Cast Assemblies
Die Casting, V.4, pp. 70-72, March, 1946

Use of molded synthetic rubber gaskets provides means of eliminating extra handling operations in assemblies where a gasket member is required for sealing.

S-098
Spruck, R.F. and others
Use Metallizing Tape For High Quality Ceramic Metal Seals
Cer. Ind., V.79, pp. 88-91, Sept., 1962

S-099
Squire, J.W.
Uprate Flanges With An O-Ring Seal
Oil and Gas Journal, V.56, pp.111-113, Oct. 20, 1958

Advantages of using an O-ring seal with flat-face flanges for the ASA 300 welding-neck series up to the 24-in. size are pointed out in this report.
Discussion of flange design, bolt stresses, and practical considerations.

S-100
Stahl, E.P.
Ball and Roller Bearing Seals
Product Engineering, V.18, pp.103-6, April, 1947

Designs of common types of oil and grease seals for excluding foreign material from ball and roller bearings are shown. Proper application of different types of seals are discussed. Typical seal arrangements for industrial and automotive equipment are described.

S-101
Stahl, E.P.
Selection of Seals of Various Duties
Machinery, V.81, pp. 367-73, 1952

Discussion of types of seals:
1) Simple felt seals act as reservoirs
2) Metal labyrinth
3) Unit type - spring loaded leather seals
4) Unit type - synthetic rubber seals
5) Unit type - with double leather or synthetic rubber sealing members
6) Springless unit type seals
7) Mechanical seals

S-102
Stahl, E.P.
Sealing Devices For Sleeve Bearings
Electrical Manufacturing, V.53, pp.133-6, June, 1954

Review of available types of seals and their proper selection for lubricant retention and dirt exclusion.

S-103
Stahl, E.P.
Correct Practice For Oil And Grease Seals
Power Engineering, V.61, pp.70-72, Dec., 1957

To do a good job, any ball and roller bearing seal must: 1) retain lubricant in the bearing, 2) prevent dirt from entering. Selection of the right seal is all important, as is knowledge of slingers, shrouds, and other supplementary parts designed to help primary bearings do their job even better. Synthetic rubber seals. Combination seals. Large unit on lip-type seal. Installation and removal techniques.

S-104 T
Stair, W.K.
Liquid-Buffered Bushing Shaft Seals For Large Gas Circulators
Int.Conf.on Fluid Sealing, Paper C5 (14 pp.), April 17-19, 1961,
British Hydromechanics Research Assn., Harlow, Essex, England

The principles of hydrodynamic lubrication are employed in the analysis of liquid-buffered bushing shaft seals. The study includes the prediction of buffer fluid leakage, operating eccentricity ratio, and pressure drop across the seal necessary for satisfactory seal operation. It is shown that seal pressure drop cannot be established by an arbitrary specification of specific leakage rate.

S-105
Stair, W.K.
Bibliography on Dynamic Shaft Seals
Univ. of Tennessee, Engn. Exp. Sta. Dept. of Mech. Engng., 353 references (97 pp.), May, 1962 (Preliminary Issue)
References published through Dec., 1961.

S-106
Staller, J.J.
Oil Buffer Packing Gland - A153162
PB 36915, Off.Tech.Serv.(3 pp.), Jan., 1944
(Springfield Armory Eng.Dept. Tech.Rept. W0377)

Tests were made on various samples of packing glands for Cal. 50, M2 aircraft machine guns, using lengthened and standard dimension packings of Super Seal #7 material, and lengthened samples of Super Seal #3 material.

S-107
Staples, B.G.
Flexible Corrosion Resistant Plastic Solves Gasket Problems
Materials & Methods, V.32, pp.48-49, Dec., 1950

Teflon rings and tape prove good gasket and gasket protecting materials for use in chemical processing equipment.
Plastic is one of the most inert to chemical attack and can be used at temperatures up to 500°F, without decomposition, and at pressures to 300 psi.

S-108
Staples, B.G.
Gasketing Glass-Steel Equipment
Corrosion Technology, V.6, pp.151-152, May, 1959

Research conducted by Pfaulder Co. to develop suitable gasket designs that utilize materials that combine chemical resistance of glass with physical properties of rubber; recommendations for proper installation of Teflon-shielded gaskets; other methods of gasketing.

S-109
Starr, A.M.
Seal
U.S. Patent No. 2,442,922, June 1, 1948, to U.S. Atomic Energy Commission

Details and design of a rod extending into a vacuum chamber through a seal which, although leakproof, permits rotational, translational, and rocking motion of the rod.

S-110
Steckelmacher, W.
Seals and Gaskets for Ultra High Vacuum Systems
Vacuum, V.12, March/April, 1962

Seals which remain vacuum tight while being baked at high temperature to promote rapid degassing. Demountable joints have been made employing thin wires, solid rings of various shapes, and metal foil gaskets. The design of suitable flanges, shaped projections, and any associated springs are considered. Data are given of experimental results achieved with various seals covering a range of materials and dimensions.

S-111
Stephens, H.R.
High Pressure Teflon Seals and Closures
Plastics Tech., V.3, No. 2, pp. 113-114, Feb., 1957

Self-sealing principle is used to design packing rings and closures that provide excellent high-pressure seals; Teflon packing ring has been used in very simple self-sealing closure designed to hold 22,500 psi of pressure in 1-inch diameter chemical reaction vessel.

S-112 T
Stephens, J.A.
Properties of Rubber Seal Materials
Int. Conf. on Fluid Sealing, Paper F1, April 17-19, 1961
British Hydromechanics Research Assn., Harlow, Essex, England

A survey of various types used in manufacture of lip seals, and pertinent properties discussed. Strength and wear resistance, elastic recovery, stability at temperature extremes, and resistance to fluids of eight rubbers are compared. An opinion as to the significance of these properties is given.

S-113
Sterzer, F.
Simple High-Temperature Vacuum-Tight Mica Window
Rev. Sci. Instr., V.28, pp.208-9, 1957

A vacuum-tight mica seal was obtained by screwing together a thin sheet of mica and a flat annealed copper ring between two stainless-steel flanges.

S-114
Stevens, A.F.
Split Lantern Rings in Pump Packing Glands Reduce Oil Losses, Increase Life of Packing
National Petroleum News, V.38, R254, p. 2, April 3, 1946

S-115
Stevens, J.B.
Current Status of Axial Face Seals
SAE Jour., V.64, pp.46-50, Sept., 1956

Application of seals to aircraft gas turbine main shaft bearings, gear boxes or accessory pads and accessories; limitations with respect to sealing media, temperatures, pressures and rubbing surface speeds; influence of these factors on selection of materials.

S-116
Stevens, J.B.
Bellows Type Mechanical Seal
Mechanical Design, THE SEALS BOOK, pp.44-47, Jan. 19, 1961

The main function of the metal bellows in axial mechanical seals is to replace the static seal element. They are useful in special applications where an all-metal seal is necessary. These applications are generally those where high temperature is encountered or where the liquid or gases used would adversely affect organic seal materials.
Metal bellows seals will operate from -400°F to + 1200°F. Sealing surfaces must be able to accept the thrust loads developed by the pressure drop across the seal. A maximum pressure drop of 1000 psi is reasonable.
Selection factors and their significance is covered.

S-117
Stevens, J.B., Greiner, H.F.
Factors In Designing for Mechanical Seal Applications to Hydraulic Equipment
Proc. National Conference on Industrial Hydraulics, V. VII, pp. 131-42, 1953

Thorough presentation of terms and classification; advantages and limitations of mechanical seals; balanced and overbalanced; materials for packing and their limitations, including Teflon, Kel-F, polyethylene.

S-118
Stevens, J.B. and Greiner, H.F.
Mechanical Seals for Hydraulic Equipment
App. Hydraulics, V.7, pp. 70-73, June, 1954

Design and application of eight types of mechanical seals for hydraulic equipment are covered. These are:
1. Jam packing rotary unbalanced seal.
2. "V" packing stationary balanced seal.
3. "V" packing stationary unbalanced seal.
4. O-ring rotary balanced seal.
5. Rubber bellows- rotary unbalanced seal.
6. Rubber diaphragm-stationary unbalanced seal.
7. Metal bellows-stationary balanced seal.
8. Piston ring-stationary balanced seal.

S-119
Stewart, W.L.
Designing and Testing Journal Box Seals
Lubrication, V.15, pp. 242-245, 264, June, 1959

Increased use of lubricating pads replacing cotton waste as lubricant medium on railway journal box showed notable increase in oil consumption; problems encountered in attempt to seal back end of journal box; factors to consider in design of railway sealing applications; details of equipment for testing seals designed to operate in standard 5-1/2 x 10 AAR journal box; standard test procedure consisting of six tests.

S-120 T
Stodla, A. (Translated by L.C.Loewenstein)
Steam and Gas Turbines
Sixth Edition, McGraw-Hill Book Co., Inc., New York, N.Y., V. 1, pp. 189-194, 1927

Review of Becker's analysis of flow through annular clearances, with graphical experimental results. For labyrinths, an equation similar to that of Martin is derived but in a different manner. In addition, modifications to the formula allow its use for radial labyrinths and critical flow. A graphical method which gives better accuracy for critical flow is provided. Experimental data are reported.

S-121
Storozhnik, A.A.
Hydraulic Seal of the Charge Distributor When Working at Elevated Top Pressure
British Iron and Steel Industry Translation Service
BISI-2238

Hydraulic seals, distributors, pressure.

S-122
Stott, T.C.
Some Notes on Synthetic Rubber Oil Seal Applications
Instn. Automobile Engrs., J., V.12, pp. 161-187, July-Sept., 1944

Automobile and kindred applications involving high surface speeds and very high standard of sealing performance. It is for these applications that the synthetic rubber seal is eminently suited. Notes deal principally with synthetic rubber seals running under such conditions, and are based on results of various investigations and tests.

S-123 T
Stratton, A.C.
Shaft Sealing Means
U.S. Patent 2,243,227, May 27, 1941

A mechanical face seal is actuated by means of a collar of rubber or neoprene with a "V"-shaped annular groove around its circumference. A garter spring in this groove tends to widen the rubber collar, forcing the rotating seal ring against the stationary seal face. The rotating seal ring is driven by friction with the rubber collar, which forms its own static seal with the shaft.

S-124
Straus, F.R.
Performance Cycling Test of High Temperature Hydraulic O-Ring Packings
PB-123720, Off. Tech. Serv., Project 1371, Task 13495, AF WADC TN WCLS 54-71

Description of methods used to determine the mechanical performance of synthetic O-ring packings in hydraulic fluids at temperatures up to +500°F.

S-125
Straus, F.R.
Evaluation and Testing of Hydraulic O-Rings
Included in WADC Tech.Rept. 57-478, Elast. for Air Weapons, WADC University of Dayton Joint Conference, Oct., 1957

S-126
Straus, F.R.
Development of O-Rings for Aircraft Application
Applied Hydraulics, V.6, pp. 82, 84, 86, 114-5, Aug., 1953

Design of early aircraft O-rings; military specifications issued in 1942; development of compounds for use in molding O-rings; functioning of O-ring compounds down to -40°F was proved in 1942-1943 cold weather operational tests; development of hydraulic packings in the last 10 years; present-day specifications.

S-127
Streck, F.O.
Performance Test of a Two-Coolant Retention Sodium Pumpshaft Freeze Seal
Atomics International Dev., North American Aviation, Inc., Canoga Park, Calif., NAA-SR-Memo. 4119 (15 pp.), July 15, 1959

The operation of the freeze-seal type sodium pump requires a shaft freeze-seal capable of retaining sodium. Test characteristics noted.

S-128 T
Strickland, H.R.
Bearing and Seal Assembly for Turbines
U.S. Patent 2,496,897, Feb. 7, 1950

Labyrinth seal and bearing assembly which may be installed and removed axially with the rotor in place. The radial labyrinths are of such design that longitudinal expansion of the rotor has no effect.

S-129
Strohm, R.T.
Metallic Packings
Am. Elect'n., Oct., 1905

States the advantages of metallic packings, and illustrates and describes several forms.

S-130
Stubenrauch, E.H.
Getting The Most Out Of Gaskets
Chemical Engineering, V.69, pp. 123-128, Sept. 3, 1962

Materials discussed: Teflon, Armalon, Viton, silicone, metal fiber composites, compressed asbestos, woven and folded asbestos cloth, rubber, vegetable fiber, and metals. Applications of various metal gaskets and six designs for metal gaskets are tabulated.
Flange design and surface finish for proper sealing of gaskets includes table of five basic types of gasketed joints with characteristics and suitable types for each.
Also discusses bolting procedures, seal maintenance, and changing of gaskets.

S-131 T
Summers, A.B.
Multi-Disk Floating Contact Mechanical Shaft Seal
U.S. Patent 2,711,914

This invention uses the sealed shaft to drive a number of gears of decreasing size. These driven gears in turn drive floating gears mounted on the shaft or shaft collar. Sealing disks between the driving shaft, and adjacent floating gears, and the housing effect the seal. By suitable ratios and varying the number of gears, any maximum speed between sealing elements may be held.

S-132 T
Summers-Smith, D.
Laboratory Investigation of the Performance of a Radial Face Seal
Int. Conf. on Fluid Sealing, Paper D1 (12 pages), April 17-19, 1961
British Hydromechanics Research Assn., Harlow, Essex, England

The performance of a radial face seal in terms of leakage, wear and friction torque in a rig in which the liquid pressure and contact loading of the seal face could be varied independently. Sealed water at pressures up to 50 psi. Friction and wear measurements indicate that for normal operation a fluid film (approx. 100 micro-inches thick) is generated between the sealing faces.

S-133
Sussman, S.
Mechanical Shaft Seals
Air Cond., Heat. & Vent., V.57, pp. 61-66, July, 1960

Factors governing use, performance, and failure of seals which incorporate two rings which have their faces at right angles to pump shaft; examination of their application to open spray and closed circulating systems.

S-134
Swartz, J.H.
Nonmetallic Diaphragms
Machine Design, V.20, pp. 153-8, 206, June, 1948

Chemical and physical properties of rubber and other diaphragm materials; design considerations; various applications such as pressure regulators, control valves, and accumulators.

S-135
Swartz, J.H.
"O" and "V" Ring Packings in Industrial Hydraulics
Proceedings of National Conference on Industrial Hydraulics, V.IV, pp. 251-269, 1950

Thorough coverage, including: historial background, the Rapleff joint, function of packings, "V" ring packing, fabric "V" ring packing, vee-dam packing, homogeneous "V" rings, O-ring packing, design data, dynamic O-rings, static O-rings, and material selection.

S-136
Swearingen, J.S.
Know Your Mechanical Seals
Chem. Eng., V.62, pp. 213-16, 1955

The factors which govern the choice between types of mechanical seals for rotating shafts, or between seals and stuffing boxes, are discussed.

S-137
Swartz, J.H.
Design Recommendations for O-Ring Seals
Product Eng., V. 22, pp. 162-3, Sept., 1951

Drawings and test concerning: type of groove, groove dimensions, external and internal grooves, poor and good practice, face seal groove, plug seal, check valve seal. Table of dimensional data for standard AN or JIC O-rings and gaskets.

S-138
Sweikert, M.A. and Johnson, R.L.
Wear of Carbon Type Seal Materials With Varied Graphite Content
Am. Soc. Lubrication Engrs. Trans., V. 1, No.1, pp. 115-120, April, 1958

Study to determine influence of increased graphitic carbon content on wear and friction of carbon bodies used in sliding contact seals of turbine type aircraft engines; bodies molded with high graphite content materials and hardened by improved methods gave acceptable friction and wear properties, using hardenable stainless steel; effect of varied hardness of mating surface on wear of typical carbon was slight.

S-139-
Swota, John, Jr.
Compression Load Cell for Measurement of Seal Compression Forces
Nuclear Science Abstract- 19011 (KAPL-M-JOS1)
Cont. W-31-109-Eng.52
(Knolls Atomic Power Lab., Schenectady) (8 pp.), Mar. 12, 1962

A load monitoring device was incorporated into a Conax seal test fixture, and a test procedure was developed to determine the relationship between applied torque and the resulting compression load impressed on Teflon seals as a function of various design parameters such as follower configuration and lubrication effects. Periodic calibration indicated that load values are accurate to within ± 0.5% of full load values.

S-140
Symons, J.D.
Engineering Facts About Lip Seals
SAE Automotive Eng.Congr., Detroit, Mich., preprint 473A (14 pp.), Jan. 8-12, 1962

Description of a test program for the study of oil seals. Causes of seal leakage are investigated, and some design criteria are established.

S-141
Symons, J.D.
Lip Seals, Performance and Design Factors
Machine Design, V.34, No.5, pp. 148-154, March 1, 1962

This article discusses the friction of rubber seal materials with special attention to shaft surface finish and machine lead, modulus of seal material, corrosion of steel shafts by various elastomers, quality control specification, and economic aspects. Test details, data, curves, etc.

S-142
Symons, J.D.
Lip Seal Failures Can Be Reduced
SAE Jour., V.70, pp. 74-77, May, 1962

Studies surface finish of the shaft material upon which the seal lip rides to reduce friction and seal-shaft interface temperature. Also studies corrosion of the steel shaft by the lip-seal material.

T-001 B
Taber, R.A., and Robbins, F.A.
Teflon Based Piston Rings For Non-Lubricated Applications
Mechanical Engineering, V. 79, No. 9, pp. 834-41, Sept. 1957

Comparison of K30 material, consisting of Teflon and fine glass fibers; with cotton fabric and resin plus graphite; cotton fabric with resin plus molydisulfide; asbestos fabric and resins; glass cloth fabric with epoxy resin; both laminates and moldings.
Tables of wear test and material properties are presented.

T-002
Tangerman, E.J.
Design Proportions For "U" Leather Packings
Product Eng., V. 6, pp. 224-6, June 1935

Packing friction, stuffing-box design and packing proportions for one of four types of commonly used leather packings for light and heavy pressures for both hydraulic and pneumatic service.

T-003
Tankus, Harry
Axial Mechanical Seals
Machine Design, The Seals Book, pp. 32-42, Jan. 19, 1961

Axial mechanical seals (end face seals) form a running seal between flat, precision finished surfaces. Have been used for pressures up to 3000 psi in heavy duty process pumps and for speeds exceeding 50,000 fpm and temperatures above 500°F.
General purpose axial pump seals are discussed, as is the metal-bellows mechanical seal. The latter is used for extreme temperature service and where other types of static sealing elements cannot be used effectively.
Main advantage - low leakage
Main disadvantage - requires careful handling and installation.

T-004
Tanner, R.I.
Non-Newtonian Flow And The Oil Seal Problem
Jour. of Mech. Engng. Sci., V. 2, pp. 25-28, March 1960

Difficult to explain the action of oil seals if the fluid is regarded as Newtonian. Possible hypothesis is that the fluid, when subjected to extremely high shear rates, behaves in a non-Newtonian manner, utilizing a fairly general stress-strain rate relation. A simplified analysis of the seal problem is given. Comparison with the currently available experimental work shows that non-Newtonian effects are not likely to provide a complete explanation of the oil-seal problem.

T-005 T
Tanner, R.I.
The Reiner Centripetal Effect In Face Seals
Int. Conf. on Fluid Sealing, Paper E1 (5 pp.), April 17-19, 1961, British Hydromechanics Research Assn., Harlow, Essex, England

Two possible explanations for inflow against centrifugal force in seals have been advanced. One assuming nonperfect geometry of the sealing surfaces and the other non-Newtonian behavior of the sealing liquid. A review of the literature and order of magnitude calculations establishes the fact that non-Newtonian effect is likely to be insignificant in many practical cases. At the present time, inflow seems best explained by vibration and wobble of seal surfaces.

T-006 T
Tao, L.N., Donovan, W.
Through-Flow In Concentric And Eccentric Annuli Of Fine Clearance With And Without Relative Motion Of Boundaries
ASME Trans., V. 77, pp. 1291-1301, 1955

Derives theoretical flow equations for title conditions. Certain assumptions by authors questioned in discussion by Teichmann. Experimental data, in good agreement with theoretical equation, are presented in graphical form for Reynolds' numbers up to 30,000. Thermal effects, phase change, and compressible flow are not included.

T-007
Taplin, J.R., and Phillips, J.J.
Diaphragm Seals
Machine Design, The Seals Book, pp. 77-82, Jan. 19, 1961

A diaphragm is a dividing membrane. It spans the gap between a moving and a stationary member to prevent interchange of a fluid or gas between two separate areas or chambers.
Three classes: 1. Separating membrane - no pressure differential exists between chambers. 2. Static diaphragm - separator, little or no motion. 3. Dynamic diaphragm - acts as sealing device between stationary and moving members, usually transmits a force or pressure.
Made of cotton, Dacron, Nylon, Teflon or glass.
Depending on choice of material, temperatures up to 700°F are possible.

T-008
Tapsell, H.J.
Pipe Flange Research Committee, Second Report
Proc. Instn. Mech. Engrs., London, V. 141, pp. 433-71, and V. 142, p. 397, 1939

Research has continued to follow the original program with one addition: the relating study of creep relaxation of model flanges. The present report deals with the following sections of work: an investigation of conditions on the joint under which tightness is maintained at room temperature; investigation of behavior of compressed asbestos packing material; the completion of the examination of the properties of the materials used in the bolt assembly tests; experiments on full-scale bolted flange joints under conditions of high temperature and pressure; and creep relaxation tests on model flanges.
Curves, drawings, data tables.

T-009
Taschenberg, E.J., and Pennington, J.W.
Piston Rings For Hydraulics
App. Hydraulics, V. 8, pp. 67-70, 120, Nov. 1955

Basic information on the development of metallic sealing rings. Effective use of the proper sealing ring also requires consideration of dimensions and finishes of the hydraulic cylinders and pistons.
Most commonly used ring material is a high grade cast iron. For applications where lubrication is poor or certain corrosive conditions exist, bronze rings are frequently used. For high temperature applications, stainless steel, Ni-Resist or a high grade chromium cast iron is indicated.

T-010
Taschenberg, E.J.
Evaluation Of Design And Materials For High Speed-High Temperature Shaft Seals For Turbojet Engine Applications
WADC Tech. Report 56-267, ASTIA Document No. AD110636 (65 pp.), May 1956

A large number of materials were investigated to determine those having promising wear characteristics at room temperature and at high temperature. Shaft seal design was reviewed with respect to the adaptability of such new materials to the several designs incorporating these materials.
Design and construction of seals to operate at 200 psi, 1,000°F, and 30,000 rpm.

T-011
Taschenberg, E., and Pearson, W.
Circumferential Seals
Machine Design, The Seals Book, pp. 48-50, Jan. 19, 1961

Circumferential seals are high performance, low leakage, contact type seals for use in critical applications. Undergoing high-velocity rubbing at their primary seal surfaces, the seals offer a high order of sealing ability and accommodate unlimited relative axial motion between stationary and rotating members.
Circumferential seals described are sophisticated versions of the reciprocating rod and piston seals.
Carbon is the commonly used material.
Compressible fluids are most commonly used with these seals.
Upper limit 1000°F.

T-012
Taylor, G.I.
Stability Of A Viscous Liquid Contained Between Two Rotating Cylinders
Phil. Trans. Royal Soc., London, Series A, V. 223, pp. 289-343, 1923

First a review of previous work in flow stability by ten authorities. From general equations representing symmetrical disturbances for the case of incompressible flow between two rotating cylinders, a stability criterion is developed. Extensive experimental results, and pictures of turbulent flow conditions are presented.

T-013 T
Taylor, G.I.
Fluid Friction Between Rotating Cylinders
Proc. Royal Soc. of London, Series A, V. 157, pp. 546-564, 1936

Torque measurements were made for two general cases of concentric rotating cylinders: inner cylinder rotating with outer cylinder fixed and outer cylinder rotating with inner cylinder fixed. Critical speeds were lower for the first case. Several fluids were used, and the diameter of the inner cylinder was varied.

T-014 T
Taylor, G.I., Saffman, P.G.
The Effects Of The Compressibility At Low Reynolds Number
Jnl. Aero. Sci., V. 24, No. 8, pp. 553-562, Aug. 1957

The authors are concerned chiefly with explaining the results and disproving the hypothesis of Reiner, who noted a positive pressure of 0.5 atmosphere at the center of two smooth discs with one disc rotating at 7000 rpm with near zero clearance. Reiner stated this pressure was due to non-Newtonian properties of air. Authors believe the pressure must be due to longitudinal vibration, errors in perpendicularity of either axis to the rotating plane, or roughness. Experiments which support their conclusions are described.

T-015 T
Taylor, I.
Seal-Oil Systems For Refinery Centrifugal Pumps
Petroleum Refinery, V. 22, No. 3, pp. 68-72, March 1943

Reviews various methods whereby an individual pump is sealed, and discusses briefly the ways in which problems are overcome.

T-016 T
Taylor, I.
Alkylation Unit Pumping
Petroleum Refinery, V. 23, No. 8, pp. 283-289, August 1944

This article discusses the pumping problems of alkylation units in general, and the shaft sealing problems in particular.
Single mechanical seals are recommended for light hydrocarbons; packings or double mechanical seals, seals for concentrated acids. Double mechanical seals for mixtures of acids and hydrocarbons.

T-017
Teucher, S.
Cylinder Head Gasket
Automobile Eng., V. 52, pp. 298-303, Aug. 1962

T-018
Thayer, H.H.
Packing And Gaskets For Shipbuilding Purposes
Marine Engineering, V. 27, pp. 720-2, Nov. 1922

Packing of moving parts: assembled metallic packings; metallic packings, misc.; soft packing.
Packing of stationary parts: gaskets, metallic; compressed asbestos fibre; woven asbestos cloth; rubber; rubber compositions; wood fiber.
Boiler gaskets: misc. packings, canvas, corset lacing, tar felt, cup leather, paste jointings.

T-019
Thomson, J.L.
Packed Glands For High Pressures: An Analysis Of Fundamentals
Institution of Mechanical Engineers, Proc., V. 172, pp. 471-486, 1958

Recognizing the uncertain manner in which a seal is effected in a packed gland, and the numerous factors involved, the development of even an approximate theory is indicated.

T-020 T
Thomson, J.L.
A Theory Of Sealing With Particular Reference To The Packed Stuffing Box
Int. Conf. on Fluid Sealing, Paper B1, British Hydromechanics Research Assn., Harlow, Essex, England, April 17-19, 1961

After making the assumption that the relationship between the contact stress and pressure gradient in the stuffing-box seal is linear, it is shown that contact stress increases exponentially and fluid pressure decreases exponentially with distance. With modification, the same principle is applied to the O-ring seal. The influence of contact stress on wear and friction is then considered.

T-021
Thomson, W. E.
High Turbine Vacuum By Attention To Packing
Electrical World, V. 80, p. 490, Sept. 2, 1922

Discussion of methods of obtaining good vacuum:
1) Attention to the carbon packings - primary cause of low vacuum,
2) Turbine starting procedure,
3) Look for air leaks in turbine casing,
4) Check carbon packing spring.

T-022
Thoren, T.R.
Sealing Aviation Fuel System Equipment
Mech. Engng., V. 66, pp. 663-667, Oct. 1944

Synthetic rubber seals satisfactory for sealing aviation fuel-system equipment. Introduction of aromatic blended aviation fuels and requirements of operation down to -65°F required alteration of seals, both in construction and rubber composition.
Manufacturers of aviation fuel-system equipment have made extensive programs of testing to evolve seals now in construction. Paper covers these, specifically for fuel booster pumps, selector cocks, and engine driven pumps.

T-023
Thorn, F.C.
Gaskets
Ind. & Engng. Chem., V. 28, pp. 164-170, Feb. 1936

An attempt is made to classify gasketed joints by types, and to indicate the materials and construction employed in modern gaskets for various types of joint and for the various fluids handled through these joints.

T-024
Thorn, F.C.
Packing Centrifugal Pump
Indus. & Eng. Chem., V. 31, pp. 929-33, Aug. 1939

Attempt made to present case of packing manufacturer; although remarks are applicable primarily to centrifugal pump, many of them are equally true of rotary displacement and turbo pumps, and to other stuffing box applications involving rotary shafts. Drawing showing various packings.

T-025
Thorn, F.C.
Performance Of Centrifugal Pump Packing
Power Plant Engng., V. 45, pp. 80-81, Sept. 1941, pp. 67-69, Oct. 1941

Power plant operator has two ideals in mind, no leakage and no friction. Article covers test methods devised for determining quantitatively what the balance between these two factors should be as applied to centrifugal pumps.
Packing rings cut from braided asbestos coils, with butt joints.

T-026
Thorn, F.C.
Sealability Of Gasketing Materials On Smooth And Wavy Flanges
ASTM Bul, No. 203, pp. 43-44, Dec. 1955

Results of tests on subject matter indicated for three 1/16" gasket materials; red sheet, cork paper with glue-glycerine binder, and compressed asbestos sheet.

T-027
Thorness, R.B., Nier, A.O.
All Metal Valve For Ultra-High Vacuum Use
Rev. Sci. Instrum. (U.S.A.), V. 32, No. 7, pp. 807-10, July 1961

A demountable solderless all-metal valve suitable for ultra-high vacuum applications is described. A gasketed stainless steel diaphragm transmits the motion. One modification is useful for providing a straight path for passage of a beam of particles or light between chambers which it may be necessary to evacuate separately. A group of valves, machined from a single block may form a complete manifold. The basic mechanism has also been employed for transmitting motion in a high vacuum system; for example, adjusting a slit.

T-028
Tilbe, H.E.
Operation Of A 1200 U.S.G.P.M. Shaft Seal Pump At 1050 PSIA And 500°F
R58CAP10 Tech. Report, Canadian Gen. Elec. Co., Ltd., (17 pp.), Feb. 7, 1958

Data are presented of shaft seal performance during the development period of a large shaft seal pump. Pump design modifications necessary to obtain satisfactory performance from seals are described. Total leakage rates since pump modifications have averaged 1 imperial gallon per day per seal. Operation and maintenance data of the seals is given.

T-029 B
Tipton, F.W.
Design Data For O-Rings And Similar Elastic Seals, Part I
WADC Tech. Report 56-272, (104 pp., 15 fig., 12 tab), Nov. 1956, (PB 121898) (Off. Tech. Ser.)

One of three reports describing work of Boeing Airplane Company under U.S.A.F. contract to determine relationship between physical properties of the seal materials and behavior of the complete seal. No definite relationship was found. The test related to static, rotating and reciprocating shafts.

T-030 B
Tipton, F.W., Trepus, G.E., and others
Design Data For O-Rings And Similar Elastic Seals, Part III
WADC Tech. Report 56-272, (88 pp., 39 fig., 19 tab), April 1958, (PB 131902)

One of three reports describing work of Boeing Airplane Company under USAF Contract. Determine relationship between physical properties of the seal materials and behavior of the complete seal, but no definite relationship was found. The test related to static, rotating and reciprocating shafts.

T-031
Tipton, F.W.
Design Handbook For O-Rings And Similar Elastic Seals
WADC Tech. Report 59-428, (60 pp.), October 1959

A summation of work done under Air Force Contract AF 33(616)-2867 and AF 33(616)-5722. It is presented in the form of a handbook covering the mechanism of O-ring sealing, the relation of physical properties to sealing, the effect of cavity configuration, back-up rings and adverse mechanical conditions on seal life, and the design of seals for specific systems. It also contains test procedures and a bibliography of published articles pertinent to O-ring seals.

T-032
Tisch, R.E.
Friction Study Of Hydraulic Seals
WADC Tech. Rept. 55-228, June 1955

T-033
Todd, H.E., Miazga, J.F.
How To Compound Silicones For Seals
SAE Journal, V. 67, p. 106, June 1959

Brief discussions of compounds of 60-70 durometer hardness, compounds containing precipitated silica as a filler. Compounds with fumed silica fillers, diatomaceous silica fillers, ditertiary butyl peroxide as the best of catalysts for low compression, and the pronounced effect of heat on virtually all the properties of silicone rubber compounds.
Parts of paper No. 50V.

T-034
Todd, H.E., and Miazga, J.F.
Properties Of Silicone Rubber For High Temperature Static Seals
SAE Transactions, V. 68, pp. 224-231, 1960

Investigation of silicone rubber compounds for static seal applications demonstrates the importance of tests to evaluate these materials for specific properties and at environmental temperatures which simulate anticipated service requirements. Relationships established between these properties and compound composition; presented as guide for selection of suitable materials for specific seal applications. Maximum temperature of 500°F indicated.

T-035
Toepel, R.R.
Designing Lip Seals For Long Service Life
SAE Journal, V. 70, p. 92, March 1962

Discussion of lip seal design for rotating shafts for automatic transmission, particularly the two main seals. Includes factors in design and application, solving the OD problem, shaft requirements.

T-036
Toffer, H., Amrein, R.L.
Plastic To Metal Vacuum Seal For Rotating Targets
Review of Scientific Instruments, V. 31, pp. 348-9, March 1960

T-037 T
Tourret, R.
Notes On The Development Of Roots-Type Cabin Superchargers
Royal Aircraft Establishment Tech. Note No. SME379, October 1946

A seal described in this report, called a Rotol seal, makes use of a viscosity screw pump in conjunction with a pair of labyrinth seals. The seal, which is of the contactless type was virtually 100 percent oil tight.

T-038
Towler, F.H.
Reciprocating Seals
The Engineer, V. 187, p. 229, February 25, 1949

Working conditions encountered by reciprocating seals compared with those of rotary seals, the mechanism of the automatic seal under static conditions of motion and friction, packingless pistons, and J.F. Stewart's shaft hydraulic seal for aircraft pumps. Diagrams and figures.

T-039
Towler, F.H.
Reciprocating Seals
IME Proc., V. 160, pp. 536-540, 1949

Various types of reciprocating seals are discussed; hemp, leather, rubber, piston rings, and O-rings; conditions for proper operation.
Pressures from 500-10,000 lbs. per sq. inch are indicated.

T-040
Towlson, J.T.
Frictionless Metallic Packing
Power, V. 70, p. 257, Aug. 13, 1929

Original packing box ground out, bronze rings inserted. Precludes the possibility of the packing box being a guide for the piston rod. Dispenses with all kinds of adjustable packing and split rings.
Drawing is presented; assembly and operation discussed in detail.

T-041
Tracy, H.
Improper Suction Piping Can Knock Out Mechanical Seals in Short Order
Power, V. 106, pp. 160-1, July 1962

Poorly designed suction piping can do a lot of damage. Example presented. Pump handling propane at 100°F. Pressure differential of 285 psig.
Drawing and detailed description.

T-042
Tracy, H.E.
Select Best Pump Seal
Chem. Eng., V. 64, No. 4, pp. 239-254, April 1957

Centrifugal pump leakage devices can be divided into zero leakage types and controlled leakage techniques; recommended methods for preventing or eliminating leakage of hydrocarbons, water, chemical solutions, molten metals, and liquid gases at various temperatures and pressures; use of mechanical seals, throttle bushing, floating rings, and canned pumps; materials of construction are discussed only as materials are affected by pressure and temperature.

T-043
Treiber, K.L.
Hydraulic Problems Encountered During The Development Of Corps Of Engineer Equipment
Proc. National Conference on Industrial Hydraulics, V. 8, pp. 5-24, 1954

A detailed description of two mechanical seals used on two stage and four stage portable pumps (In the panel discussion by V.E. Vorhees).

T-044 B
Trepus, G.E.
Design Data For O-Rings And Similar Elastic Seals, Part II
W.A.D.C. Tech. Report 56-272, (106 pp.), Sept. 1957

One of three reports describing work of Boeing Airplane Company under U.S.A.F. contract to determine relationship between physical properties of the seal materials and behavior of the complete seal. No definite relationship was found. The test related to static, rotating and reciprocating shafts.

T-045
Trepus, G.E., Roper, R.S., Hickman, W.R.
Design Data For O-Rings And Similar Elastic Seals
WADC-TR-56-272, Part IV

Continuation of the study initiated under Air Force Contract AF 33(616)2867. The purpose of this study was to determine design criteria for O-rings, back-up rings and other elastomeric seals.

T-046
Trepus, G.E., Roper, R.S., Hickman, W.R.
Design Data For O-Rings and Similar Elastic Seals
WADC Tech. Rept. 56-272, Part V

Continuation of a study initiated under Air Force Contract AF 33(616)-5722. The purpose of this study was to determine design criteria for O-rings, back-up rings and other elastomeric seals.
Seals for use in cryogenic systems,
Effects of ozone on elastomeric seal materials,
Literature review of age control techniques,
Abstracts.

T-047
Trevoy, D.J., and Torpey, W.A.
Shaft Seal For Vacuum Apparatus
Anal. Chem., V. 24, p. 1382, 1952

A shaft seal is constructed of a 0.081 in. drill rod matched to a precision capillary tube of the same nominal diameter.

T-048 B
Troyan, J.E.
Hints For Plant Start Up - Part III - Pumps, Compressors, And Agitators
Chemical Engineering, V. 68, pp. 91-4, May 1, 1961

Typical problems arising from the use of mechanical seals, with emphasis on lubrication, are discussed in a part of the article. The author also considers stuffing boxes and packings.

T-049
Trummel, J.M.
Irradiation Test Of Static Buna - N O-Ring Seals
Oak Ridge National Lab. Tenn., CF-54-8-226
Contract W-7405-Eng-26, Aug. 25, 1954, Decl.
Feb. 20, 1956

The object of this test was to determine the life expectancy of four groups of Buna-N O-ring oil seals under gamma irradiation. These static seals were used to seal Gulfspin 60 oil at 75 psi pressure differential at about 125°F in a test arrangement placed in the flux of low intensity testing reactor. Test results noted.

T-050
Trutnovsky, K.
Labyrinth Slits In Piston Engines (In German)
Forschung aud dem Gebiete des Ingenieurwesens, V. 8, pp. 131-43, May-June 1937

Usual packings discussed and apparatus described for investigating losses in new form of packing; characteristic size of labyrinth slit was modified to measure influence of these modifications on losses; example of calculating losses in advance by means of formulas for flow in tube; advantages of labyrinth slits.

T-051 T
Trutnovsky, K.
Labyrinth Packing (In German)
VDI-Zeitchrift, V. 83, No. 29, pp. 857-858, July 22, 1939

T-052 (continued abstract)

The advantages of labyrinth seals in high pressure and high temperature applications are discussed. Characteristics of this type of seal in comparison with other types are shown. This report lists 16 references.

T-052
Trutnovsky, K.
Design And Action Of Stuffing Boxes (In German)
VDI Zeit., V. 85, pp. 383-7, April 19, 1941

Causes of leakage; methods and means for eliminating leakage are shown in schematic drawings; special attention paid to materials requiring no lubrication; data on leakage and friction losses and packing dimensions.

T-053 T
Trutnovsky, K.
Contactless Seals (In German)
VDI, Verlag, Berlin 1943, Published by authority of the Alien Property Custodian Under License No. A828 by J.W. Edwards, (155 pp.), 1943

This book, which includes 72 references, discusses theory, design, and construction details of clearance seals and labyrinth seals. This complete study of contactless seals is the only book which has been found entirely devoted to this subject. Dynamic seals and buffered bushing seals are briefly discussed.

T-054 B
Trutnovsky, K.
Research On Labyrinth Glands (In German)
D.I.Z., V. 101, No. 18, pp. 752-5, June 21, 1959

This paper summarizes recent work in Germany on labyrinth seals as used in steam turbines. It appears that little can be gained by using conical shaft discs and housing grooves in axial seals. A simple arrangement using radial discs and rectangular ones having a depth, width and pitch of 3, 5.5, 8 mm gives a good performance clearance of 0.38 mm.

T-055
Turnbull, A.H.
Vacuum Techniques For Beginners
A.E.R.E. - Rep. No. G R 752 - 23.8.51, (UK Atomic Energy Research Establishment Report)

An introduction to vacuum technique written for the engineer who wishes to familiarize himself with the working principles and operating conditions of a vacuum plant. The following subjects are covered: rotary pumps, diffusion pumps, pump matching, types of gas flow, vacuum measurements, vacuum components, vacuum unions, shaft seals, vacuum flanges, vacuum valves, vacuum grease, wax, etc.

T-056 B
Turnbull, D.E.
The Sealing Action Of A Conventional Stuffing Box
British Hydromechanic Research Association, RR 592, (16 pp., 16 fig.), July 1958

A brief review of previous work on the frictional properties of various soft packings. Definitions are suggested. Several non-dimensional parameters are described. A choice is made of an approximation to the friction characteristics. Using this, the theoretical axial pressure distribution along a packing is calculated. Experimental data and analysis.

T-057 B
Turnbull, D.E., Nau, B.S.
Some Effects Of The Elastic Deformation Of The Sealing Rings Of A Mechanical Or Radial Face Seal On Its Sealing Behavior
British Hydromechanic Research Association, RR 644, (5 pp., 9 fig.), Feb. 1960

This report examines the effect of elastic deformation of the faces of a radial-face seal. It is shown that at pressures of the order of 1600 lb/sq. in. the local deformation is several times greater than the usual thickness of the oil film and this is liable to give rise to prohibitive leakage rates. A possible method of overcoming this effect is described.

T-058 B
Turnbull, D.E., Nau, B.S.
A Simplified Analysis Of The Characteristics Of Fabric-Rubber Seals For Reciprocating Shafts
British Hydromechanic Research Association, RR 648, (5 pp., 7 fig.), Feb. 1960

T-059
Turnbull, J.C.
Tension Stresses In Glass Coatings And In Glass-Metal Seals In The Annealing Range
Amer. Ceram. Soc. Jour., V. 41, No. 9, pp. 372-376, Sept. 1958

Strength of glass coatings applied to metals decreases with time at temperatures in the lower annealing range. This is caused by decrease in the volume which glasses undergo. Tension strains are thus produced in directions parallel to the surface. In this temperature range, the production of tension stress due to density increase is partly offset by stress release due to viscous flow. Significant tension stresses of similar origin were observed to appear in glass - metal seals during baking in the lower annealing range.

T-060 B
Turton, B.F.
Eliminate Trapped Oil To Increase Packing Life
Applied Hydraulics and Pneumatics, V. 11, No. 11, pp. 101-2, Nov. 1958

Trapped oil between two seals back to back, can result in poor packing performance.
Design precautions taken to prevent this.
Some common failings, and examples of wear.

U-001
Ubbelohde, A.R.
Pyrolitic Graphite
Elec.Rev., V.170, pp.91-92, Jan. 19, 1962

Has remarkable thermal, mechanical and electrical properties. Because of the marked preferred orientation of crystallites, there is pronounced antistrophe in directions parallel and perpendicular to the plane of deposition. Co-deposition of carbon with other atoms, such as boron, may extend even further the technical applicability of these new pyrolitic materials.

U-002
Ullman, J.R.
Commercial Seals as Seats in a Bakeable Valve
Am.Vac.Soc., Trans. Eighth Nat.Vac. Symposium, Second International Congress, V.2, p. 1323, 1961

A simple bellows-sealed 3-in. valve is described which is based on the commercially available "Marman Conoseal" tube joint for the gate seat. The application is evaluated and the valve operating characteristics are given for the test valve, which gave leakage conductance of 10^{-6} to 10^{-10} atm/cm^3/sec for air.

U-003
Underwood, C.A. and Darnell, J.R.
Gaskets and Packings and How to Use Them
Power Plant Eng., V.27, pp.813-6, Aug. 15, 1923

Proper packings to be used on high and low pressure steam reciprocating engine shafts; combinations of duck, rubber and asbestos make good low pressure packings.

U-004
U S. Aeronautical Board (with amendment 2 July, 1946)
Hydraulic Packing Rings (Backup)
PB 40299, Off.Tech.Serv.,(9 pp.), May, 1945

These back-up rings of leather are intended for use with hydraulic O-rings operating at pressures up to 3000 psi and at temperatures up to 71°C (160°F). Materials and workmanship, detail requirements, methods of sampling, inspection and test are specified.

V-001
Van Heerden, P.J.
Metal Gaskets for Demountable Vacuum Systems
Rev.Sci.Instrum., V.26, No.12, pp. 1130-1, Dec., 1959

Two different constructions of copper gasket seals were investigated for a demountable vacuum system. They were 5 1/2 inches in diameter, and for both a leak rate lower than 10^{-6} microliter/second could be obtained.

V-002
Van Koppen, G.M.
A Metal Vacuum Valve
Appl.Sci. Research, B3, V.141, 1953

A 3-way valve with O-rings for all seals is described. After 5000 revolutions, the O-rings did not need replacement.

V-003
Van Wicklin, W.A.
Lincoln Power Steering Gear
National Conf. on Industrial Hydraulics, V.12, pp. 67-82, Oct., 1958

Discussion on seals, power cylinder seals, cover seals, shaft seals. "D" ring of 70 durometer synthetic rubber. O-ring seal. Brief general discussion.

V-004 B
Vermes, G.
A Fluid Mechanics Approach to the Labyrinth Seal Leakage Problem
Trans. A.S.M.E., V.83, pp.161-9, April, 1961

The paper describes investigations of labyrinth seals carried out recently; derives new theoretical, and semi-theoretical formulas for computation of the leakage which agrees within 5 percent with the tests for three different types of seals; off design performance of the seals is treated theoretically and experimentally

V-005
Vermes, G.
Leakage Flow in Straight Labyrinth Seal, Nomograph
Power, V.106, p.62, January, 1962

This nomograph is based on "A fluid mechanics approach to the labyrinth seal leakage problem," Transaction ASME-Journal of Engineers for Power, April, 1962.
It gives the leakage flow, lbs. per sec., of a seal with N sealing points, spaced 5 inches apart, with a radial clearance C, around a straight shaft. Equations given.

V-006
Vermes, G. and Hahn, R.M.
New Approach to Labyrinth Seal Analysis
Allis-Chalmers Elec.Rev., V.25, No.4, pp. 24-25, 1960

Minimizing steam leakage through the shaft seals of a turbine has provoked studies and investigations for more than 50 years. Higher pressures and reduced space have aggravated this problem. It has been found that fluid mechanics provides more precise solutions to seal problems, resulting in greater simplicity and increased seal efficiency. Approach applicable to steam turbines, gas turbines, compressors, blowers, etc.

V-007
Victor, J.B.
Gaskets: Their Functions, Application, and Design
Product Engng., V.1, pp. 274-275, 283, June, 1930

A 3-page article on various materials that are used for gaskets such as paper, cork, rubber, asbestos, corrugated metal and metal-enveloped asbestos; their design and application.

V-008
Vieweg, Capt. W.V.R., Brown, F.W.
The Design of O-Ring Seals for Rockets
Tech. Memo. No.1046, U.S.Naval Ord. Test Sta., China Lake, Calif., March 17, 1956

V-009
Vitale, J.A.
Report on High Pressure Gasket Test
MIT Laboratory for Nuclear Science and Engineering Technical Report No. 1, (NP-1039) (7 pp.), Feb. 1, 1947

A gasket arrangement suitable for high-pressure work was developed and tested as applied to a high-pressure cloud chamber. A satisfactory gasket scheme for this application must be as nearly leak-proof as possible at 3000 psi, even when the initial or tightening pressure is relaxed due to deflection, be easily assembled, not require too close machining tolerances, and not be very position sensitive. The pump used was a standard hydraulic jack of 5-ton capacity.

V-010 T
Vogelpohl, G.
Contributions to the Knowledge of Bearing Friction (In German)
VDI-Forschungsheft, No.386 (28 pp.), 1937

On the basis of experiments, the author concludes that cavitation in a journal bearing is a function of the clearance diameter rates, viscosity and shaft speed. When the relative clearance is low, the cavitation will occur at lower speeds. Oil cavitation differs from water cavitation. Treated oil (degassed) results in less cavitation. The observations by the author may be of value in study of cavitation and/or gas release in buffered bushing seals.

V-011
Von Maur, J.D.
Use of Rubber Gaskets for Joint Making Purposes
American Gas Journal, V.140, pp.17-20, June, 1934

V-012
Vueta, M.P.
Hydraulic Fracturing in the Grosknett Field
(MTWL: 832, 6 pp.) Order from OTS or ETC - Trans. No 61-7380 (Trans.of Neftyano Khozyaistvo (USSR), V. 34, pp. 22-24, 1956

Hydraulic seals, mineral oils, production petroleum.

W-001
Wadey, W.G.
Standards for Vacuum Fittings
Vacuum, V.4, No.1, pp.53-67, Jan. 1954

A report on a system of vacuum flanges sealed by O-rings which is being considered for presentation to the American Standards Association by a committee of the technical society. The Committee on Vacuum Techniques for adoption as an American Standard. Drawings and tables included.

W-002
Wagner, R.E.
These Gaskets Are Sprayed On
Product Engineering, V.28, pp.90-91, Oct.14, 1957

This new technique uses a standard metallizing method to coat cover plates with aluminum. Type 1100 aluminum alloy built up to 0.032" thickness. Re-use of joint limited to three or four times. Details of preparation, application, and use are presented. Pictures included.

W-003
Wakefield, J.E.
Metallizing of Packing Areas
Welding Journal, V.28, pp.875-876, Sept., 1949

Procedures for rebuilding worn packing areas on pump shafts, sleeves, rams, plunger, turbine shafts, and other parts.

W-004
Wakeman, W.H.
Packing Piston Rods
Am. Mach., V.21, pp.465-6, June 23, 1898

Gives points on the packing and oiling of piston rods.

W-005 T
Walker, J.E.
Shaft Sealing Means for Pumps
U.S. Patent 2,311,641, Oct. 12, 1943

Seal for a suction lift centrifugal pump used to pump fluids containing abrasive materials. The sealing members are closely spaced synthetic rubber parts which deform to allow passage of abrasives. A synthetic rubber jacket is vulcanized to a metal sleeve keyed to the shaft.

W-006
Walker, W.R.
Design Handbook for O-Rings and Similar Elastic Seals
WADC Tech.Report 59-428, Part II (103 pp.), April, 1961

Data presented herein is concerned with hydraulic and pneumatic systems utilizing static and dynamic type seals at temperatures exceeding 275°F and for static applications at cryogenic temperatures. Hydraulic systems; pneumatic systems; fuel seal systems; vacuum seals; cryogenic systems seals; aging and age control.

W-007
Walley, J.W.
Magnetic Seal for Rotating Shafts
AEI Engng., V.2, pp. 89-93, Mar./Apr., 1962

A flexible seal is formed by a membrane of iron particles in oil, the iron particles being attracted by a magnetic field between a rotating shaft and its bearing housing. Advantages are absence of wear in the gland itself and ability to allow for shaft displacement. Possible applications are electrical machinery in mines and oil refineries, gas and vacuum systems, turbines, and gear boxes.

W-008 T
Walmsley, J.G., Ward, H.A., Jr.
Mechanical Shaft Seals; Troubles and Trends
Petroleum Refiner, V.32, No.9, pp.203-206, Sept., 1953

Description of the design operation, maintenance, and cost of mechanical seals. The difference between balanced and unbalanced seals and the relation of the seal loading to seal life is considered.

W-009 T
Walti, F.D.
The Oil-Free Reciprocating Compressor
Engineering, V.177, pp. 533-535, April 23, 1954

Describes the use of labyrinths to replace piston rings in a reciprocating compressor. Leakage rates which approach those of piston rings have been obtained. The chief advantages are lack of oil contamination of the compressed substance and low friction losses. Labyrinth profiles are shown, and experimental data correlating leakage rates. Clearance gap, profile, length of seal, and initial pressure are included.

W-010
Anon.
Prerequisites for the Study of Piston Ring Seals
PB-36533, Off.Tech.Serv., 40 pp., Jan. 1941

Former improvements on gas sealing piston rings were concerned mainly with materials and processing. The construction shape in most cases is determined by the square or rectangular cross-section of the "Ramsbottomring." Previous skills combined with new methods with consideration of secondary phenomena lead to correct evaluation of piston ring seals. Photographs and diagrams are included.

W-011 T
Wannier, G.H.
A Contribution to the Hydrodynamics of Lubrication
Quarterly of Applied Mathematics, V.8, No.1, pp. 1-32, April, 1950

Comparison between the Reynolds equation and Stokes equation. The author suggests that the formulas for load capacity and friction coefficient developed in the paper have more rigor and generality than the normally used equations, but that they may be of limited usefulness due to lack of understanding of the low-pressure region of the bearing. The author theorizes the possibility of reverse flow in the divergent flow region.

W-012
Warburton, H.
White Metal Stuffing Box Rings
Mech. World, V.110, pp. 173-5, Sept.12, 1941

Packing material in stuffing boxes of steam engines, steam pumps, fuel and other pumps, etc. must, to function efficiently, be able to withstand varying temperatures and pressures to which it is subjected without becoming hard and dry with consequent scoring of rod; metallic packings for general and special purposes are described.

W-013 T
Ward, J.
Chart for Leakage in Labyrinth Packing
Engineering, V.128, p. 65, July 19, 1929

A formula is given which is commonly used in Great Britain and the U.S. for calculation of leakage through a steam turbine dummy piston or labyrinth seal in which the cross-sectional area of flow is constant. The author has developed a chart for rapid evaluation.

W-014
Warring, R.H.
Felt Washer Seals
Power Transmission, V.26, No.308, pp. 698-99, 1957

W-015 B
Warring, R.H.
Metal O-Rings
Hydraulic Power Transmission, V.4, No.37, pp. 34-5, Jan., 1958

W-016
Warring, R.H.
Dynamic Applications of O-Rings in Hydraulic Services
Hydraulic Power Transmission, V.5, No.52, pp. 250-253, April, 1959

Basically simple seal effective at fulfilling requirements of dynamic operation, available in wide range of materials for meeting specified mechanical requirements and for resistance to particular hydraulic fluids; grooves for rings may be trapezoidal, V-shaped, or rectangular for static seals; simple installation, excellent sealing characteristics, and simple maintenance noted.

W-017 T
Warring, R.H.
Hydraulic Seal Friction and Performance Factors
Design News, V.14, pp. 54-57, Nov. 9, 1959

A chart for estimating seal friction is included with a discussion of seal friction and the variation of friction coefficient with pressure.
This article is concerned only with lip seals.

W-018
Warring, R.H.
Evaluation of Frictional Losses with Seals
Hydraulic Power Transmission, V.5, No. 60, pp. 772-775, Dec., 1959

Not all mechanical losses are due to simple rubbing friction; extrusion of seal under working conditions may produce severe wedging of flexible type seals and packing; chart enables determination of friction force for elastomer seals by relating seal diameter, pressure of system, and coefficients of friction for varying pressures.

W-019
Warschauv, D.M., Paul, W.
Unsupported Area High Pressure Seal
Rev.Scien.Instr., V.28, No.1, p.62, Jan., 1957

Describes modification of the unsupported area seal developed by P.W. Bridgeman
Neoprene rubber, optical plugs, Teflon, were used.

W-020
Wasbauer, A.M.
Improved Closures for Anti-Friction Bearings
Product Eng., V.4, pp. 418-9, Nov., 1933

Felt seals, metal sealing rings, labyrinths, grooves and slingers used for sealing bearing housings.

W-021
Waters, E.O., Weatstrom, D.B., Williams, F.S.G.
Design of Bolted Flanged Connections
Mechanical Engineering, V.56, pp. 736-8, Dec., 1934

The authors are commenting on the important phases of flange design with the hope that they may cast light on some of the reasons that lie behind various sections of rules and codes of the ASME, API-ASME for pressure vessels. Areas covered are gasket design, bolting design, and flange design. Also presented is a table of contact pressure ratios.

W-022
Waters, E.O., Weatstrom, D.B., and others
Bolted Flanged Connections: Formula for Stresses
ASME Trans., V.59, pp.161-169, April, 1937

This paper outlines a revised analysis based on the ring, tapered hub, and shell being considered as three elastically coupled units loaded by a bolting moment, a hydrostatic pressure, or a combination of the two. Design formulas and charts are given for the computation of stresses that are likely to be critical, and their application is illustrated by three problems taken from current commercial practice.

W-023
Watson, A.
Stuffing Boxes Dispensed With on Engines and Pumps
Instn. Mining Eng., Trans., V.25, pp.259-263, 1902-1903

A paper read before the Min. Inst. of Scotland, on stuffing boxes without glands. Also abstract of discussion.

W-024
Watson, C.L., O'Neil, J.E.
Electronic Equipment Pressurization System Study
PB 131713, Off.Tech.Serv. (302 pp.), Jan., 1958

A study and evaluation of various pressurization systems which might be employed to pressurize the sealed containers and replenish the air lost from the containers through mechanical seals.
AD 148270

W-025 B
Webster, E.A.
The Evaluation of High Temperature Hydraulic Seals
WADC Tech.Report 58-120 (96 pp., 71 fig., 39 ref.), Jan., 1955

Report of an extensive testing program carried out by the Douglas Aircraft Company under Air Force Contract with the object of developing high pressure piston seals suitable for a temperature range of -65°F to 300°F. The investigation covered some sixty different types of seals and material, including determinations of temperature limitations of current O-ring seals and back-up rings. A bibliography of thirty-nine publications relating to high-temperature seals is included.

W-026
Webster, E.G. and Larkin, R.G.
Choosing Rubberlike Materials for Use With Synthetic Hydraulic Fluids
SAE Jour., V.67, No.3, pp. 41-43, March, 1959

Factors in selection of basic gum or family of polymers for seals and gaskets, flexible hose liners, line clamp cushions and accumulator diaphragms in aircraft applications; methods of predicing material compatibility with particular fluids are "similarity rule" and "solubility parameter concept"; attributes of three typical O-ring materials with regard to hardness, ultimate tensile strength and compression set.

W-027
Webster, E.W.
Vacuum Tight Mechanisms
Electronic Engng., V.17, pp.53-57, July, 1944

Typical joints allowing movement include the simple cylindrical cone, Wilson's sliding seal, rubber tubing and metallic bellows. Magnetic or electromagnetic operation is best for many applications. Many practical notes are given.

W-028
Wehrse, R.
Glass-Sealing for Pressurized Cabins
Lilienthal Ges. Luftfahrtforsch, p. 71, Report 129
Spec. Lib. Assn. Trans. 3192 (24 pp.), V.3, No.8, August, 1957

W-029 B
Weimer, H., Gilbert, E.
Non-Lubricated Reciprocating Compressors. Part II-Synthetic Carbon as Sliding Contact Material and Its Industrial Application (In German)
Vereis Deutscher Ingenieure Zeitschrift, V.101, pp. 596-602, May 21, 1959

In many aspects, the properties of synthetic carbon differ fundamentally from those of metals employed for sliding contact duties. Consequently, there are a number of viewpoints that must be taken into consideration when designing synthetic carbon machine elements. Successful examples quoted.

W-030 B
Weitzel, D.H., Robbins, R.F., and others
Elastomers for Static Seals at Cryogenic Temperatures
Paper D6; Proc. 6th Cryogenic Engineering Conference, pp. 219-27, August, 1960

W-031
Weitzel, D.H. and others
Low Temperature Static Seals Using Elastomers and Plastics
Review of Scientific Instruments, V.31, pp. 1350-1, Dec., 1960

Demountable high vacuum or high pressure static seals for source at cryogenic temperatures can be made from common elastomers such as natural rubber, nitrile rubber, and Viton A, as well as from high strength plastics such as Mylar and Nylon. Flanges must be designed to complement the properties of the seal material. Examples, test data, and discussion presented.

W-032
Wells, V.W.
Frequent Cause of Packing Failure
Power Plant Eng., V.39, p. 352, June, 1935

Failure due to the fact that the inside follower plate did not properly grip the packing with the result that when the leather became wet and soft, friction of the cylinder gradually stretched and pulled it out from under the follower plate.

W-033
Werkenthen, T. ., Swenson, A.D. and others
Sealing and Sea-Aging Properties of Rubber Gaskets
Rubber Age, V.56, pp. 389-896, Jan., 1945, pp. 513-8, Feb., 1945

Account of investigation of medium, soft, natural and synthetic rubber gaskets for various types of closures. (Bibliography)

W-034
Wertheim, F.E.
Gaskets and Packing for Hydraulic Power
Heating-Piping, V.4, pp. 202-3, March, 1932

Brief article and drawings. Advantages of hydraulic power; packing piston for hydraulic cylinder; pipe joints; designing a copper gasket; use of soft iron gasket; piping should be constrained to avoid whip.

W-035
Wertheim, F.E.
Designing Copper Gaskets for Hydraulic Piping
Heating-Piping, V.5, pp.88-9, February, 1933

Author-reader comments presented on previously published article. Derivation of the Baumann formula used in design of flanged joints is also given. Table of data - Compression Tests of Sheet Copper, Drawing.

W-036
Wertheim, F.E.
Holding Properties of Gaskets Studied
Heat, Piping & Air Cond., V.8, pp. 367-9, July, 1936
(Abstract translation of a German paper by E.Seibel and others in Forsch.Gebiete des Ingen., V.5, pp. 298-305, Nov.-Dec., 1934)

Investigation of holding properties of gaskets subjected to steam heat; various gasket materials and shapes.

W-037
Westbrock, A.J., Cornell, J.J.
Seals, Bofors 40 MM Gun, Recoil Cylinder Packing Ring
PB 31685, Off. Tech. Serv. (28 pp.), March, 1944
(Chrysler Corp.Eng.Div., Tech.Rept. G-61806.12)

Results of tests made to compare the durability of various sealing elements for the recoil cylinder packing ring of a Bofors 40 mm gun to determine which type gives the best sealing quality in the range of temp. from -40°F to 150°F. Seals of the original design and material proved most satisfactory. Illustrations, graphs and tables are included.

W-038
Wexler, A. and others
Low Temperature Gaskets
Review of Scientific Instruments, V.21, pp. 259-60, March, 1950

Gold wire, 0.020 in. dia., sealed the copper container.
Because of the general interest in the tightness of the gasket when the seal is exposed to superfluid helium, the test method was elaborated so as to decrease the lower limit to 5 x 10^{-9} micron liter/sec.
Details of this specific case presented.

W-039
Whalen, J.J.
Leakage and Elastic Characteristics of Compressed Asbestos Sheet Packing
ASME Paper N58-SA-28, p.27, June 15-19, 1958

Variables investigated were bolt load, gasket width, gasket thickness and flange surface finish. Leakage data were correlated with respect to m-value; it was found that m-value is not constant and typical curves were derived from various leakage pressures and area ratios; new design equations derived to predict required total bolt area necessary for joint tightness.

W-040
Whalen, J.J.
How to Select The Right Gasket Material
Product Engineering, V.31, pp. 52-56, Oct. 3, 1960

Presents a 4-step procedure: (1) calculate total bolt force applied to joint at time of installation; (2) select a material that will seat; (3) check the hydrostatic end force, and (4) specify the flange-surface finish.
To follow this procedure, 5 tables are given of: (1) pressure-temperature value for gasket materials; (2) stress areas for flange bolts; (3) minimum seating stresses for typical gasket materials; (4) typical gasket cross-sections, and (5) safety factors for gasketed joints.

W-041
Whalen, J.J.
Select and Apply Gaskets Effectively
Chemical Engineering, V.69, No.20, pp.83-88, Oct. 1, 1962

Some points to remember when specifying gasketed joints, with an emphasis on installation bolt force and seating stress.
How to determine the suitability of gasket material, compressed asbestos or metal gasket.
Gasket application; minimum stress for sealing; calculating installation bolt force; gasketed joint analysis; check points.

W-042 B
Whalley, E.
O-Ring Seals for Pressures of 10 Kilobars
J. Sci. Instrum., V.36, No.1, pp. 47-8, January, 1959

Description of O-ring seals that have been used up to 10 kilobars successfully. Drawings included.

W-043
Wheeler, W.R., Carlson, M.A.
Ultra-High Vacuum Flanges
Transactions of the Vacuum Symposium, 1961

A study has been made of the performance characteristics of four leading designs of all metal flange seals. The aim was to determine which of the designs offered the best combination of performance, reliability, convenience of use, and practical manufacture. Data are presented on leak rate, sealing life, take out to 880°C, resistance to mechanical damage, technique of use, and corollary information. Analysis of the sealing mechanism is also presented to explain the differences in performance.

W-044 T
Whipple, R.T.P.
Herringbone Pattern Thrust Bearing
Atomic Energy Research Establishment T/M29 (8 pp.), August 24, 1949

The bearing consists of a plane circular disc or ring rotating about its axis parallel and slightly separated from a stationary disc or ring on which a set of grooves is arranged in a herringbone pattern. The moving plate drags the fluid into the grooves and develops a pressure which reacts against the axial load, thus forming a thrust bearing.

W-045 T
Whipple, R.T.P.
Theory of the Spiral Grooved Thrust Bearing with Liquid or Gas Lubricant
Atomic Energy Research Establishment T/R622, March 6, 1951

The stated object of this paper was to calculate the thrust for the spiral groove thrust bearing and to find the optimum shape of the grooves at low speed. This type of bearing uses constant-depth grooves cut into the surface of a circular plate to generate, hydrodynamically, a thrust carrying pressure distribution within the bearing.

W-046
White, C.M. and Denny, D.F.
The Sealing Mechanism of Flexible Packings
Gt.Brit., Ministry of Supply, Scientific and Technical Memorandum No.3-47, Jan., 1947

Tests made to show how pressure is distributed over the surface of flexible packings and how they deform under pressure. Modifications to reduce extrusion.

W-047
White, J.C.
Why Gaskets Fail
Nat. Engr., V.49, pp. 166-7, March, 1945

Analytical discussion of basic requirements of tight gasket and some of the reasons why gaskets fail in normal service; design details of ring gasket that remained tight for 12 years under difficult service conditions; unit pressures on fibrous and ring gaskets; why fibrous gaskets leak.

W-048
White, W.P.
Improvements In Or Relating To Sealed Joints
British Patent 896, 352, May 16, 1962

A sealed flanged joint is designed for use between pipes of metals having different thermal expansion coefficients.

W-049 T
Whitefield, J.E.
Shaft Seal
U.S. Patent 2,732,232, Jan. 24, 1956

A shaft seal which is converted into a labyrinth after installation. A hollow annular ring made of soft metal (softer than the shaft) is press fitted into the housing. The shaft has a circumferential Vee-shaped ridge machined on it and fits through the seal with small clearance. The hollow space in the sealing ring is provided with a pressure fitting and after installation the shaft is pressurized, forcing the soft metal into the grooves and forming a labyrinth seal.

W-050 T
Whitley, S., Williams, L.G.
The Gas-Lubricated Spiral-Groove Thrust Bearing
United Kingdom Atomic Energy Authority TG Report 28 (RD/CA)(33 pp.), 1959

An existing hydrodynamic theory of load-carrying capacity is developed, and optimum values of the four important groove variables are obtained. Three thrust plates of outside diameters 5.55, 6.5 and 9.5 inches, respectively, are investigated experimentally, and the results are compared with theory.

W-051
Whitley, S., and Williams, L.G.
Principles of Gas-Lubricated Shaft Seals
Jour.Mechanical Engng. Science, V.4, pp.177-187, June, 1962

Four seal types for rotating shafts: hydrodynamic disc seal (based on the spiral-groove thrust plate), hydrostatic disc seal (based on a hydrostatic full-face thrust plate), hydrodynamic sleeve seal (based on a hydrodynamic full journal bearing), and hydrostatic sleeve seal (based on a hydrostatic journal bearing).
Examples of each type for use on a 2 in. shaft at 9000 rpm with zero pressure in the sealed system. Seals used to separate oil-lubricated bearings from the atmosphere can also be used for sealing a system containing gas at high pressure and temperature, such as in a gas-cooled reactor.

W-052
Wiest, E.N.
Metallic Stuffing Box Packing
Am. Mach., V.19, p. 645, July 2, 1896

A two-ring pac. ing, illustrated and described, with composition of alloy for such packing.

W-053 T
Wigg, R.E., Battle, N.
Improvements In Or Relating To Sealing Means
Great Britain Patent 834, 923, published May 11, 1960

A modified design of the screw viscosity pump arranged to provide a shaft seal against leakage of pressurized gas in conjunction with the self-aligning visco seal. Mechanical seals are provided and arranged to become automatically effective under low speeds and stationary conditions. Three designs are shown.

W-054
Wigotsky, V.W.
Silicone Rubber for Seals, O-Rings and Gaskets
Design News, V.16, pp. 8-11, April 24, 1961

Good retention of elastomeric properties and flexibility when exposed to temperatures of 500°F constantly and up to 600°F intermittently; flexibility without cracking when exposed to temperatures as low as -130°F. Good immunity to water and humidity, and little effect on physical and electrical properties by a variety of chemicals, oils and fluids.

W-055 T
Wilcock, D.F.
Turbulence in High-Speed Journal Bearings
Trans.Amer.Soc.Mech.Engrs., V.72, pp.825-834, August, 1950

Measurements of operating characteristics of journal bearings up to high surface speeds have revealed abnormalities beyond a certain critical value. A rapid increase in bearing torque, power loss, and oil film temperature occurs as speed increases beyond the critical value, while oil flow decreases below normal. These phenomena are attributed to instability or turbulence in bearing oil film.

W-056
Wilkinson, H.N.
Seals for Roll Neck and Roller Table Bearings
Engineer, V.204, pp.409-415, Sept. 20, 1957

Problems discussed are pertinent to steel mill installations. This report covers special problems involved and surveys the various kinds of seals available, their suitability and main requirements for their efficient operation.
The lip seal is dealt with in detail with temperatures to 150°F and pressures of 30 lb.per sq.inch being measured.

W-057 T
Wilkinson, S.C.W.
Development in the Use of Radial Face Mechanical Seals for Gas Sealing Applications
Int.Conf.on Fluid Sealing, Paper C3 (7 pp.), April 17-19, 1961
British Hydromechanics Research Assn., Harlow, Essex, England

Outlines four years of progress, 1956-1960, in gas sealing techniques by Crane Packing Ltd.
Early experimental investigation led to the design of the main coolant gas circulator drive shaft seal for a nuclear power station. Problem and its solution of coolants covered.
The limiting conditions for gas seals have been made as high as those for liquid seals.

W-058
Wilkinson, S.C.W.
Mechanical Seal Design
Engineering Materials and Design, V. 5, pp. 572-578, 664-667, Aug.-Sept., 1962

Radial face-type shaft seals. The following design features are discussed: surface finish, mechanical seal distortion, hydraulic face distortion, heat dissipation paths, adequate cooling circulation, wetting properties, fluid pressure relative to face loading, seal balance, and solids in suspension.

W-059 T
Williams, A.E.
Machined Carbon Products
Mechanical World, V.130, pp.107-09, August 3, 1951

This article describes some general features of two uses of carbon products: mechanical seals, and vanes for pumps and compressors.

W-060
Williams, B.G
Mechanical Seals. I: A Survey of Present Day Practice.
Chemical & Process Engineering, (I) V.36, pp.73-78, March, 1955; (II) V.36, pp.124-126, April, 1955

I - Types and application. Graphs and diagrams.
II- Operation, maintenance, and applications.
Graphs and diagrams.

W-061
Williams, H.T.
Grooveless O-Ring Vacuum Seal for Bell Jars and Other Demountable Components
Rev. Sci. Instr., V.26, p. 1207, 1955

A seperate retaining plate for supporting an O-ring at the bell jar seal is described which did not enter into the vacuum seal but served as a backing to keep the O-ring from being pushed into the vacuum system while the proper O-ring compression is maintained. Use of this modification did away with machined retaining grooves.

W-062
Willis, W.W.
An All-Metal Sealing Joint
Missile Design and Development, V.4, pp. 18-19, August, 1958

Describes Marmen Conoseal joint and gives some test data indicative of its performance.

W-063
Wilson, R.R.
Vacuum Tight Sliding Seal
Rev. Scientific Instruments, V.12, pp. 91-93, Feb., 1941

The operation and construction of a vacuum-tight sliding seal by means of which unlimited translational and rotational motion in vacuum can be obtained is described. The seal is made by an appropriately distorted rubber gasket and a table has been prepared which gives the dimensions of the gaskets to be used for various sizes of sliding rods. Drawings included.

W-064
Wilson, W.H.
Blind Gasket
Heating & Piping, V.1, p. 85, May, 1929

The blind gasket, its description and application. It is common practice to cut it out of steel plate with an oxy-acetylene torch. The part projecting from the disc, or handle, must be narrow enough to pass between the bolts, and readily identifies the presence of blind gasket

W-065
Windenberg, D.F.
Master Charts for the Design of Vessels Under External Pressures
PB L 85991, Off.Tech.Serv., May, 1947 (Reprint from Trans.of ASME, pp.345-351, May, 1947)

This paper presents the theoretical equations underlying non-dimensional charts suitable for the design of circular cylindrical steel vessels under external pressure. The methods used in constructing the chart and their advantages and limitations are discussed. It is shown also how the chart can be used for the design of pressure vessels constructed of non-ferrous and other materials which have non-linear compressive stress-strain curves.

W-066
Wingrove, T.R.
Packing a Compressor Piston
Ice & Refrig., V.27, pp. 4-5, July, 1904

Discusses the wrong and right way of packing a rod.

W-067
Winkhaus, A.
Steam Loss in Labyrinth Packings (In German)
Zeit. des Ver. Deut. Ing., V.66, pp.804-807, Aug. 26, 1922

Discussions based on tests.

W-068
Wirta, R.W.
Gold Gaskets for Process Piping
G.E.Co., Hanford Atomic Products Operation, Richland, Wash. Contract W-31-109-eng-52, HW-48480 (5 pp.), Feb. 14, 1957

Gold exhibits excellent physical gasketing properties on a 1-inch flange with pressures less than 80 psi. Not suitable for use with stainless steel.

W-069
Wisander, D.W. and others
Wear and Friction of Filled Polytetrafluoroethylene Compositions in Liquid Nitrogen
ASLE Trans., V.2, No.1, pp.58-66, April, 1959

Bearings and seals of missile power plants have to work under extreme conditions and it is necessary to investigate materials at very low temperatures for this purpose. Described are results of tests on PTFE materials of molded form and extruded form with different fillers including carbon, graphite, glass, metal and ceramic in concentrations up to 25%.

W-070
Wisander, D.W., Hady, W.F., Johnson, R.L.
Friction Studies of Various Materials in Liquid Nitrogen
NACA TN 4211, 1958

W-071
Wisander, D.W., and Johnson, R.L.
Wear and Friction of Impregnated Carbon Seal Materials in Liquid Nitrogen and Hydrogen
N-85890, 1960 Cryogenic Eng.Conf., Boulder, Colo. Advances in Cryogenic Engng., V.6, pp.210-218, 1960

Experimental wear and friction studies were conducted with a series of impregnated carbon materials sliding against metal surfaces in liquid nitrogen (-320^oF) and in liquid hydrogen (-423^oF) at sliding velocities to 12,000 f/min. These mechanical carbons have potential use in dynamic seals for these fluids.
Data are presented for a basic carbon grade, commercial impregnated carbons, various experimental impregnants of basic mechanical carbons, and commercial plastic matrix bodies.

W-072 B
Wizeman, K.C.
Metallic Packing: Best for Gases
Power, V.102, No.8, pp.120-2, August, 1958

A description of metallic packings in reciprocating compressors, installation, lubrication, packing material, the radially slit ring, split case, metallic scraper rings.

W-073
Wolf, J.E. and Connelly, R.E.
Development of Seals for Rocket Engine Turbopumps
ASLE Trans., V.2, No.1, pp.25-31, 1959

Principle of liquid rocket engine with turbine driven pumps and sealing problems involved; operating conditions of typical turbopump seals and allowable static leakage rates; development of oxidizer, fuel, impeller, hot gas, and oil seals; seal installation, testing and development.

W-074 T
Wolfson, M.R.
Shoe Design and Material Studies for High Speed Sliding Friction Applications at Snort
Navord, Report 5626, ASTIA Document No.AD-206332, Aug. 27, 1958

Investigation of friction, wear, lubrication, material selection, and component design for sledshoes used at the Supersonic Naval Ordnance Research Track. A mechanism of wear of unlubricated rubbing surfaces postulates the formation of a molten film at the contact area. Some limited data confirms the validity of the theory.

W-075
Wood, D.F.
Rubber Boot Gas Seal Properties, Tests
Knolls Atomic Power Lab., Schenectady, N.Y., KAPL-M-EDL-19 Contract W-31-109-eng-52, 68 pp., June 1, 1953

A series of tests was conducted for the purpose of evaluating the properties of the rubber boot seal used on the control drive shaft of a submarine.

W-076
Wood, D.F.
Rotating Plug Rotating Seal Test
Knolls Atomic Power Lab., Schenectady, N.Y., KAPL-M-EDL-120 Contract W-31-109-eng-52, (35 pp.), Aug. 23, 1956

An investigation was conducted for the purpose of developing a seal and determining the proper seal lubricant to control the gas leakage from the S1G reactor during operation of the rotating plunger. The flow of diffusion gas, i.e., seal leakage, was in the order of 0.001 standard cubic feet per hour of helium at 10 psig. Plastilube No.1 proved the most satisfactory seal lubricant.

W-077
Wood, D.F.
O-Ring Tests; S1G/S2G
Knolls Atomic Power Lab., Schenectady, N.Y., KAPL-M-EDL-127 Contract W-31-109-eng-52, (24 pp.), June 28, 1957

An investigation was conducted to determine the operating characteristics and limits and the optimum use conditions of O-ring seal on the S1G/S2G control plug shaft. Temperature range of 70^oF to 150^oF with a pressure differential of 40 psig.

W-078
Wood, F.W., Jr
5 Important Gasket Materials
Prod.Engng., V.30, pp.93-95, Dec. 7, 1959

Quick-reference facts to aid selection among paste, felt, cork, rubber, and metallic seals.

W-079 T
Wo H.J.
Labyrinth Seal
U.S.Patent 2,781,210, February 12, 1957

A method of forming a labyrinth seal in a substance capable of plastic flow. A plate on the housing has a bore lined with a substance such as lead. The inwide diameter of the lining is slightly smaller than the outside diameter of a circumferentially grooved collar which is secured to the shaft. When the seal plate is forced over the shaft collar, and the shaft is subsequently rotated, frictional heating caused a plastic flow of lining into the collar grooves, and forms a labyrinth seal.

W-080
Wood, T.D.S.
Packings in PTFE Tolerate Rough Shafts
Engineering, V.188, p.387, Oct. 23, 1959

Chevron rings made of polytetrafluoroethylene for packing the glands of high-pressure valves. Advantages of these rings are low friction on valve spindles, chemical inertness, and stability to 300^oC. Tests were made to determine the influence of shaft surface finish on the wear rate of the rings, and to compare the performance of machined and molded rings. It was found that shaft roughness has no effect on wear rate, except that initial friction and leakage were high with very rough shafts. Also molded rings wore less than machined rings.

W-081
Wood, T.H.
Mechanical Shaft Seals in Chemical Industry
Trans.Instr.Chem.Engrs.(London), V.32, pp.73-80, 1954

The design, operation, and materials of construction of mechanical seals for both balanced and unbalanced types are presented. Working pressures vary from 2×10^{-4} mm of Hg to 600 psi, and temperatures from -90^oF to 600^oF at speeds up to a maximum of 6000 fpm.

W-082 B
Woodhouse, H.
Centrifugal Pump Packings and Seals, Part I, Packings and Glands
Petrol.Refin., V.36, No.1, pp.163-6, Jan., 1957

The first article is devoted to construction features and application of various types of stuffing boxes and glands. The author describes how seals work and shows why different types are required in different cases.

W-083 B
Woodhouse, H.
Centrifugal Pump Packings and Seals, Part II, Packings
Petrol.Ref., V.36, No.2, pp.173-7, Feb., 1957

This article indicates the various types of packing that are used in stuffing boxes and contains a comprehensive guide for the selection of the correct packings for the fluid being pumped.

W-084 B
Woodhouse, H.
Centrifugal Pump Packings and Seals, Part III, Mechanical Seals
Petrol.Refin., V.4, pp.207-11, April, 1957

The third article discusses mechanical seals, where they should be used, what types are available, and how they should be handled.

W-085 B
Woodhouse, H.
Centrifugal Pump Packings and Seals, Part IV, Mechanical Seal Maintenance
Petrol. Refin., V.5, pp.243-6, May, 1957

Starting with a warning that mechanical seals must be handled and fitted with great care, the author demonstrates how to trace cause of seal troubles, discusses where leakage is likely to occur, and then gives appropriate remedies for each trouble.

W-086 B
Woodhouse, H.
Centrifugal Pump Packings and Seals, Part V, Packing Maintenance
Petrol.Refin., V.6, pp.193-7, June, 1957

In the last article in this series, the author attributes the majority of stuffing box packing troubles to faulty maintenance. The correct procedure for repacking and maintaining the gland is described, and a sensible trouble-shooting chart is given.

W-087
Woodhouse, H.
Know Facts About Pump Packings and Seals
Pipe Line Industry, V.6, No.6, pp. 55-58, June, 1957; V.7, No.1, pp. 56-59, July; No. 2, pp. 55-59, August; No. 3, pp.44-47, Sept.; No. 4, pp. 58-62, Oct., 1957

Construction features, installation, and maintenance of packing boxes and glands, packings, mechanical seals, mechanical seal maintenance and packing maintenance for centrifugal pumps used on large pipe line systems.

W-088 T
Woodrow, J.
Viscosity Plates- Flow and Loading
Atomic Energy Research Establishment, E/M (6 pp.)
Dec. 22, 1949

The radially inward flow between annular viscosity plates is investigated by methods of lubrication theory. The flow rates and pressure distribution are obtained for isothermal flow of gas and for incompressible fluid. The resultant forces on the plate due to the radial flow are worked out for the same two cases. (Author)

W-089 T
Wordsworth, D.V.
The Viscosity Plate Thrust Bearing
Atomic Energy Research Establishment, E/R 2217, October, 1952

Part I reconsidered the theory presented by Whipple, indicating that the load carrying capacity of the viscosity plate thrust bearing increased approximately linearly with speed. Considering aerodynamic effects, substantially agree with Whipple. Part II outlines the study of other possible causes of experimentally observed deviations.

W-090
Wright, E.F.
Reciprocating Pump; Rod, Piston and Plunger Packing
Power, V.91, pp. 346-8, May, 1947

Selected questions and answers that give practical understanding of construction, operation, characteristics, troubles and remedies (17 total). Photographs and drawings included.

W-091 T
Wright, R.
Mechanical Seals Eliminate Leakage and Simplify Handling of Liquids
Ind. and Engng. Chem., V.42, Supplement pp. 97A-98A, Aug., 1950

A brief descriptive discussion of mechanical face seals and their relation to design is presented, and some unique designs are shown.

W-092
Wroble, W.R.
Designing Hydraulic Cylinders for High Pressures
Mach., V.57, pp. 149-152, Sept., 1950

Outlines a method for applying simple formulas and new sealing principles to high-pressure cylinder designs.

Y-001
Yarwood, J.
Isolation Valves for Vacuum Systems
Vacuum, V.3, pp.398-410, October, 1953

The chief types of vacuum valves are classified and reviewed. Requirements of vacuum valve are reviewed. Development of new and special service valves described. Valves with rubber seals, valves with silver chloride seals, valves with metal-to-metal seals, valves for ultra-high vacuum. Data, drawings, and photos. References.

Y-002
Yarwood, J.
Symposium on Some Aspects of Vacuum Science and Technology- London, January, 1962
Brit. J. Appl. Physics, V.13, No.7, pp. 301-5, July, 1962

1) Introduction; 2) Cryogenic pumping; 3) Molecular sieve pumping; 4) Ultra high vacuum; 5) Seals and gaskets: wire gasket, the welded seal, Parkas seal with coolant channel, knife-edge gaskets; 6) Mass spectrometry; 7) Leak detector.
Gold wire, stainless steel, aluminum wire, Viton, neoprene, diamond-shaped copper, etc.
Data and drawings.

Y-003
Young, E.L.
Air-drying System for EBWR Turbine Seals
Nucleonics, V.15, No.4, pp. 105-106, April, 1957

How drying and moisture recovery system applied to Experimental Boiling Water Reactor steam plant at Argonne National Laboratory will prevent loss of valuable moderator and escape of radioactivity to atmosphere; system is automatic and remote controlled; schematic diagram of type of seal developed for 5000-kw. turbine, and of drying and moisture recovery system.

Y-004
Young, J.R.
Vacuum Limits of Rubber O-Ring Joints
Rev.Sci. Instrum., V.29, No.9, pp. 795-6 (Notes Section), Sept., 1958

In this note, the author indicates that the limiting pressure in many systems having neoprene rubber O-rings is due to the liberation of butane from the rubber.
(Rubber O-ring: 2×10^{-6} to 1×10^{-6} mm Hg for 3 days)
(Teflon O-ring: 5×10^{-8} mm Hg for 24 hours)

Y-005
Young, J.R.
Cleaning Techniques for Rubber O-Rings Used In Vacuum Systems
Rev.Sci. Instrum., V.30, No.4, p. 291 (Notes Section), April, 1959

Rubber O-rings which are properly cleaned or, if carefully handled not cleaned at all, do not limit the vacuua obtainable under many conditions.

Z-001
Zabriskie, W. and Sternlicht, B.
Labyrinth-Seal Leakage Analysis
Jour. Basic Engineering, ASME Trans., V. 81, pp. 332-340, Sept., 1959

Study of fluid-flow aspect of gas leakage through straight-through labyrinth seals of type commonly used in gas and steam turbines. Includes staggered and unstaggered seals of the axial type. Leakage calculation methods within 20% accuracy are presented.

Z-002
Zotov, V.A.
Research on Helical Groove Seals
Russian Engineering Journal (Translation of Vestnick Mashin (USSR), V.10, pp. 3-7, October, 1959)

After examining the three components of the flow past a helical groove or screw seal, the author shows that the ratio of the pressure rise to the product of the fluid viscosity and seal speed is equal to a function of six geometrical quantities. Some experimental results of a one-inch diameter shaft with speeds up to 10,000 rpm and pressures up to 710 lbs/sq.in. are presented.

SUBJECT INDEX

I. KINDS OF SEALS

KINDS OF SEALS, CONT'D

Gaskets - Piece of material which is clamped between static faces

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A-078 A-081 A-082 A-100 A-101 A-121 A-132 A-133
A-141 A-157 A-167 A-172 A-177 A-188 A-201 A-210
A-224 B-046 B-065 B-070 B-098 B-106 B-108 C-030
C-044 C-049 D-067 D-068 E-010 E-011 E-018 E-025
E-026 E-028 F-002 F-006 F-035 F-036 F-037 G-018
G-025 H-005 H-025 H-037 H-044 H-057 H-068 H-072
J-004 H-027 K-018 K-033 K-048 L-001 L-003 L-014
L-015 L-021 M-068 M-075 M-082 M-088 M-097 M-098
M-099 M-112 N-004 N-019 O-001 O-008 P-032 R-009
R-010 R-011 R-017 R-028 R-039 R-042 S-007 S-015
S-022 S-054 S-060 S-064 S-076 S-077 S-078 S-080
S-081 S-082 S-097 S-108 T-018 T-023 T-026 U-003
V-007 V-011 W-002 W-033 W-039 W-040 W-047 W-078

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A-078 A-088 A-221 G-004 M-082 P-012 R-045 W-026

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A-116 A-215 D-044 K-034 K-038 K-047 P-023 S-107
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H-081 L-038 L-045 R-042 S-046 S-094 W-038 Y-002

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A-008 A-221 A-241 F-013 H-019 H-020 M-011 M-082

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A-069 A-077 A-085 A-221 D-005 H-036 K-020 K-041
K-048 M-060 P-014 P-033 P-040 S-016 W-026 W-034
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A-008 A-105 B-047 C-017 D-065 K-048 P-012 R-017
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A-078 A-079 E-033 K-049 M-016 S-079 T-017

In Nuclear Plants

B-115 K-010 K-050

In Petroleum Industry

A-221 B-092 C-010 D-066 F-032 K-034 L-025 L-044

In Pressure Range 0 to 100 P.S.I.

A-085 A-134 D-056 M-067 M-107 P-014 S-016 S-107
W-068

In Pressure Range 0 to 5000 P.S.I.

A-204 B-002 I-012 K-048 V-009

In Pressure Range 0 to 10,000 P.S.I.

A-183 E-009 F-035 F-039 G-021 M-069

In Temperature Range -300 to $0^{\circ}F$

B-010 L-038 L-045 M-016 R-042 S-046

In Temperature Range -70 to $300^{\circ}F$

A-093 A-135 B-010 E-035 K-048 W-054

In Temperature Range 0 to $1000^{\circ}F$

A-204 B-002 B-010 D-056 G-021 H-060 M-016 R-047
R-064

In Vacuum Sealing

A-013 A-208 B-006 B-010 B-082 B-116 B-117 B-124
C-051 C-071 D-014 E-002 E-031 F-033 F-041 G-013
G-035 H-036 H-060 H-064 H-074 H-081 K-031 K-051
L-004 L-050 M-010 M-016 N-022 P-024 P-040 R-026
R-042 R-047 R-064 S-094 S-110 S-113 T-027 W-063
V-001 Y-002

In Valve Sealing

A-074 B-006 K-051 L-050 M-112 S-137 T-027

Hermetic Seal - A permanent seal by fusion or adhesion primarily for moisture and air-tightness

B-111 K-053 L-011

Labyrinth Seal - A device employing a tortuous leakage path, usually by fins on either the stationary or rotating member

A-007 A-018 A-030 A-036 A-059 A-063 A-073 A-104
A-112 A-117 A-129 A-194 A-198 A-226 B-007 B-023
B-034 B-055 B-058 B-084 B-112 B-120 B-129 C-038
C-075 C-076 D-020 D-055 D-069 E-007 E-032 F-001
F-034 F-045 G-016 G-017 G-043 H-033 H-059 H-067
J-013 K-008 K-009 K-011 K-036 K-044 L-009 L-047
M-018 M-019 M-046 M-093 M-102 P-013 P-021 P-044
P-056 R-044 R-063 S-012 S-024 S-038 S-088 S-093
S-101 S-120 S-128 T-037 T-050 T-053 T-054 V-004
V-005 V-006 W-009 W-013 W-020 W-049 W-067 W-079
Z-001

Lip or Radial Seal - Seal producing positive line contact circumferentially between the seal's inside diameter and a shaft or a rod

A-007 A-080 A-113 A-118 A-189 A-201 A-203 A-209
A-227 B-011 B-052 B-063 B-087 B-120 C-006 C-017
C-057 D-017 D-018 D-019 D-029 D-032 F-020 F-031
G-022 G-032 G-041 H-050 H-055 H-075 H-084 I-001
I-003 I-006 I-009 J-002 J-007 J-008 J-009 J-011
M-048 M-106 P-017 P-037 R-004 R-034 R-062 S-057
S-058 S-070 S-093 S-103 S-112 S-140 S-141 S-142
T-035 W-017 W-056

Magnetic Seal - A seal produced by applying a magnetic field to concentrate magnetic particles suspended in the fluid

A-174 A-192 A-218 A-220 L-029 W-007

KINDS OF SEALS, CONT'D

Packing or Stuffing Box - Deformable material compressed into a cavity surrounding a rod or a shaft

Static

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B-076	F-002	F-028	G-009	G-029	H-051	H-054	H-058
J-002	K-034	K-050	L-050	L-021	M-010	M-067	M-068
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W-034	W-039						

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B-074	B-109	B-127	B-129	C-002	C-026	C-031	C-032
C-066	D-023	D-025	D-028	D-030	D-031	D-039	D-040
D-047	D-050	D-051	D-052	D-063	E-005	E-019	E-025
E-026	F-002	F-019	F-022	F-029	G-022	G-037	G-039
H-008	H-014	H-016	H-039	H-040	H-042	H-051	H-052
H-063	H-068	H-070	I-013	J-012	H-018	J-021	K-006
K-032	K-033	K-036	K-037	L-020	L-021	L-034	L-040
M-001	M-006	M-018	M-024	M-028	M-031	M-033	M-038
M-045	M-068	M-089	M-094	M-102	M-105	N-001	N-008
N-009	N-018	O-002	O-009	P-018	P-036	R-001	R-031
R-036	R-037	S-009	S-010	S-032	S-037	S-042	S-065
S-069	S-088	S-095	S-129	T-019	T-020	T-040	T-051
T-052	T-053	T-056	W-004	W-018	W-032	W-039	W-046
W-052	W-067						

In Aircraft Sealing

A-090	A-152	C-013	C-025	D-021	G-036	H-024	K-022
M-026	M-059	M-072	M-073	S-066	S-106	T-038	

In Automobile Sealing

A-179	A-222	C-006	G-032	R-043

In Bearing Sealing

A-047	A-051	A-099	A-102	B-048	E-032	G-041	P-017

In Chemical Industry

A-056	A-061	A-128	B-027	B-071	B-075	C-027	C-063
D-043	D-046	G-001	H-029	K-034	K-046	L-042	P-052
P-053	S-011	S-044	S-111	S-136	T-016	T-048	

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T-048	W-066	W-072					

In Hose and Pipe Connections

A-053	C-003	F-016	H-024	W-034

In Torque Converters

G-032

KINDS OF SEALS, CONT'D

In Temperature Range 0 to 2000°F

A-097

In Hydraulic Equipment

A-026	A-039	A-060	A-090	A-091	A-099	A-105	A-143
A-150	A-152	A-176	A-179	A-217	B-008	B-105	C-013
C-024	C-025	C-034	C-060	C-061	C-078	D-004	D-021
D-049	D-058	D-059	D-060	E-015	F-018	F-025	F-028
G-032	G-036	H-023	H-024	H-038	H-056	H-083	J-010
J-032	K-022	L-032	M-020	M-025	M-026	M-027	M-034
M-051	M-059	M-072	M-096	N-021	P-029	P-038	R-007
R-017	R-020	S-028	S-066	S-067	S-070	S-071	S-089
S-090	S-106	S-117	S-118	S-135	T-002	T-038	T-060
W-034							

In Internal Combustion Engines

A-094	A-179	D-053	K-049	R-065	T-050

In Iron and Steel Industry

A-024	A-124	A-127	J-032	M-109	P-017

In Liquid Metal Sealing

G-014

In Munitions Sealing

F-025	R-007	S-106	W-037

In Naval Ships

A-070	H-002	M-061	S-014	T-018

In Nuclear Plant

C-028	J-019

In Petroleum Industry

A-047	A-108	A-203	B-048	D-027	F-005	K-034	M-029
P-030	P-037	P-038	P-049	S-061	S-114	T-016	W-012
W-082	W-087						

In Pneumatic Equipment

A-032	A-152	A-176	B-042	C-013	C-034	D-004	F-025
H-038	H-083	M-025	N-027	P-030	R-007	R-017	T-002
T-060							

In Pressure Range 0 to 100 P.S.I.

C-042	C-060	K-003	L-046	M-032	M-067

In Pressure Range 0 to 5000 P.S.I.

A-058	A-126	A-128	B-105	H-023	H-024	I-012	M-093
S-066							

In Pressure Range 0 to 10,000 P.S.I.

A-171	B-061	B-089	B-099	B-103	G-012	J-023	L-007
M-030	S-019	S-068	S-111				

In Pumps

A-026	A-032	A-042	A-076	A-087	A-089	A-094	A-108
A-122	A-140	A-217	B-061	B-075	B-133	B-134	C-001
C-003	C-028	C-042	C-048	C-065	D-011	D-043	D-045
E-015	F-003	G-001	G-014	H-029	H-065	H-066	I-002
J-001	J-023	K-003	K-004	K-005	K-026	L-037	L-041
L-042	L-046	M-020	M-076	M-109	N-021	N-023	P-041
P-052	P-053	P-057	R-065	S-031	S-043	S-061	S-114
T-016	T-024	T-025	T-038	T-048	W-003	W-012	W-023
W-082	W-083	W-084	W-085	W-086	W-087	W-090	

In Railway Vehicles

A-017	A-020	A-025	A-027	A-029	A-032	A-037	A-046
B-131	C-022	C-052	K-023	P-058	R-038		

In Steam Engines

A-026	A-034	D-048	H-002	H-009	J-022	M-066	U-003
W-012							

In Temperature Range -70 to 300°F

A-135	E-034	M-032	S-066	W-037

In Temperature Range 0 to 1000°F

A-075	E-001	H-083	K-046	M-030	M-093	S-019

In Gas and Steam Turbines

A-030	A-036	A-044	A-114	B-007	B-126	C-028	D-002
F-001	G-043	M-002	M-093	N-027	P-013	P-026	P-044
P-045	R-027	R-049	S-024	T-021	W-003	W-013	

In Vacuum Sealing

A-058	R-003	T-021

In Valve Sealing

A-017	A-026	A-027	A-028	A-067	A-074	A-090	A-122
A-125	A-232	B-039	B-071	C-003	C-028	C-052	C-060
C-078	D-011	F-007	H-002	J-001	M-093	P-058	R-043
W-080							

Ring Seal - A positive contact seal whose primary component is an independent ring-proportioned element

Felt, Quad, Delta, X, or other cross sections of ring seals

General

G-003

Dynamic

D-027	G-022	T-054	V-003	W-010	W-016	W-027

Static

E-003	E-024	F-039	H-041	W-016	W-062

"O" Ring - A ring seal with a circular cross section

A-005	A-015	A-116	A-139	A-145	A-148	A-180	A-182
A-187	A-197	A-201	A-214	A-218	A-222	A-225	C-012
C-019	C-035	C-039	C-040	C-041	C-043	D-029	D-042
D-063	E-003	E-004	E-012	E-013	E-034	E-035	G-005
G-008	G-015	G-022	H-011	H-030	H-057	H-083	J-002
J-016	J-027	J-029	L-029	M-028	M-037	M-049	M-052
M-055	M-056	M-075	M-100	M-108	N-026	O-001	P-010
P-047	R-001	R-053	S-015	S-034	S-035	T-020	T-029
T-030	T-031	T-044	T-045	T-046	V-003	W-006	W-025
W-054							

In Aircraft Sealing

A-151	A-187	A-190	B-064	C-025	C-039	D-021	F-048
G-003	H-024	M-059	M-072	M-073	P-009	P-047	R-018
R-053	S-125	S-126					

In Cryogenic Fluids

A-216	E-003	E-014	H-030	H-052	T-046	W-006

In Hydraulic Equipment

A-091	A-143	A-153	A-165	A-169	A-176	A-190	A-216
B-008	B-064	B-079	B-105	B-118	C-005	C-011	C-017
C-024	C-025	C-039	C-045	C-060	D-021	D-035	D-062
E-014	F-018	F-038	G-003	H-004	H-023	H-024	H-038
H-083	J-015	K-012	K-025	M-034	M-050	M-051	M-053
M-057	M-059	M-072	P-008	P-009	P-020	P-035	P-047
R-018	S-070	S-118	S-124	S-125	S-126	S-135	U-004
V-003	W-006	W-015	W-016	W-025			

In Launch Vehicles

A-236	E-003	E-014	J-025	R-018	V-008

In Nuclear Plants

H-004	K-010	P-001	W-077

In Petroleum Industry

D-027	D-035	F-048	J-028	P-030	S-099	T-049

In Pipe Connections

A-216	B-064	C-005	C-068	D-010	D-061	D-072	E-003
G-005	G-010	H-024	H-036	H-038	H-058	M-017	P-040
S-030	W-001						

In Pneumatic Equipment

A-160	A-176	B-064	C-036	D-072	F-038	H-038	H-083
P-030	W-006						

In Pressure Range 0 to 100 P.S.I.

A-015	B-090	C-023	C-042	C-060	P-001	T-049

KINDS OF SEALS, CONT'D

In Pressure Range 0 to 5000 P.S.I.

A-160	B-035	B-105	D-010	D-062	F-012	H-023	H-024
I-012	U-004						

In Pressure Range 0 to 10,000 P.S.I.

A-130	B-001	D-003	E-035	F-010	F-039	J-017	L-007
P-008	P-015	T-039	W-042				

In Pumps

A-163	C-042	D-003	M-004	S-030

In Space Ship Sealing

A-190	R-054

In Vacuum Sealing

A-013	B-015	D-061	G-010	G-013	H-030	H-036	H-058
H-064	K-002	K-051	K-052	L-043	M-017	M-090	M-108
P-040	R-032	R-041	S-041	S-045	S-072	V-002	W-001
W-006	W-061	Y-004	Y-005				

In Valve Sealing

A-163	B-035	C-060	K-051	M-004	M-017	M-050	S-030
S-045	S-137	V-002					

Piston Ring - A one piece ring seal with a discontinuous circumference

General

B-047

Dynamic

A-017	A-021	A-051	A-098	A-148	A-230	A-240	B-073
B-086	B-099	C-034	D-003	D-038	D-048	D-063	F-017
E-026	F-027	F-050	G-023	H-002	H-012	H-082	J-002
K-017	M-051	P-015	P-017	P-039	P-056	P-058	S-017
S-037	S-040	S-052	S-073	S-118	T-001	T-009	T-038
T-039	W-009	W-010	W-025				

Static

B-099	J-004

Segmental Rings - A ring seal composed of several butting or interlocking segments

A-035	A-114	A-201	A-226	B-020	B-097	B-120	B-121
C-046	C-075	D-027	D-058	J-021	K-027	P-032	R-018
R-027	S-038	S-040	T-011				

"U" Ring - A ring seal having a "U" shaped cross section

A-095	A-146	A-147	A-176	B-047	C-023	C-024	C-056
D-027	D-032	D-058	F-018	F-027	G-018	G-022	G-036
H-070	J-002	M-027	M-034	S-070	S-073	S-090	T-002

KINDS OF SEALS, CONT'D

"V" Ring - A ring seal having a "V" shaped cross section.

A-041	A-125	A-136	A-214	A-216	B-045	B-047	C-023
C-024	D-058	D-063	E-024	E-032	F-003	G-022	J-002
J-032	M-026	M-087	M-030	M-032	M-034	S-065	S-069
S-070	S-090	S-[illegible]	S-135	W-016	W-080		

Special Sealing Concepts (Self sealing fuel tanks, injection seals, ceramic to metal seals, etc.)

A-002	A-003	A-147	A-171	A-181	A-230	B-001	B-009
B-036	B-050	B-057	B-089	B-090	B-099	B-114	B-117
B-125	C-051	D-003	D-062	E-009	E-035	E-036	F-010
F-039	G-012	G-021	H-016	H-061	H-071	J-017	J-023
J-025	K-027	L-007	L-027	L-033	L-034	L-045	M-005
M-007	M-069	M-074	M-075	N-001	N-007	N-011	N-015
N-027	O-009	P-008	P-015	P-034	R-010	R-021	R-022
R-028	R-051	S-003	S-018	S-019	S-068	S-087	S-098
S-131	T-059	W-042	W-049	W-079			

Screw Seal - A device comprising a threaded shaft rotating in a close clearance plain bore (or plain shaft in threaded bore), sometimes called visco-seal or screw pump.

A-233	A-235	A-240	B-074	B-077	B-112	F-046	F-047
H-076	R-058	R-059	T-037	W-053	Z-002		

II. SPECIFIC INTEREST AREAS IN SEALING

Aging or Shelf Testing

A-012	A-134	A-149	A-154	C-031	E-011	H-082	K-025
M-057	N-026	S-081	T-046	W-006	W-033		

Back-up Rings - A washer type of reinforcement placed on the side of a packaging, ring seal, or similar seal opposite the sealing interface.

A-153	A-164	B-001	B-057	B-105	D-042	F-009	G-026
H-010	H-011	H-023	J-002	K-012	K-037	M-072	P-046
R-054	T-031	T-045	T-046	U-004	W-025	W-061	

Back Diffusion - in (Bushing, Screw, Labyrinth)

A-123	B-020	B-025	B-080	S-012	S-024

Balancing of Forces - The balancing of forces due to pressure on a face seal to reduce contact pressure at the sealing interface.

C-046	F-024	G-008	H-015	H-066	H-077	K-035	L-048
M-022	M-023	M-042	M-043	M-095	N-003	N-005	R-012
R-014	R-048	S-037	S-049	S-050	S-117	S-118	W-008
W-058	W-081	W-088					

Bellows - Used alone or as a component of another seal in static or limited motion application.

A-077	A-082	A-106	A-107	A-109	A-137	B-059	B-069
C-020	E-028	H-013	H-073	K-002	L-021	M-013	M-036
M-103	R-018	R-062	S-006	S-033	S-116	S-118	T-003
U-002	W-027						

Cavitation

B-095	B-100	D-013	N-010	V-010

Compatibility

Material - Fluid: The ability of a material to exist in intimate contact with a fluid without detrimental chemical, bacterial, radiation, or other effects.

A-009	A-040	A-049	A-097	A-103	A-126	A-127	A-158
A-192	A-206	A-215	B-031	B-052	B-067	B-071	C-001
C-027	C-029	C-041	D-053	D-068	E-015	E-034	G-022
H-018	H-020	H-069	H-075	H-082	J-027	K-033	K-035
K-047	L-014	L-039	M-024	M-041	M-082	M-088	M-101
M-109	M-113	O-004	P-012	P-037	P-045	P-053	P-056
R-017	S-002	S-013	S-021	S-052	S-081	S-107	S-108
S-141	S-142	T-046	W-016	W-026	W-054	W-080	

Material-Fluid-Material: the ability of relative motion to exist at the sealing interface in the presence of a seal material, fluid, and mating material.

A-098	A-150	A-170	A-184	A-185	B-013	B-033	B-093
B-094	C-011	C-077	D-004	F-048	G-002	G-004	J-015
K-025	K-046	L-017	M-106	P-038	R-033	S-025	S-043
S-044	S-071	S-092	S-112				

Chemical Properties

A-044	A-098	S-210	G-022	R-017	L-036	M-113	O-009
P-031	R-017	S-021	S-052	S-108	S-134	S-142	W-043
W-080							

Endurance Testing

A-006	A-007	A-047	A-149	B-006	B-097	B-115	C-009
C-033	C-038	C-065	D-017	D-034	G-019	G-036	H-029
I-013	J-009	J-027	K-033	L-007	L-022	L-032	M-023
M-042	M-051	M-077	N-014	O-006	P-037	S-095	S-114
T-031	T-049	T-060	V-002	W-008	W-037	W-043	

Faces - Effect of face surface contours or other face characteristics on the sealing capabilities.

A-174	A-231	B-101	C-054	F-022	G-011	G-020	H-015
L-022	L-033	M-091	N-003	N-024	S-004	S-039	T-057
W-058							

Flange Design

A-013	A-069	A-101	A-141	A-177	A-241	B-010	B-062
B-070	D-072	E-009	F-013	F-033	F-035	G-010	G-029
H-005	H-054	K-020	K-021	K-040	K-047	L-003	L-004
L-044	M-010	M-011	M-060	N-019	R-026	R-028	R-039
R-046	R-064	S-013	S-023	S-035	S-054	S-077	S-110
S-130	S-080	S-099	T-008	T-055	W-021	W-022	W-031
W-035	W-039	W-043					

Flange Serrations

A-101	B-010	C-066	F-033	K-041	M-010	S-110

Floating Types - sealing without directly fastening the seal to any other component. (excluding ring seals)

A-199	B-058	B-080	B-081	B-084	B-103	C-028	E-005
G-011	G-044	H-067	M-051	M-075	M-093	R-007	S-048
S-131	T-042						

Friction

General Discussion

A-058	A-072	A-091	A-098	A-125	A-139	A-146	A-149
A-156	A-208	B-054	B-118	B-124	C-005	C-017	C-023
C-025	C-041	C-080	D-018	D-022	D-026	D-032	D-035
D-036	F-031	G-008	G-022	G-023	G-025	G-041	H-055
H-057	H-062	J-003	M-037	M-059	M-064	M-101	R-023
S-053	S-075	S-141	S-142	T-002	T-013	T-032	T-038
V-010	W-009	W-011	W-017	W-020	W-070	W-079	W-080

In Face Seals

B-129	G-008	G-041	H-055	I-010	K-035	L-036	M-042
M-043	N-005						

In Lip Seals

B-129	W-009	W-020	W-079

SPECIFIC INTEREST AREAS IN SEALING, CONT'D

Friction, Cont'd

In Packings

A-046	A-058	A-060	A-061	A-072	A-091	A-125	A-237
B-041	B-129	C-025	D-023	D-053	G-022	G-041	L-032
M-059	T-002	T-025	T-038	T-040	T-052	T-056	W-018
W-032	W-080						

Friction, Torque or Power Loss Due to Friction

A-126	A-209	B-006	B-128	C 081	D-040	H-046	H-050
S-132	S-139	T-013	W-055				

Gas Sealing Applications

General

A-007	A-010	A-011	A-109	A-117	A-159	A-166	A-193
A-194	A-198	A-199	A-216	B-012	B-014	B-020	B-021
B-022	B-024	B-053	B-068	B-080	B-081	B-084	B-125
C-020	C-037	C-038	C-074	C-075	D-008	D-020	E-008
E-030	F-003	F-004	F-017	F-044	F-045	F-046	F-049
G-040	G-042	H-017	H-033	H-045	H-047	H-071	H-076
J-013	J-020	K-008	K-009	K-016	K-027	L-008	L-033
M-008	M-070	M-087	M-110	N-016	O-006	P-005	P-022
P-025	P-028	P-030	P-055	R-029	R-030	R-052	S-003
S-012	S-017	S-096	S-104	S-120	T-007	T-055	U-002
V-006	V-010	W-007	W-010	W-045	W-050	W-051	W-053
W-073	W-075	W-076	W-088	Z-001			

In Gaskets

A-049	A-079	A-172	E-010	G-035	M-082	M-107	N-016
T-017							

In "O" Rings

A-130	A-216	C-011	G-005	P-001	P-030	P-035	S-041

In Packings

A-097	A-179	A-198	A-217	B-061	D-045	D-046	F-003
J-019	K-046	M-020	M-031	M-104	N-016	P-030	R-031
S-011	S-042	S-111	W-072				

In Face Seals

A-198	A-217	A-231	B-013	B-059	C-038	C-054	C-073
C-075	D-001	L-033	M-014	M-062	O-005	P-073	S-011
S-115	S-116	W-053	W-057				

Heat Transfer - The ability of a seal to conduct heat and/or withstand thermal cycling.

A-103	A-149	B-075	C-076	D-006	D-018	D-039	D-053
E-001	G-031	H-077	K-007	K-015	L-028	M-043	M-082
M-084	M-096	S-086	S-052	S-079	T-033	W-055	W-058
W-079							

Hydrostatic Type

A-069	A-104	A-224	G-008	L-003	L-026	W-022	W-051

SPECIFIC INTEREST AREAS IN SEALING, CONT'D

Installation

A-009	A-021	A-118	A-176	B-042	B-107	C-011	C-041
C-050	C-065	D-011	D-048	D-060	E-017	E-020	E-021
F-018	F-023	G-002	G-007	G-024	H-041	H-068	L-022
L-041	L-042	M-002	M-049	N-008	P-023	P-057	S-007
S-015	S-056	S-057	S-065	S-084	S-085	S-088	S-095
S-103	S-108	T-003	W-016	W-041	W-049	W-072	W-073
W-087							

Leakage

General Discussion

A-062	A-178	E-008	H-022	J-031	E-007	G-003	L-010
L-049	M-046	M-047	M-049	R-057			

Dynamic Seal Leakage

In Lip Seals

A-007	B-119	D-017	D-019	D-032	H-055	J-009	S-140

In Face Seals

A-123	A-231	B-073	B-101	B-112	B-130	C-038	D-009
D-027	F-034	G-001	H-055	H-066	H-067	M-007	M-039
M-040	M-042	M-043	M-062	S-004	S-039	S-132	T-003
T-042	T-057	W-024	W-053	W-085	W-091		

In Labyrinth Seals

A-007	A-018	B-023	B-112	C-038	D-020	D-069	F-034
F-045	G-043	H-033	H-059	H-067	J-013	K-009	K-011
K-036	K-044	L-047	M-018	M-019	M-093	P-013	R-044
S-024	S-120	V-004	V-005	V-006	W-009	W-013	W-067
Z-001							

In Packing or Stuffing Box

A-018	A-060	A-102	A-107	A-232	A-237	B-126	B-133
B-134	C-061	D-025	D-027	D-039	E-005	F-019	F-028
G-001	G-043	H-014	H-038	H-066	I-012	I-013	K-022
K-036	L-007	L-037	M-018	M-072	M-073	M-093	N-027
P-013	P-030	P-057	R-003	R-031	S-024	S-042	S-114
T-021	T-025	T-052	W-013	W-039	W-067	W-080	W-085

In "O" Rings

A-163	B-118	D-027	D-035	F-038	I-012	L-007	M-072
M-073	P-030						

In Dynamic Seals in General

A-011	A-068	A-107	A-136	A-137	A-156	A-233	B-014
B-025	B-054	B-066	B-073	B-080	B-112	B-132	C-009
D-027	D-032	D-034	E-005	E-036	F-009	F-034	G-026
G-030	H-006	H-007	H-017	H-021	H-035	H-046	H-047
H-077	I-012	K-027	K-042	K-044	L-007	L-008	M-007
M-013	M-093	N-015	N-027	P-007	P-030	P-035	S-005
S-024	S-104	S-119	T-028	T-042	U-002	W-009	W-039
W-053	W-073	W-076	W-080				

SPECIFIC INTEREST AREAS IN SEALING, CONT'D

Leakage, Cont'd

Static Seal Leakage

A-004	A-156	B-009	B-049	B-062	B-118	C-016	C-049
E-029	F-028	F-042	H-038	H-071	K-001	K-010	M-062
M-092	M-107	N-015	N-019	S-064	S-086	V-001	W-039
W-043	W-047	W-053					

Leakage Test Techniques or Detection

A-014	E-033	H-034	H-064	K-010	K-036	P-027	S-064
Y-002							

Leakage by Molecular Diffusion or Permeation

B-043	B-077	D-008	D-014	H-020	J-030	J-031	L-050
M-033	P-001	P-055	R-015	R-057	S-081	W-076	

Leakage Testing Results

A-231	B-075	B-130	C-028	E-033	H-022	H-050	I-013
M-009	M-016	S-015	T-042				

Inverted Type - (Seal rotates with the shaft)

A-035	A-080	D-039

Loading and Sealing Characteristics - The installation or loading requirements and their influence upon seal life, wear, performance, etc. (Degree of compression tolerances, etc.)

A-006	A-033	A-069	A-223	B-039	B-118	B-124	C-005
C-081	D-040	E-024	E-028	E-034	F-039	H-060	I-013
K-037	K-038	M-031	M-053	M-055	P-023	R-039	R-061
S-034	S-036	S-054	S-078	S-082	S-139	V-009	W-041
W-058							

Lubrication

General

A-091	A-136	A-198	A-234	A-238	B-016	B-021	B-025
B-038	B-095	B-100	B-110	B-112	B-125	C-067	C-072
D-034	D-036	D-054	F-038	F-049	F-050	G-008	G-011
H-062	H-082	I-005	J-003	J-006	K-007	K-028	L-024
L-026	L-029	L-033	M-005	M-032	M-055	M-063	M-070
M-113	O-003	P-002	P-025	P-051	R-023	R-029	R-030
S-051	S-104	S-139	T-006	T-012	T-013	W-011	W-045
W-050	W-051	W-055	W-074	W-076	W-088		

In Lip Seals

B-063	F-031	J-007	J-008

In Face Seals

A-198	B-112	C-073	D-001	D-009	F-029	G-008	G-011
H-066	I-010	L-033	M-005	M-014	M-022	M-043	M-091
N-003	P-073	S-004	S-039	S-132	T-005	T-048	T-057

In Packings

A-091	A-198	C-063	C-065	F-029	H-066	M-006	M-032
M-104	P-041	S-095	T-048	W-004	W-072		

SPECIFIC INTEREST AREAS IN SEALING, CONT'D

Lubrication, Cont'd

Solid

A-028	A-052	A-053	A-085	A-098	A-239	B-035	B-041
B-061	B-093	C-067	E-001	G-020	G-034	H-026	H-048
K-001	L-036	M-013	M-041	N-018	N-020	O-004	P-016
P-043	R-008	R-062	S-061	S-138	T-001	U-001	

Lubricant Materials

A-119	A-184	A-239	B-055	B-056	B-110	B-125	C-077
D-034	D-036	E-014	F-009	F-031	G-004	G-028	H-010
H-066	H-080	R-065	K-012	K-015	K-016	K-022	K-025
K-035	L-017	L-029	L-042	M-095	M-006	M-032	M-042
M-058	M-065	P-028	P-037	P-039	R-060	S-025	S-033
S-071	T-013	V-012	W-076				

Magnetic

A-217	E-022	R-001	W-007	W-027

Manufacturing Methods

A-003	A-082	A-088	A-099	A-110	A-114	A-117	A-118
A-237	B-085	B-111	C-045	C-078	D-047	G-019	H-084
L-031	N-016	P-027	P-038	S-010			

Material Properties - (Mechanical and Physical)

of Elastomers (Buna-N, GR-S, Butyl, etc.)

A-169	B-043	B-083	C-031	D-053	E-010	F-048	K-012
M-106	N-020	P-037	S-141				

of Rubber (natural)

B-041	B-083	D-026	D-036	D-053	D-071	H-051	K-022
L-003	L-012	L-014	L-016	M-059	M-067	M-094	N-026
S-112	S-134	S-141					

of Rubber (synthetic)

A-093	A-134	A-143	A-154	A-158	A-168	A-169	A-185
B-083	D-071	H-020	L-014	L-016	M-097	M-098	O-004
P-031	S-124	W-054					

of Plastics (nylon)

A-046	B-083	C-031	M-106	P-039

of Metals

A-006	A-040	A-046	A-101	A-121	A-172	A-204	B-071
C-053	E-027	G-006	G-029	G-031	H-014	H-028	H-073
L-003	L-036	M-012	M-036	M-060	M-064	N-020	O-009
S-002	S-013	S-074	S-086	S-108	S-138	S-141	T-059
W-012	W-035	W-068					

SPECIFIC INTEREST AREAS IN SEALING, CONT'D

Material Properties -(Mechanical and Physical) Cont'd
of Other Materials

A-005	A-049	A-069	A-082	A-093	A-098	A-101	A-134
A-135	A-139	A-149	A-158	A-165	A-167	A-168	A-169
A-185	A-206	A-210	A-225	A-237	A-239	B-008	B-033
B-036	B-041	B-043	B-047	B-067	B-083	B-087	B-098
C-019	C-020	C-040	C-041	C-053	D-018	D-042	D-057
D-071	F-012	F-006	F-010	F-024	F-048	G-006	G-023
G-025	G-027	G-029	G-031	H-005	H-051	J-014	K-012
K-022	L-003	L-012	L-014	L-036	L-041	M-067	N-003
N-027	P-047	R-035	R-061	S-002	S-003	S-026	S-052
S-053	S-074	S-108	S-138	T-004	T-008	T-029	T-030
T-031	T-044	T-057	T-059	U-001	W-026	W-037	W-039
W-046	W-054	W-068					

Metallic

Gaskets

A-013	A-022	A-078	A-079	A-100	A-101	A-121	A-133
A-172	A-208	A-221	A-224	B-002	B-006	B-010	B-082
B-124	C-044	C-051	D-056	D-066	D-067	D-068	E-002
F-033	G-013	H-025	H-036	H-060	K-018	K-031	K-041
K-050	L-003	L-004	L-050	M-010	M-011	M-099	N-004
P-014	P-024	R-026	R-047	R-064	S-013	S-016	S-079
S-110	T-027	V-007	W-034	W-035	W-038	W-041	W-064
W-068	W-078	Y-002					

Piston Rings

D-048	P-015	P-058	S-037

Face Seals

A-174	B-028	H-079	M-103

General

A-004	A-013	A-025	A-052	A-058	A-067	A-097	A-103
A-130	A-162	A-174	A-178	A-202	A-207	A-236	B-028
B-040	B-071	B-075	B-083	C-007	C-051	C-052	C-080
D-007	D-046	D-048	D-072	E-027	F-008	F-019	F-026
F-028	G-009	G-013	H-014	H-022	H-031	H-032	H-036
H-038	H-054	H-073	J-012	J-023	J-025	K-006	K-023
K-040	K-050	L-010	L-040	L-049	L-050	M-010	M-020
M-045	M-066	M-071	M-074	M-103	M-108	N-007	N-011
N-013	O-002	P-015	P-058	R-036	R-037	R-040	R-053
S-037	S-041	S-063	S-127	S-129	V-002	W-003	W-012
W-015	W-020	W-029	W-034	W-043	W-052	W-062	W-072
Y-001							

Molding

A-131	A-239	A-169	A-020	A-125	B-008	B-094	G-029
C-035	C-045	C-013	D-071	E-001	E-026	F-030	J-027
M-059	M-106	M-112	N-017	P-036	P-037	P-038	P-041
S-007	S-097	S-126	S-138	T-001	W-069	W-080	

SPECIFIC INTEREST AREAS IN SEALING, CONT'D

Pressure above 5000 P.S.I.

A-040	A-130	A-147	A-171	A-183	A-230	B-001	B-036
B-050	B-057	B-089	B-099	B-103	D-003	E-009	E-035
F-010	F-039	G-012	G-021	H-079	J-017	J-023	L-007
L-016	M-030	M-069	N-015	P-008	P-015	S-019	S-068
S-111	T-039	W-042					

Pressure Transmission, Pressure Gradients

A-018	A-128	B-045	C-081	D-031	D-040	H-006	H-007
H-017	K-008	M-038	M-102	P-013	T-020	T-056	W-045
W-046	W-088						

Radiation Effects

A-188	B-053	B-098	B-106	H-004	K-015	N-020	T-049

Reciprocating Motion

General

A-058	A-072	A-083	A-147	A-148	A-217	A-220	B-028
B-029	B-030	B-050	B-073	B-088	B-117	B-119	C-034
C-057	D-003	D-027	D-032	D-034	D-064	F-003	F-011
G-028	I-009	I-012	J-004	J-022	J-023	J-024	K-044
M-048	N-027	R-052	S-089	S-109	T-011	T-058	U-003
W-009	W-029						

In "O" Rings

A-091	A-145	A-148	A-225	B-079	C-025	C-042	D-003
D-027	E-034	H-023	I-012	M-055	S-034	T-029	T-030
T-039	T-044						

In Packings

A-025	A-026	A-038	A-051	A-058	A-072	A-076	A-091
A-099	A-179	A-217	B-028	B-042	B-061	C-025	C-034
C-042	C-052	C-063	C-066	D-027	D-048	E-034	F-003
F-028	H-002	H-009	H-023	I-012	I-013	J-022	J-023
M-006	M-045	M-104	P-058	R-038	R-065	S-037	S-061
S-089	T-038	T-040	U-003	W-004	W-066	W-072	W-090

In Piston Rings

A-021	A-051	A-098	A-148	B-073	C-034	D-003	D-038
D-048	E-017	F-050	H-002	J-004	P-039	P-058	S-017
S-037	S-052	T-038	T-039	W-009			

Reliability and Service Life

A-032	A-038	A-049	A-085	A-122	A-125	A-128	A-195
C-032	G-039	H-020	H-037	H-063	K-025	M-062	M-077
M-084	M-109	N-010	R-002	S-052	T-035	W-043	

Selection

"O" Rings

A-201	A-005	A-160	[illegible]-018	H-038	J-029	M-034	M-056
S-015	S-135						

Lip Seals

A-201	D-017	G-041	M-048	P-037	S-103

Face Seals

A-201	B-129	C-001	D-015	G-041	H-080	M-103	M-022
M-062	P-036	R-006	S-101	S-115	S-116	S-026	S-136
S-084	S-085						

Packings

A-042	A-201	B-049	B-109	B-129	C-001	C-032	D-045
D-049	D-060	F-018	F-028	G-041	H-038	H-039	H-063
I-002	K-049	L-032	L-046	M-020	M-034	N-008	P-029
P-036	R-031	S-135	S-067	S-071	W-083		

General

A-012	A-064	A-119	A-137	A-149	A-166	A-172	A-201
A-210	B-060	B-070	B-081	B-108	B-122	B-129	C-010
D-038	D-065	D-067	E-020	E-029	F-009	F-013	F-018
F-035	F-037	I-008	K-016	K-047	K-049	L-011	M-015
M-034	M-103	N-019	P-016	P-022	P-023	P-037	P-051
R-025	S-015	S-038	S-080	S-096	S-101	S-102	S-135
T-034	W-026	W-040	W-041	W-074	W-078		

Shaft Eccentricity, Misalighment, Whirl, Vibration, Whip

A-005	A-009	A-038	A-051	A-136	B-016	B-069	B-072
C-063	D-008	D-018	D-025	D-035	H-050	H-077	I-005
J-019	M-036	M-062	M-091	N-005	N-012	S-040	S-048
S-104	T-005	W-007					

Shaft Surface Finish and Hardness

A-004	A-005	A-015	A-091	A-107	A-120	A-121	A-174
A-209	B-035	B-108	B-129	C-023	C-025	C-063	D-003
D-009	D-017	D-018	D-026	E-020	E-027	F-010	G-019
G-023	G-026	G-028	H-041	H-055	H-075	H-077	J-010
J-011	J-023	K-001	K-017	M-034	M-053	M-055	P-008
S-026	S-036	S-068	S-116	S-130	S-141	S-142	T-009
W-039	W-040	W-058	W-080				

Speed, High - (Greater than 4000 fpm)

A-009	A-166	A-212	B-005	B-030	B-091	B-097	C-046
C-054	E-021	E-022	F-034	F-047	G-027	G-034	M-003
M-023	M-052	M-053	O-005	P-022	R-031	R-034	R-048
S-033	S-053	S-058	S-060	S-075	S-122	T-003	T-010
W-051	W-055	W-071	W-074	W-081			

Spring Loaded Types

Face Seals

In Aircraft Sealing, Fuel Tanks, Airframes

B-013	H-079	S-115

in Launch Vehicle Sealing (Rockets, Missiles)

C-054	H-003

in Hydraulic Equipment Sealing

A-217	I-006	S-028	S-117	T-043

In vacuum Sealing

A-170	C-075	H-064	S-055	W-081

Lip Seals

B-120	F-020	G-032	G-041	H-055	H-084	I-006	R-062
S-093							

Segmental Rings

A-035	B-120	C-075	D-027	S-040

"O" Rings

B-079	B-090	C-041	D-027	D-062	G-008	H-064	K-012

General

A-037	A-041	A-055	A-068	A-094	A-096	A-102	A-107
A-112	A-127	A-128	A-137	A-140	A-198	A-217	A-230
A-240	B-069	B-072	B-073	B-074	B-090	B-112	B-120
C-001	C-002	C-038	C-048	C-066	C-075	D-027	D-043
D-044	D-062	E-001	E-020	F-005	F-028	F-029	F-034
G-001	G-011	G-032	G-037	G-041	H-064	H-066	H-067
J-022	K-005	L-020	L-021	L-033	M-005	M-007	M-013
M-033	N-027	P-036	P-049	P-054	R-036	R-062	S-011
S-028	S-040	S-050	S-073	S-093	S-101	S-110	S-117
S-131	S-136	T-016	T-021	W-053	W-084	W-085	W-087

Standards in Sealing

A-048	A-074	A-095	A-229	B-122	F-009	H-053	H-084
L-003	M-015	M-061	O-001	S-009	S-030	S-067	S-119
T-023	W-001						

Stresses in Sealing Elements

A-006	A-153	A-154	B-032	B-033	B-035	B-045	E-003
E-011	F-048	H-055	K-021	K-037	L-014	L-038	M-041
M-056	M-106	N-020	N-026	O-001	P-014	R-023	R-061
S-003	S-081	S-082	S-099	S-139	T-020	T-059	V-009
W-022	W-040	W-041	W-058				

SPECIFIC INTEREST AREAS IN SEALING, CONT'D

Suspended Solids and Abrasives in Fluids

A-217	A-220	F-029	R-001	J-011	L-036	N-023	N-024
N-025	P-053	R-001	W-005	W-007	W-009	W-058	

Temperature Below -65°F

A-015	A-135	A-191	A-216	B-010	B-036	B-059	B-124
C-029	E-034	E-035	H-030	H-032	H-081	I-011	K-046
K-048	K-052	L-017	L-018	L-038	L-045	M-013	M-101
P-028	R-042	S-012	S-033	S-046	S-094	S-116	T-046
W-006	W-030	W-038	W-054	W-070	W-071	W-081	

Temperature Above +1000°F

A-015	A-097	A-133	A-202	A-204	A-205	A-207	B-057
B-097	B-124	B-125	C-007	C-053	E-024	F-017	G-027
G-031	H-060	H-082	M-003	M-016	N-020	O-006	S-053
S-116							

Testing, Specification, Evaluation, Performance

of Lip Seals

A-118	A-203	B-052	C-017	D-032	D-029	F-020	F-031
H-050	I-003	J-008	J-011	P-017	R-062	S-140	S-141

of Face Seals

A-231	B-072	B-074	B-097	C-053	C-054	C-038	D-001
D-015	E-028	F-015	G-034	G-034	I-010	L-033	L-036
M-014	M-023	M-039	M-091	M-095	N-003	N-024	O-005
P-017	R-048	R-062	S-018	S-021	S-026	S-055	S-084
S-132	T-043	W-024	W-057				

Packings or Stuffing Box

A-023	A-060	A-143	A-150	A-171	A-179	A-203	B-026
B-047	B-072	B-074	B-075	B-103	B-105	C-003	C-024
D-025	D-028	D-039	D-040	D-045	D-063	F-019	F-025
G-014	G-029	G-036	H-024	H-054	H-056	K-037	K-050
M-038	M-059	M-072	N-027	P-017	P-030	P-046	R-001
R-007	S-031	S-032	S-069	S-088	S-106	S-124	T-008
T-025	T-050	T-056	U-004	W-018	W-046	W-067	W-080

Gaskets

A-088	A-121	A-133	A-141	A-204	A-210	B-002	B-046
B-047	B-082	B-106	C-017	C-044	D-056	E-028	E-033
G-004	H-005	H-037	H-044	H-081	K-048	K-050	M-019
N-022	O-001	O-008	R-028	R-045	S-046	S-054	S-077
S-110	S-137	T-026	V-009				

"O" Rings

A-143	A-153	A-160	A-169	A-187	A-216	B-064	B-079
B-090	B-105	B-118	C-017	C-023	C-024	C-068	D-029
D-062	D-063	E-013	G-005	G-015	H-024	K-025	L-029
M-052	M-053	M-059	M-072	N-026	O-001	P-001	P-030
S-034	S-035	S-124	S-125	S-126	S-137	T-031	U-004
W-025	W-077						

SPECIFIC INTEREST AREAS IN SEALING, CONT'D

Testing, Specification, Evaluation, Performance, Cont'd

of Bushings

A-199	B-081	B-091	D-008	F-046	R-029	V-010

of Segmental Rings

A-226	B-097

of Freeze Seals

C-037	C-079	C-081	S-006	S-127

of Felt, Quad, Delta, X, or other Cross Sections of Rings

C-015	E-024	H-041	P-030	W-010	W-062

of "V" Rings

A-041	A-216	C-023	D-063	E-024	S-069

of Piston Rings

B-086	D-063	K-017	S-073	T-009	W-009	W-010	W-025

Labyrinth Seals

A-226	C-038	D-020	D-069	E-007	H-033	H-059	K-009
K-011	S-088	S-120	T-037	T-050	V-004	W-009	W-067

of Hydraulic Equipment, Sealing

A-149	A-134	A-158	A-161	A-164	A-216	B-047	B-051
B-056	B-085	C-007	C-008	C-024	C-058	C-059	D-062
F-027	G-002	G-019	G-036	H-022	H-078	J-010	K-017
L-006	M-012	M-021	M-044	M-084	M-085	P-003	R-024
S-023	W-025						

of Vacuum Seals

B-104	H-031	K-040	U-002	W-031	W-043	Y-001

of Valve Seals

B-051	C-081	H-021	M-021	S-006	T-022	U-008	W-080
Y-001							

Theoretical or Analytical Evaluation

of Labyrinth Seals

A-059	A-226	B-129	D-055	E-007	G-016	G-017	G-043
H-059	K-008	K-009	M-018	M-019	M-102	P-021	S-088
S-120	T-053	V-004	V-006	Z-001			

of Bushings

B-091	D-008	F-014	J-003	R-058	S-048	S-104	T-053

of Screw Seals

A-233	A-235	B-074	R-058	R-059	Z-002

SPECIFIC INTEREST AREAS IN SEALING, CONT'D
Theoretical or Analytical Evaluation, Cont'd
of Face Seals

B-074	B-101	B-128	B-129	D-009	F-024	H-055	I-006
L-019	L-033	L-048	M-039	N-005	S-004	T-005	

General

A-002	A-006	A-059	A-084	A-136	A-226	A-234	B-003
B-004	B-016	B-038	B-045	B-063	B-074	B-095	B-096
B-100	B-129	C-005	C-069	D-013	D-023	D-030	D-040
D-054	D-064	G-016	G-040	G-043	G-045	H-006	H-007
H-017	H-055	H-077	I-003	I-006	K-007	K-021	K-038
K-043	K-053	L-024	L-026	L-027	L-033	M-018	M-052
M-079	M-080	M-100	M- 02	N-011	N-015	N-019	O-003
P-002	P-008	P-020	P-034	P-055	R-021	R-022	R-023
R-037	S-003	S-005	S-032	S-068	S-083	S-088	T-004
T-006	T-012	T-014	T-019	T-020	T-053	T-056	T-058
W-011	W-022	W-041	W-043	W-045	W-047	W-050	W-065
W-074	W-088	W-089					

Thermal Effects

A-009	A-158	A-208	A-237	B-096	G-006	H-068	H-073
J-019	J-020	J-026	J-027	K-052	L-023	L-038	M-041
M-094	N-020	O-009	R-055	S-053	T-059	U-001	W-048

Wear and Wear Particles

A-120	A-238	B-013	B-126	C-046	E-017	E-022	G-023
H-014	H-077	H-082	J-011	J-020	L-023	L-029	M-006
M-040	M-062	M-081	N-023	P-034	P-045	S-061	S-112
T-001	T-010	T-060	W-007	W-009	W-080		

Wear Due to Rubbing

A-098	A-128	B-005	B-016	B-018	B-033	B-125	C-041
C-057	C-067	D-039	F-046	G-027	K-044	L-036	M-003
M-023	N-012	N-027	S-020	S-053	S-074	S-132	S-138
T-020	W-012	W-069	W-071	W-074			

III. SEALING MATERIALS

Asbestos and Asbestos Compositions

A-039	A-051	A-052	A-053	A-064	A-065	A-066	A-067
A-101	A-113	A-132	A-133	A-134	A-141	A-241	B-060
B-071	B-109	C-032	D-005	D-048	D-056	D-066	F-003
F-018	F-022	G-029	H-005	H-052	H-083	J-022	K-018
K-049	L-003	L-011	L-014	M-030	N-016	P-018	P-041
P-056	R-024	R-045	S-130	T-001	T-008	T-018	T-025
T-026	U-003	V-007	V-039	W-041			

Carbon and Graphites

A-028	A-036	A-041	A-044	A-052	A-053	A-063	A-064
A-070	A-082	A-098	A-104	A-108	A-114	A-158	A-174
A-239	B-005	B-007	B-028	B-030	B-035	B-041	B-052
B-058	B-061	B-093	B-109	B-110	B-128	B-129	C-067
C-075	D-001	E-001	E-026	F-024	G-006	G-007	G-020
G-034	H-048	H-082	J-020	K-001	K-027	K-035	K-046
K-049	L-036	M-002	M-013	M-041	M-081	M-091	M-111
N-018	P-016	P-043	P-053	P-056	R-008	R-027	R-062
S-024	S-038	S-061	S-073	S-138	T-001	T-011	T-021
U-001	W-029	W-059	W-069	W-071			

Cements, Lutings and Adhesives

A-001	A-188	A-216	B-067	B-106	B-123	K-052	P-006

Ceramics and Cermets

B-125	C-051	C-053	C-071	E-002	G-007	G-027	G-031
H-080	H-082	M-022	S-053	S-098	W-069		

Cork and Cork Compositions

A-051	A-064	A-113	A-134	A-141	A-241	B-046	C-032
E-026	F-036	H-019	H-051	K-018	K-049	L-014	P-056
T-026	V-007	W-078					

Felt

A-064	A-112	A-113	A-167	B-023	B-066	B-023	C-032
G-025	K-018	M-015	M-086	P-007	R-001	R-035	S-002
S-093	S-101	T-018	W-014	W-020	W-078		

Fibers

A-039	A-064	A-065	A-066	A-070	A-086	A-088	A-132
A-134	A-141	B-026	B-078	C-032	F-022	F-036	G-025
K-018	K-022	K-039	K-049	L-003	L-014	L-042	M-006
M-020	M-030	M-104	N-018	P-041	P-058	R-037	S-113
S-130	S-135	T-001	T-007	T-018	T-039	T-058	U-003
V-007	W-047						

Glass and Glass Fibers

A-002	A-003	B-052	B-125	D-010	D-041	D-061	E-002
F-041	G-009	G-025	G-035	H-074	H-080	K-045	L-045
M-[illegible]	M-022	M-090	R-032	S-008	S-043	S-072	S-074
8	T-001	T-007	T-059	W-028	W-069		

SEALING MATERIALS, CONT'D

Leathers

A-051	A-055	A-064	A-067	A-099	A-113	A-141	B-011
B-031	B-042	B-043	B-131	C-032	C-078	D-004	D-052
D-058	D-060	E-032	F-031	G-028	H-056	H-069	H-070
H-083	J-032	K-018	K-049	L-012	M-024	M-027	M-029
M-033	P-017	R-004	R-005	R-006	R-007	S-051	S-057
S-065	S-066	S-070	S-090	S-101	T-002	T-018	T-039
W-032							

Metallic Seals

Aluminum

A-064	A-071	A-078	A-121	A-208	A-221	B-124	C-051
E-027	F-033	H-054	H-060	H-071	H-072	H-081	K-049
M-016	P-041	R-064	S-062	S-079	S-094	W-002	Y-002

Brass

A-064	B-069	E-015	K-052	R-040

Bronze

A-064	C-042	E-032	K-049	M-022	P-015	R-036	T-009
T-040							

Ferrous Alloys

A-081	A-103	A-130	A-133	A-202	A-208	A-220	B-002
B-005	B-054	B-124	C-007	C-016	C-067	D-003	D-066
E-002	E-003	E-024	E-027	F-019	F-026	G-019	H-031
H-060	H-082	K-001	K-041	K-046	K-047	K-052	L-004
L-043	L-045	M-003	M-012	M-024	M-081	N-004	N-012
N-016	P-024	R-036	R-047	R-051	S-002	S-016	S-062
S-079	S-086	S-108	S-113	S-138	S-141	S-142	T-009
W-007	W-064	Y-002					

Metal Fiber Composites

A-019	A-028	A-040	A-052	A-066	A-070	A-101	A-133
A-204	A-205	A-207	B-071	D-066	F-022	H-028	P-041
R-007	R-045	S-002	S-074	S-086	S-130	V-007	

Plated Metal Coatings

A-015	A-130	A-192	B-002	B-100	C-067	E-024	J-020
M-081	S-016						

Other Metal and Alloys

Used in Face Seals

A-174	B-028	B-030	B-059	B-129	C-053	D-001	G-008
G-034	H-079	L-036	M-022	M-023	M-041	M-103	P-017
R-062	S-018	S-033	S-101	S-116	S-118	T-003	

SEALING MATERIALS, CONT'D

Metallic Seals, Cont'd

Other Metal and Alloys, cont'd

Used in Gaskets

A-013	A-022	A-064	A-065	A-066	A-078	A-079	A-085
A-100	A-101	A-172	A-224	B-002	B-006	B-010	B-082
B-124	C-044	C-051	C-071	D-056	D-067	D-068	E-002
E-006	E-031	F-033	G-013	G-035	H-025	H-036	H-072
H-074	H-081	J-004	K-018	K-031	K-048	K-049	K-050
K-051	L-001	L-003	L-038	L-045	L-050	M-010	M-011
M-016	M-060	M-099	M-107	N-004	P-014	P-024	P-040
R-026	R-045	R-047	R-064	S-013	S-110	S-113	S-130
T-018	T-027	V-001	V-007	W-034	W-035	W-038	W-041
W-068	W-078	Y-002					

Used in "O" Rings

A-013	A-015	A-236	B-079	C-025	D-010	D-062	E-014
F-018	G-008	G-013	H-036	H-038	H-083	J-002	J-025
K-051	K-052	L-029	M-017	M-108	P-015	P-040	P-032
R-041	R-053	S-041	S-072	S-118	U-002	W-015	

Used in Other Ring Seals

A-037	A-058	A-072	A-075	E-024	J-021	L-010	L-050
M-010	S-069	W-027	W-062				

Used in Packings

A-019	A-020	A-023	A-024	A-025	A-028	A-032	A-037
A-038	A-039	A-046	A-050	A-052	A-058	A-061	A-065
A-067	A-072	A-075	A-097	B-026	B-028	B-040	B-071
B-075	B-109	B-129	C-025	C-052	D-004	D-046	D-048
F-007	F-016	F-018	F-022	F-025	F-028	G-009	G-029
H-008	H-009	H-014	H-038	H-042	H-083	J-012	J-021
J-022	J-023	K-006	K-023	K-049	K-050	L-040	M-006
M-020	M-045	M-066	M-093	M-104	N-018	O-002	O-009
P-017	P-041	P-058	R-036	R-037	S-037	S-069	S-118
S-129	T-018	T-040	W-003	W-012	W-034	W-052	W-072

Used in Piston Rings

D-038	D-048	H-012	J-002	J-004	K-017	P-015	P-017
P-056	P-058	S-037	S-040	S-118	T-009		

Used in Other Seals

A-004	A-006	A-066	A-106	A-162	A-174	A-178	A-192
A-202	A-205	A-207	B-018	B-050	B-125	B-129	C-007
C-008	C-051	C-080	D-007	E-024	F-008	G-006	G-028
G-031	H-013	H-022	H-031	H-032	H-061	H-073	H-075
H-078	J-020	J-022	J-023	K-039	K-040	K-050	L-010
L-028	L-045	L-049	M-012	M-036	M-064	M-071	M-074
M-093	M-103	N-007	N-011	N-013	N-020	O-009	P-017
P-056	R-018	R-024	R-055	R-062	S-002	S-008	S-018
S-040	S-063	S-086	S-101	S-127	T-036	T-059	W-005
W-020	W-029	W-043	W-048	W-065	W-069	W-071	W-079
Y-001							

SEALING MATERIALS CONT'D

Organic, Synthetic Materials

Chloroprene (Neoprene)

A-075	A-076	A-080	A-082	A-161	A-241	B-057	B-083
B-100	B-104	B-106	C-056	C-077	D-014	D-071	E-011
E-013	E-031	F-025	H-026	H-027	H-044	H-058	J-031
K-012	K-025	K-037	L-017	L-038	M-058	N-022	P-030
P-041	P-046	S-123	Y-002	Y-004			

Elastomers (Buna-N, GR-S, Butyl, etc.)

In Face Seals

M-106	P-037	S-141

In Gaskets

A-001	A-081	A-215	B-115	E-010	E-011	J-027	K-033
N-016	P-040						

In "O" Rings

A-153	A-169	A-187	B-079	C-012	C-024	D-062	E-014
F-048	J-027	K-012	P-040	R-018	S-125	T-045	T-046
T-049							

In Packings

C-013	C-024	C-031	D-053	K-033	N-016	P-038	P-046
W-018							

In Other Seals

B-043	B-055	B-056	B-083	B-084	B-085	B-086	B-088
B-104	C-008	C-024	C-070	D-062	D-070	E-014	G-028
G-033	G-038	H-018	H-047	H-071	J-005	J-026	J-027
K-017	K-039	L-017	L-018	L-028	L-039	M-113	N-016
N-020	P-006	P-037	R-018	R-033	S-091	S-092	W-030
W-031							

Fluoro-Elastomers (Kel-F, Teflon, Viton)

In Gaskets

A-135	A-167	A-215	B-047	B-106	C-044	F-026	F-033
G-004	G-018	G-021	H-064	H-074	H-081	K-033	M-067
M-068	N-022	P-040	S-107	S-108	S-130	Y-002	

In Lip Seals

B-052	G-022	G-032	S-070

In "O" Rings

A-165	A-169	A-190	A-197	B-064	B-090	B-118	C-041
E-014	E-034	F-048	G-003	G-022	H-011	H-030	H-038
H-064	P-040	P-047	R-001	S-070	Y-004		

Organic, Synthetic Materials Cont'd

Fluoro-Elastomers (Kel-F, Teflon, Viton) Cont'd

In Other Seals

A-122	A-125	A-126	A-127	A-135	A-161	A-168	A-184
A-185	A-206	A-237	B-012	B-036	B-047	B-051	B-054
B-061	B-078	B-089	B-090	C-077	D-004	D-006	E-001
E-014	E-024	E-026	E-034	F-021	F-022	G-003	G-018
G-021	G-022	G-023	G-032	G-033	H-038	H-064	H-080
J-030	J-031	K-033	L-039	M-067	M-068	M-101	N-012
P-046	P-054	R-001	R-015	S-017	S-070	S-111	S-117
S-139	T-001	T-007	W-019	W-031	W-069	W-080	

Plastics (Nylon etc.) Used in

Gaskets

A-064	A-088	A-141	A-157	C-044	E-026	H-068	K-018
P-012	P-014	R-042	S-107				

"O" Rings

A-236	C-012	R-054

Lip Seals

M-106

Other Seals

A-028	A-029	A-039	A-046	A-052	A-108	A-124	A-131
A-159	B-012	B-078	B-083	C-004	C-024	C-027	C-031
C-077	E-026	H-068	L-028	M-041	N-012	P-039	P-041
S-008	S-062	S-069	S-117	T-007	T-036	W-031	W-071

Silicones

A-093	A-158	A-211	A-215	B-051	B-060	B-126	C-011
C-029	C-070	C-077	D-071	E-003	E-013	E-026	E-034
E-035	F-017	F-021	G-004	G-032	G-033	H-004	H-068
H-081	I-011	J-031	K-012	K-035	L-039	M-058	M-082
N-022	P-001	P-006	P-038	P-040	R-041	S-130	T-033
T-034	W-054						

Rubber, Natural Used in

Gaskets

A-033	A-064	A-065	A-074	A-085	A-141	A-208	A-241
B-116	B-117	C-044	D-014	D-056	E-031	F-032	F-033
F-039	F-041	G-013	H-037	J-027	K-018	L-003	L-014
M-067	M-082	P-040	S-130	T-018	U-003	V-007	V-011
W-033	W-063	W-078					

Lip Seals

B-052	J-002	R-004	S-057	S-058	S-070	S-093	S-112
S-141							

SEALING MATERIALS CONT'D

Rubber, Natural Used in

"O" Rings

A-222	A-236	B-064	C-024	C-045	D-035	D-071	F-039
G-003	G-013	J-002	J-017	J-027	K-002	K-025	K-052
M-059	N-026	P-040	R-001	S-070	S-118	T-039	Y-004
Y-005							

Packings

A-016	A-065	A-070	A-074	A-099	A-222	A-228	B-026
B-028	B-041	B-109	C-024	C-032	D-053	F-003	F-022
H-051	H-052	K-022	M-059	M-067	M-094	P-018	P-041
R-001	S-010	S-061	S-069	S-070	S-118	T-018	U-003

Other Seals

A-074	A-159	A-184	B-009	B-019	B-028	B-032	B-083
B-117	C-024	D-022	D-026	D-036	D-041	F-003	F-039
F-043	F-051	G-003	H-041	H-062	J-002	J-005	J-017
J-027	K-039	L-012	L-016	M-062	M-103	M-111	P-056
R-006	S-062	S-069	S-070	S-092	S-093	S-118	S-123
S-134	T-039	T-058	W-027	W-031	W-075	Y-001	

Rubber, Synthetic - (A specific synthetic compound is not stated) Used In

Gaskets

A-008	A-081	A-093	A-132	A-134	A-215	A-221	B-106
E-026	F-036	H-020	H-068	K-018	K-048	L-014	M-097
M-098	O-008	R-009	S-097	S-033	W-054		

Lip Seals

A-118	A-189	B-011	C-057	F-031	G-032	J-007	J-008
S-103							

"O" Rings

A-143	A-169	C-011	C-012	C-024	D-027	D-071	D-072
E-013	G-015	H-023	M-028	M-052	P-001	R-041	S-030
S-035	S-124	V-003	W-054				

Packings

A-143	C-013	C-024	D-004	D-027	D-050	D-051	D-052
E-026	G-032	H-023	H-068	M-028	M-030	M-032	R-003
S-071	S-124						

Other Seals

A-154	A-158	A-159	A-161	A-168	A-185	A-200	A-211
B-032	B-078	B-083	B-085	B-104	C-024	C-029	C-077
D-027	F-026	F-009	F-017	F-021	F-030	F-042	G-033
I-004	I-005	J-026	J-030	J-031	L-013	L-016	M-030
M-032	M-086	M-101	M-103	O-004	P-031	P-056	R-005
R-006	S-101	S-122	T-022	T-033	T-034	V-003	W-005
W-019	W-031	W-054					

IV. APPLICATIONS

Aircraft Jet Engines

B-067	B-097	C-046	G-034	S-073	T-010

Aircraft Sealing - (fuel tanks, airframes, canopies, etc.)

Face Seals

B-013	B-034	H-079	S-115

Packings

A-152	C-013	C-025	C-039	D-021	G-036	H-024	K-022
M-026	M-059	M-072	M-073	P-046	S-066	S-106	T-038

Gaskets

A-078	A-088	A-221	G-004	M-082	P-012	R-045	W-026

"O" Rings

A-151	A-187	A-190	B-064	C-025	C-039	D-021	F-048
G-003	H-024	M-059	M-072	M-073	P-009	P-047	R-018
R-053	S-125	S-126					

General

A-090	A-111	A-117	A-131	A-164	A-166	A-185	B-018
B-055	B-056	B-111	C-009	C-029	C-055	F-008	G-003
G-036	H-022	H-043	I-011	J-020	K-013	K-014	K-016
K-017	K-029	K-045	L-005	L-016	L-018	L-039	M-026
M-036	M-054	M-071	M-084	P-003	P-005	P-011	P-031
R-002	R-018	R-019	R-060	S-062	S-091	S-138	T-022
T-037	T-038	W-026	W-028				

Automobile Sealing

A-081	A-096	A-179	A-222	B-011	B-033	B-046	B-085
B-086	C-006	C-021	C-039	C-070	E-028	G-032	G-033
H-049	H-075	L-012	L-031	R-004	R-043	S-040	S-057
S-100	S-122	S-140	V-003				

Bearing Sealing

Face Seals

B-120	C-006	F-034	G-008	G-041	H-067	M-014	M-086
P-017	S-093						

Packings

A-047	A-051	A-099	A-102	B-048	C-006	E-032	G-041
P-017							

"O" Ring

G-008

APPLICATIONS, CONT'D

Bearing Sealing, Cont'd

General

A-048	A-051	A-057	A-068	A-112	A-113	A-129	A-136
A-173	A-189	B-016	B-023	B-025	B-066	B-069	B-096
B-114	B-120	B-121	B-122	C-006	C-033	E-032	F-021
F-023	F-034	G-041	G-042	H-048	H-049	H-053	H-067
J-009	K-042	L-009	M-008	M-015	M-070	P-007	P-017
P-025	P-056	R-025	R-030	S-040	S-048	S-093	S-096
S-100	S-102	S-103	S-128	W-020	W-051	W-056	

Chemical Industry Sealing

A-056	A-061	A-106	A-109	A-116	A-128	A-184	A-202
A-215	A-230	B-027	B-057	B-071	B-075	C-027	C-063
D-015	D-016	D-043	D-044	D-046	G-001	H-029	H-080
K-034	K-038	K-046	K-047	K-053	L-042	M-039	M-077
M-083	N-014	O-004	P-023	P-050	P-051	P-052	P-053
R-014	R-048	R-051	S-011	S-021	S-044	S-107	S-111
S-130	S-136	T-016	T-047	T-048	W-007	W-041	W-060
W-081	W-091						

Compressor, Sealing

A-076	A-098	A-106	A-109	A-137	A-193	A-194	A-198
A-199	B-084	C-074	C-075	D-045	F-003	G-034	J-013
M-063	M-078	M-104	O-007	P-042	S-027	S-036	S-049
S-050	S-052	T-048	V-006	W-009	W-029	W-059	W-066
W-072							

Cryogenic Fluid, Sealing

A-216	B-059	B-061	E-003	E-014	H-012	H-028	H-030
H-032	H-081	K-052	L-017	L-038	L-045	P-028	R-042
S-046	S-074	S-094	T-046	W-006	W-030	W-031	W-038
W-069	W-070	W-071	Y-002				

Engines, Internal Combustion and Diesel

A-078	A-079	A-094	A-179	A-187	B-119	D-053	E-013
E-033	F-044	H-041	H-084	K-049	L-031	N-010	M-016
P-031	R-065	S-017	S-027	S-052	S-079	T-017	T-050

Electric Machine, Sealing

A-102	A-119	A-195	A-217	B-023	C-072	C-073	G-005
G-042	K-007	K-039	M-014	M-063	M-082	P-026	W-007

Electrical Cable Entry Gland, Terminals, etc.

A-131	D-041	F-051	G-045	H-027

Electronic Tube Sealing

F-024	F-027

Equipment - (electrical, electronic, etc.)

A-008	A-221	A-241	B-111	C-020	F-013	H-019	H-020
M-011	M-082	M-108	W-024				

APPLICATIONS, CONT'D

Exclusion Sealing

A-056	A-068	A-102	A-112	A-214	B-066	B-069	B-070
E-032	I-009	K-042	L-008	M-048	M-086	S-096	S-100
S-102	S-103						

Hose and Pipe Connections and Quick Disconnects

A-006	A-053	A-069	A-077	A-085	A-115	A-164	A-175
A-191	A-216	A-221	B-049	B-064	C-003	C-004	C-005
C-014	C-015	C-016	C-037	C-055	C-058	C-059	C-064
A-068	C-072	D-005	D-010	D-041	D-061	D-072	E-003
E-029	F-004	F-016	G-005	G-010	G-019	G-029	H-024
H-036	H-038	H-054	H-058	J-014	K-020	K-030	K-041
K-048	M-017	M-047	M-060	P-014	P-033	P-040	R-002
R-015	R-016	R-060	S-008	S-016	S-030	T-008	T-055
W-001	W-026	W-034	W-035	W-036	W-048	W-064	W-068

Hydrogen-Cooled Generators and Motors, Sealing

B-020	B-022	B-024	G-008	H-067	K-028	M-014	P-055
P-073	R-029						

Hydraulic Equipment, Sealing

Packings

A-026	A-039	A-060	A-091	A-095	A-099	A-105	A-143
A-150	A-152	A-176	A-179	A-217	B-008	B-047	B-105
C-013	C-024	C-025	C-034	C-039	C-060	C-061	C-078
D-004	D-021	D-049	D-058	D-059	D-060	E-015	F-018
F-025	F-028	G-032	G-036	H-023	H-024	H-038	H-056
H-083	J-010	J-032	K-022	L-032	M-020	M-025	M-026
M-027	M-034	M-051	M-059	M-072	M-096	N-021	P-029
P-038	P-046	R-007	R-017	R-020	S-028	S-066	S-067
S-070	S-071	S-089	S-090	S-106	S-117	S-118	S-124
S-135	T-002	T-038	T-060	U-004	W-034		

Gaskets

A-008	A-105	B-047	C-017	D-065	K-048	P-012	R-017
S-089	W-026	W-034	W-035				

"O" Rings

A-091	A-143	A-153	A-165	A-169	A-176	A-190	A-216
B-008	B-064	B-079	B-105	B-118	C-005	C-011	C-017
C-024	C-025	C-039	C-045	C-060	D-021	D-035	D-062
E-014	F-018	F-038	G-003	H-004	H-023	H-024	H-038
H-083	J-015	K-012	K-025	M-034	M-050	M-051	M-053
M-057	M-059	M-072	P-008	P-009	P-019	P-020	P-035
P-047	R-018	S-070	S-118	S-124	S-125	S-126	S-135
U-004	V-003	W-006	W-015	W-016	W-025		

Face Seals

A-217	I-006	S-028	S-117	S-118	T-043

APPLICATIONS, CONT'D

Hydraulic Equipment, Sealing, Cont'd

"V" Ring, "U" Ring, Quad, Delta X, or Other Cross Sections of Rings

A-090	A-095	A-176	A-216	B-047	C-024	C-034	D-058
F-018	F-027	G-003	G-036	J-032	M-026	M-027	M-034
S-070	S-090	S-135	V-003	W-016			

Lip Seal

B-011	C-017	C-057	D-018	G-032	H-084	I-006	I-009
T-035	W-017						

General

A-012	A-014	A-086	A-096	A-111	A-149	A-154	A-158
A-161	A-162	A-164	A-168	A-184	B-017	B-029	B-031
B-035	B-051	B-053	B-055	B-056	B-078	B-085	C-004
C-007	C-008	C-021	C-055	C-058	C-059	C-070	C-072
C-077	D-033	D-038	D-062	E-014	E-036	G-002	G-019
G-033	H-001	H-006	H-007	H-022	H-062	H-078	J-010
K-013	K-014	K-015	K-016	K-017	L-005	L-006	L-016
L-027	L-049	M-012	M-021	M-044	M-051	M-054	M-058
M-071	M-084	M-085	M-101	P-003	P-008	P-011	P-054
R-018	R-019	R-024	R-060	S-023	S-118	S-121	S-134
T-002	T-009	T-032	T-038	V-012	W-025	W-026	W-092

Instruments, Precision - (Optical and Electrical)

A-214	B-012	E-004	R-042	W-019

Iron and Steel Industry, Sealing

A-024	A-040	A-124	A-127	A-209	D-033	F-020	J-032
K-039	K-042	M-109	P-017	R-033	S-121	W-056	

Launch Vehicle Sealing - (Rockets, Missiles)

A-175	A-178	A-236	B-019	B-044	B-051	B-059	C-034
D-007	E-003	E-014	H-003	J-025	K-013	K-017	L-039
M-036	M-106	P-004	P-028	R-002	R-018	V-008	W-062
W-069	W-073						

Liquid Metal Sealing

A-007	A-156	A-181	B-003	C-037	C-038	C-080	C-081
E-024	G-014	G-020	H-046	K-040	M-074	N-007	S-006
S-018	S-127	T-042					

Mixers and Agitators, Sealing

A-122	B-027	B-060	D-043	G-030	M-083	N-012	R-051
T-048							

Munitions Sealing

F-025	O-008	R-007	S-003	S-106	W-037

Naval Ships (Submarines)

A-070	A-078	A-228	C-037	C-058	E-013	H-002	H-037
M-061	R-025	T-018	S-014	W-075			

APPLICATIONS, CONT'D

Nuclear Plant, Sealing

A-007	A-156	A-193	A-194	A-195	A-199	A-211	A-213
A-223	B-080	B-081	B-084	B-115	C-028	C-037	C-050
G-020	G-030	H-004	H-035	H-045	H-046	H-047	H-076
J-019	K-001	K-010	K-040	K-050	M-008	M-014	M-046
M-087	P-001	R-052	S-006	S-012	S-039	S-063	S-127
W-051	W-057	W-075	S-076	W-077	Y-003		

Petroleum, Petrochemical Industries Sealing

A-047	A-048	A-055	A-068	A-082	A-083	A-108	A-110
A-118	A-144	A-159	A-185	A-203	A-206	A-221	B-025
B-048	B-049	B-062	B-092	B-110	B-119	C-010	C-054
C-056	D-001	D-015	D-027	D-035	D-066	F-005	F-009
F-020	F-023	F-030	F-031	F-032	F-034	F-048	H-049
H-050	H-055	H-069	H-080	H-084	I-003	I-008	J-007
J-008	J-028	K-021	K-034	L-008	L-017	L-019	L-025
L-031	L-044	M-029	M-086	M-113	N-017	P-007	P-030
P-037	P-038	P-049	R-005	R-012	S-051	S-061	S-096
S-099	S-100	S-102	S-103	S-114	S-122	S-140	T-004
T-015	T-016	T-022	T-049	V-012	W-008	W-012	W-073
W-082	W-087						

Pneumatic Equipment Sealing

A-010	A-032	A-098	A-106	A-132	A-160	A-164	A-176
A-194	A-199	A-239	B-031	B-042	B-051	B-064	B-068
B-081	B-084	C-008	C-009	C-013	C-034	C-036	C-046
C-074	C-075	D-004	D-072	F-025	F-038	H-083	K-016
K-029	K-048	M-025	M-054	M-071	M-087	N-027	O-006
P-030	P-048	R-007	R-017	S-073	S-104	T-002	T-060
W-006							

Pressure Range - 0 - 100 psi

A-009	A-015	A-085	A-095	A-134	A-186	A-212	B-053
B-056	B-090	B-130	C-023	C-042	C-054	C-060	D-056
E-021	E-029	F-004	H-076	H-079	K-003	K-042	L-010
L-046	L-048	M-032	M-044	M-067	M-107	O-005	O-006
O-007	P-001	P-014	P-043	R-034	R-055	S-012	S-016
S-033	S-053	S-063	S-107	S-116	S-132	T-009	T-010
T-028	T-041	T-049	T-057	W-068	W-081	Z-002	

Pressure Range - 0 - 5,000 psi

A-058	A-126	A-128	A-146	A-160	A-162	A-202	A-204
A-205	A-207	B-002	B-035	B-105	C-004	C-009	D-010
D-062	E-022	F-012	F-026	G-011	H-022	H-023	H-024
H-028	H-079	I-012	K-048	M-043	M-093	M-101	P-046
S-002	S-066	S-074	S-086	T-003	U-004	V-009	

Pressure Range - 0 - 10,000 psi

A-040	A-130	A-147	A-171	A-183	A-230	B-001	B-036
B-050	B-057	B-061	B-089	B-099	B-103	D-003	E-009
E-035	F-010	F-039	G-012	G-021	H-012	J-017	J-023
L-007	L-016	M-030	M-069	N-015	P-008	P-015	S-019
S-068	S-111	T-039	W-042				

Pump Sealing

Packings

A-026	A-032	A-042	A-076	A-087	A-089	A-094	A-108
A-122	A-140	A-217	B-061	B-075	B-133	B-134	C-001
C-003	C-028	C-042	C-048	C-065	D-011	D-043	D-045
F-015	F-003	G-001	G-014	H-029	H-065	H-066	I-002
J-001	J-023	K-003	K-005	K-026	L-037	L-041	L-042
L-046	M-020	M-076	M-109	N-021	N-023	P-041	P-052
P-053	P-057	R-065	S-031	S-043	S-061	S-114	T-016
T-024	T-025	T-038	T-048	W-003	W-012	W-023	W-082
W-083	W-084	W-085	W-086	W-087	W-090		

Face Seals

A-094	A-123	A-140	A-217	B-059	B-102	B-107	B-130
C-001	C-018	C-038	C-048	C-054	D-001	D-012	D-015
E-021	G-001	G-007	G-011	G-034	H-066	K-005	M-022
M-083	N-024	N-025	O-007	R-012	R-013	S-021	S-027
S-033	S-039	S-047	S-084	S-133	T-003	T-016	T-024
T-042	T-043	T-048	W-059	W-087			

General

A-007	A-103	A-106	A-109	A-142	A-156	A-163	A-185
A-195	A-200	A-206	A-213	A-233	A-235	B-003	B-037
B-051	B-052	C-055	C-072	C-079	C-081	D-003	F-036
F-047	H-084	I-001	K-004	K-039	L-002	L-026	M-004
M-013	M-065	M-077	M-079	M-080	N-002	N-014	P-028
P-048	P-051	R-022	R-058	R-059	R-063	S-030	S-052
S-127	T-015	T-022	T-028	M-020	M-021	M-022	W-005
W-073							

Railway Vehicle Sealing

A-017	A-020	A-025	A-027	A-029	A-032	A-037	A-046
A-049	B-131	C-022	C-039	C-052	K-023	M-075	M-112
F-058	S-119						

Steam Engines

A-026	A-034	A-098	D-048	H-002	H-009	J-022	M-066
U-003	W-012						

Temperature Range - 0 - 2000°F

A-015	A-097	A-133	A-202	B-053	B-057	B-097	C-053
G-027	G-031	J-002	S-053	W-043			

Temperature Range - 0 - 1000°F

A-075	A-186	A-191	A-204	A-205	A-207	B-002	B-005
B-010	B-110	B-125	C-007	C-008	C-009	D-056	E-001
F-004	F-017	G-021	G-029	H-011	H-012	H-028	H-031
E-060	H-079	H-082	H-083	K-046	L-010	L-018	M-003
M-012	M-016	M-023	M-030	M-044	M-093	O-006	P-056
R-016	R-024	R-047	R-055	R-064	S-002	S-019	S-033
S-074	S-086	S-116	T-003	T-007	T-010	T-011	W-081

Temperature Range - 70 to 300°F

A-093	A-135	A-184	B-010	B-050	B-056	C-029	C-040
E-029	E-034	E-035	F-026	H-079	I-011	J-002	K-048
M-013	M-032	M-058	M-101	P-003	P-046	P-047	R-024
R-034	R-053	S-012	S-033	S-066	T-042	W-025	W-037
W-054	W-081						

Temperature Range - 300 to 0°F

A-015	B-010	B-036	B-053	B-059	B-061	E-003	H-012
H-028	H-030	H-032	K-046	L-017	L-018	L-038	L-045
M-016	P-028	R-042	S-033	S-046	S-094	S-116	T-046
W-006	W-030	W-038	W-070	W-071			

Torque Converters, Fluid Couplings, Sealing

A-096	B-033	B-085	B-086	C-021	C-058	C-059	C-070
G-032	G-033	P-054	R-062	T-035			

Turbine Sealing (Steam and Gas)

A-030	A-033	A-036	A-041	A-044	A-063	A-073	A-104
A-114	A-117	A-144	A-166	A-193	A-194	A-240	B-007
B-013	B-024	B-059	B-126	C-016	C-028	C-046	C-049
C-072	D-002	D-020	E-016	F-001	F-045	G-017	G-043
H-017	H-035	J-013	J-020	K-011	L-009	M-002	M-014
M-063	M-093	N-027	O-006	P-005	P-013	P-021	P-022
P-026	P-044	P-045	R-027	R-049	R-055	S-005	S-024
S-036	S-115	S-120	S-128	S-138	T-021	T-054	V-006
W-003	W-007	W-013	W-073	Y-003			

Vacuum Sealing

A-004	A-196	A-219	A-233	B-009	B-054	B-104	B-132
C-076	D-006	D-041	D-061	E-027	F-042	F-043	F-051
H-026	H-027	H-031	H-032	H-034	J-005	J-030	J-031
K-024	K-040	M-064	M-074	M-108	N-012	N-020	R-003
R-021	R-022	R-034	R-040	R-042	R-047	R-057	R-064
S-008	S-045	S-058	S-096	S-109	T-027	T-036	T-047
T-055	U-002	V-001	W-006	W-007	W-027	W-031	W-043
Y-001							

in Face Seals

A-170	C-075	H-064	S-055	T-042	W-081

in Packings

A-058	G-009	H-054	T-021

in "O" Rings

A-013	B-015	G-010	G-013	H-030	H-036	H-058	H-064
K-002	K-051	K-052	L-043	M-017	M-090	P-040	R-032
R-041	S-041	S-072	V-002	W-001	W-061	Y-004	Y-005

in Gaskets

A-013	A-208	B-006	B-010	B-082	B-116	B-117	B-124
C-051	C-071	D-014	E-002	E-031	F-033	F-041	G-035
H-036	H-060	H-064	H-074	H-081	K-031	K-051	L-004
L-050	M-010	M-016	N-002	P-024	P-040	R-026	S-094
S-110	S-113	W-063	Y-002				

APPLICATIONS, CONT'D

Valve Sealing

A-012	A-017	A-026	A-027	A-028	A-067	A-074	A-090
A-122	A-125	A-163	A-232	B-006	B-035	B-039	B-051
B-071	B-119	C-003	C-020	C-028	C-052	C-055	C-057
C-060	C-072	C-078	C-081	D-011	E-029	E-036	F-007
H-002	H-021	J-001	K-016	K-024	K-051	L-050	M-004
M-017	M-021	M-050	M-093	M-106	M-112	N-011	P-058
R-043	S-005	S-006	S-030	S-045	S-134	S-137	T-022
T-027	T-055	U-002	V-002	W-080	Y-001		

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1. A simplified expression for the laminar sealing capability of the screw seal and a method of optimizing the screw geometry to obtain maximum sealing capability are derived.

2. The analysis is extended to the turbulent flow case, and an expression for the turbulent sealing equation, not considering leakage effects, is derived.

3. The results of an experimental investigation of the screw seal using water and silicone fluids, verify the form of, and evaluate the constants in, the derived turbulent sealing equation. Power loss measurements are also given.

4. Seal failure, observed in both laminar and turbulent flow, is analyzed in terms of appropriate dimensionless parameters, and different mechanisms of sealing failure postulated.

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Hughes Aircraft Company Final Report
NASA Contract NAS 8-2428 (N63-23045)

The most promising materials, Adiprene L-100, cured with castor oil, and a silicone sealant XE-1027 were modified to increase their ability to withstand stresses at liquid nitrogen temperature. Further evaluation indicated that the Adiprene formulation reinforced with glass fabric resists thermal shock, bending and vibration loads at 76°K and has a coefficient of thermal contraction approximately equal to that of aluminum. Silicone polymers were synthesized that may serve as bases for sealants having improved low temperature flexibility. A fixture was designed and used for the testing of sealants vibrated at cryogenic temperature. A sealant was formulated and evaluated for the sealing of crevices in cryogenic propellant dome tanks. A literature survey is included in the report.

Supp. 035
Logan, S. E.
Temperature Energized Static Seal for Liquid Hydrogen
Advances in Cryogenic Eng. 7:556-61 (1962)

The seal consists of an elastomer molded onto a ring of Invar. When the assembly is cooled, the flange of aluminum or stainless steel contracts more than the Invar ring. The differential contraction of the metallic compounds results in compressive stress in the sealing material, which increases as the temperature is lowered. Satisfactory performance of the seal was obtained for pressures as high as 1800 psi. Temperature and pressure cycling had no deteriorating effect.

Supp. 036
Anonymous
Ceramic to Metal Seals for High-Temperature Thermionic Converters
Bendix Corp., Red Bank Div.
Quarterly Technical Report No. 3 - Contract AF33(657)-10038 (AD410101)

Work completed has enabled fabrication of alumina-molybdenum structures which can resist a 1500°C environment. Ceramic refactory metal systems were identified which are stable and usable up to 2000°C. Recommendations are made for limitation of the development activities to the most promising joint systems.

Supp. 037
Brundige, E. L.; Hanks, G.S.
Ceramic-to-Metal Seals for High Temperature Operation
Los Alamos Scientific Lab
Contract W-7405-eng-36, 36 p., April 1963. (17-34366-LAMS-291)

Ceramic-to-metal seals for operations at temperatures up to 1500°C can be fabricated through the use of tungsten-metallized alumina bodies brazed to niobium-zirconium alloy components with pallachum braze metal. Seals for operation at 1000°C or lower can be made using pallachum-cobalt braze alloy, the braze metal can be added as a foil or by plating.

Supp. 038
Allen, C. Malcolm; Grieser, Daniel R.
A Study of Rotary Shaft-Sealing Concepts for Pressurized-Water Reactor Application
Battelle Memorial Institute; Quarterly Progress Report, Jan.-Mar. 1963
Contract W-7405-eng-92, 5 p., (EURAEC-621)(17-24614-BMI-X-10083)

Studies are being made on the operation and limitation of various concepts of rotary shaft sealing for pressurized-water reactor applications by means of an instrumented transparent model. A graphite stator was fitted with eight capacitance probes to permit simultaneous recordings on an oscillograph.

Supp. 039
Armstrong, D. C.; Blair, Norman
Large Rotary Vacuum Seal
Rev. Sci. Instr., Vol. 34, p. 440-1, Apr. 1963(17-20132-LAMS)

A seal is described which permits rotation of large circular sections or parts in the walls of a vacuum chamber. This seal is an unlubricated Tec-Ring (a Teflon O-ring with a core of rubber) and the friction is low even at large diameters. The behavior of such a seal is compared with that of a lubricated Neoprene or Silastic O-ring.

Supp. 040
Matt, R. J.
The Problems of Sealing Hydrogen Peroxide in a 3000°F Environment
J. Eng. Power Vol. 84, p. 341-4 (1962)

Seal wear life can be enhanced by proper material selection. Surface damage can be expected as a result of crevice corrosion, static pocket decomposition, direct chemical attack, or mechanical agitation. This problem may be alleviated by proper design of the seal cavity. Adequate blanketing and washing of critical surfaces are of prime importance. Sludging and coking of lubricants in the seal area are incompatible with standard seal designs. New techniques and configurations are needed for the seals because the lubricants will improve more slowly than the demand for working hardware.

Supp. 041
Balar, V.
Improvement of an Indium Valve
Cesk, Casopis, Fys 12 651-2 (1962)

Usual types of joints are not tight enough for high-vacuum use; a new improved type was designed in which glass is pressed into molten Indium. This produces a perfect seal on cooling.

Supp. 042
Rothbery, Samuel B.
Liquid Shaft Seals
Product Eng. V. 34, p. 85-7. Nov. 11, 1963

Centrifugal force keeps fluid in groove. Most centrifugal liquid-barrier seals depend on shaft rotation to keep the fluid spinning. Now there is a new way; a magnetic field pumps the fluid.

Supp. 043
Dodge, Louis
Labyrinth Shaft Seals
Product Eng. V. 34, p. 75-9. Aug. 19, 1963

Flow friction is all that keeps the fluid in when you have positive clearance seals. Here is your guide to how much leaks out. Design and leakage formulas for Sleeve Seals, Interlocking Labyrinth Seals, Non Interlocking Labyrinth Seals, and Tapered Teeth Labyrinth Seals, with recommended clearances.

Supp. 044
Hamm, H. W.
Frictionless Hydraulic Seals for Rotating Shafts
Chem. Eng. V. 71, p. 160. Feb. 17, 1964

A hydraulic seal consists of a disk rotating in a narrow housing that is filled with fluid. Through friction, the disk gives the fluid a rotational motion, and the centrifugal force throws it to the outside where it effects the seal.

Supp. 045
Anonymous
Profiting by Mechanical Seal Advances
Chem. Eng., Vol. 71, p. 83. March 2, 1964

At the Philadelphia meeting of the American Society of Mechanical Engineers an important session dealt with the selection, design, and operation of mechanical seals. Much of the discussion was particularly pertinent to the sealing problem in the chemical process industries. The session was in three parts, the first dealing with the materials presently available to the seal designs, the second with ways in which these materials can be used and the environment controlled so as to make the seal a success, and the third with selection and specification procedures.
The three functional categories of an end-face mechanical seal were defined as follows:

1. Primary seal components consisting of the rotating washer and the stationary seat against which it slides.
2. Secondary seals consisting of O-rings or other suitable gasketing.
3. Hardware consisting mainly of the spring and retainer.

Supp. 046
Anonymous
<u>Mechanical Seals</u>
The Engineer, Vol. 217, p. 148. Jan. 17, 1964

A mechanical seal made for handling many liquids, including hydrocarbons, water, certain acids, and alkalies at pressures up to 150 psi and temperatures up to 230°C. Static face of Stellite stainless steel with face ring of metallized carbon.

Supp. 047
Howard, F. L.
<u>Leak-Resistant Rotation Seal for Vacuum Application</u>
Eng. & Instrumentation V. 67 C N4, p. 335. Oct-Dec, 1963

Details of seal described; packing comprises system of Teflon cones with deformable ridge that acts as bearing surfaces.

Supp. 048
Ludtke, P. R.; Weitzel, D. H.
<u>Force and Seal Evaluation of Elastomeric O-Rings</u>
Advances in Cryogenic Eng., Vol. 8, p. 467-77. 1962

Measurement of force on elastomeric O-ring while it is compressed and held at constant thickness during cool-down from room temperature to 76°K; 0.140 inch thick by one inch ID O-ring is given arbitrary initial force of about 1400C pounds after compression, outer cover is soldered to base of test rig which accelerates relaxation of elastomer. After jig cools to room temperature, force lost during relaxation is measured. O-ring is then cooled to 76K over about 2 hours, best material tested is ASD compound IV-29B of cis-4 polybutacliene.

Supp. 049
Morgan, R. E.
<u>Effect of Environment Control on Mechanical Seal Success</u>
ASME Paper 63-WA-259 Meeting Nov. 17-22, 1963.

It is established by laboratory and field experience that mechanical seal success is most dependant on adequate fluid film on seal faces. Environment control is often necessary to provide and maintain this film. Number of design factors are discussed to demonstrate some of the ways to maintain favorable environment around seal.

Supp. 050
Wood, G. M.; Manfredi, D. V; Cygnor, J. E.
<u>Performance of Centrifugal Shaft Seals for High Temperature, High Pressure Liquids</u>
Machine Design, Vol. 36, p. 129. Jan. 30, 1964

Effective sealing of liquids under pressure in rotating hydraulic machinery has become vastly complex with the trend toward high temperature and the handling of exotic fluids such as liquid metals. Experience has shown that it is most desirable to seal against inert gas with face seals than it is to seal directly against high pressure liquid at high temperature. Such a sealing arrangement is feasible if a centrifugal dynamic seal, functioning as a primary seal, establishes an interface between liquid and inert gas on the rotating vane of the dynamic seal impellers. Mechanical face seals are then used as secondary seals to separate the bearing lubricant cavity from the liquid metal region by sealing against the inert gas used to pressure balance the dynamic seal.

Supp. 051
Smith, L. L.
<u>Fibrous Composite Materials for Extreme Environment Seals</u>
Lubrication Engineering, Vol. 20, N3, p. 99-105. March 1964

Research was performed on the development and evaluation of composite materials suitable for use as static and dynamic seals at temperatures ranging from cryogenic to 1500F and at pressures up to 5000 psi. An example of a composite in silver reinforced with stainless steel fibers. The steps followed in making composite seals are described. The result is a pore-free two-phase material with sufficient conformability to seal and sufficient strength to prevent extrusion under extreme temperature and pressure conditions.
Physical properties such as recovery and stress at deformation up to ten per cent, were determined for various composites at room temperature and 1000°F. The results show that the compositer have improved recovery and strength characteristics compared to those of corresponding pure fillers. Successful seal applications using composite materials include dynamic seals for rotating and reciprocating shafts and static seals.

Supp. 052
Fisher, Edward W.; Wilkinson, Donald H.
<u>Extending the Lip Type Seal Horizon</u>
Lubrication Engineering, Vol. 20, N3, P. 87-93. March 1964

This paper briefly references the two major short comings of elastomeric oil seals; fluid compatibility and temperature limitations. The role that polytetrafluoroethylene might play in resolving these problems is briefly mentioned. From here the development of a new oil seal is traced through the laboratory test stand to field usage. Critical areas are discussed and data presented showing progress in extending the horizon of the lip type seal. Some of the design problems involved are mentioned.

Supp. 053
Charhut, F. J.
<u>Shaft Oil Leaks Can be Stopped - Properly Applied Lip or Face Seals Keep Hydraulic Fluids in Place</u>
Plant Eng., Vol. 17, N11, p. 114-17. Nov. 1963

Objectives in design and application of lip and face type seals are discussed. Prevention of escape of fluid, entry of foreign matter from outside, and installation and removal problems are considered.

Supp. 054
Tankus, H.
<u>End Face Seals in Abrasive Service</u>
Lubrication Eng., Vol. 19, N10, p. 403-9. Oct. 1963

Abrasive particles in fluids classified as either inherent to fluid (slurries) or created by thermal effect or contact with atmosphere, depending on source and type of abrasive, various methods involving cooling, heating, centrifugal separation, and special seal designs are described to reduce or eliminate excessive seal wear in shafts.

Supp. 055
Hajner, Robert C.; Zanoni, Paul J.
<u>High Pressure Pump Seal Development Status of Tasks II, III, IV, V, VI, and VII</u>
Worthing Corp. Advanced Products Div. Contract AT(30-1)-(2682)(2683)
Engineering Monthly Progress Reports.
Engineering Quarterly Progress Reports.

Supp. 056
Faber, G; Heise, M.
<u>Use of Silver and Its Alloys for Seals in Oxygen Compressors</u>
Brown Boveri Rev. Vol. 50, N6-7, p. 430-34. June - July 1963

It is shown how addition of aluminum oxide to silver improves heat-resistant properties of metal, which is not affected by oxygen; comparative tests carried out with pure silver and silver-copper alloy, and silver hardened with Al_2O_3 showed superiority of hardened silver in temperature range from 200 to 300 C. (Labyrinth Seals)

Supp. 057
Thoma, J.
<u>Seal Gaps</u>
Hydraulic Pneumatic Power - Controls
Vol. 9, N105, p. 627-31. Sept. 1963

Sealing properties of gaps between moving or stationary surfaces including influence of variation of viscosity with pressure and temperature, arrangement being used in design of hydraulic components, e.g., pistons in cylinders, valve faces, etc., arrangement of hydrostatic slider fed through center passage with oil under pressure, which escapes across sealing gap is represented as example; determination of variable viscosity; solution of equations.

Supp. 058
Schoenherr, K.
<u>Materials in End-Face Mechanical Seals</u>
ASME Paper 63-WA-254 for Meeting Nov. 17-22, 1963

Problems encountered when selecting materials are discussed concerning seal operation and sealing elements; sliding faces are given detailed attention; factors causing malfunctioning and destruction of primary seal materials.

Supp. 059
Hudson, T. W.
<u>Inquiry Specifications for Mechanical Seals</u>
ASME Paper 63-WA-203 for Meeting Nov. 17-22, 1963

Approach to seal specification, which permits consideration of operation variables, is suggested and attention called to areas in which variables affecting seal performance exists.

Supp. 060
Conklin, E. W.
<u>End Face Seals in Light Hydrocarbon Liquids</u>
ASME Paper 63-PET-29 for Meeting Sept. 22-29, 1963

Centrifugal pumps have always required some means of containing fluids inside of mechanism; end face seals replace conventional stuffing boxes where fluid hydrocarbon must be contained in spite of substantial vapor-pressure head; these seals reduce friction-power losses, eliminate wear on shaft or shaft sleeve, are characterized by minimum or invisible leakage over long service life, relative insensitivity to shaft deflection or end play, and freedom from periodic maintenance.

Supp. 061
Alford, J. S.
<u>Protection of Labyrinth Seals from Flexural Vibrations</u>
ASME Paper 63-AHGT-9 for Meeting March 3-7, 1963

Resonance was identified in several designs of stationary seals which experienced failures; resonant conditions of stationary seal occurs when velocity of rotating seal is equal to velocity of propagation of wave in stationary seal. Self-excited vibrations cause fatigue failures where damping is very small; analysis of mechanics of self-excited vibrations discloses stability criterion which correlates pressure drop through seal with natural flexural frequency weight, and length; location of support of seal seems of importance in preventing self-excited vibration.

Supp. 062
Nau, B. S.
<u>Some Recent Results in Fluid Sealing Research</u>
Int. Fluid Power Conference Proc. 1962, p. 81-90

Review and analysis of experimental and theoretical investigations carried out on behavior of radial-face or mechanical seals operating under "normal" conditions. Relationship of friction coefficient, load, film thickness, and viscosity is considered in light of available experimental evidence and it is concluded that surface tension, surface waviness and seal misalignment or oscillation, may all be important determining factors in seal operation.

Supp. 063
Sence, L. H.
<u>Floating Ring Seal Design and Application - Special Reference to High Pressure</u>
ASME Paper 63-AHGT-31 for Meeting March 3-7, 1963

Consideration of several successful installations, test data and hydraulic principles, various curves show test results of single ring seal and group of three rings in series; other curves show test results of special floating-sleeve seal arrangement for nuclear reactor recirculating pump operating under pressure of 2000 psi with controlled leakage rates as low as 1/10 gpm, data presented indicate that operation of seal rings with flow having Reynolds number of over 1000 is unstable, and may lead to accelerated wear and possible failure.

Supp. 064
Lauzan, W. B.; Shelton, B. R.; Waldheger, R. A.
<u>Use of Carbon Graphite in Mechanical Seals</u>
Lubrication Eng., V.19, N.5, p. 201-9, May 1963

Results of broad test program to obtain data on performance of carbon-graphite seal materials in conditions of dry operation, liquid sealing, and air sealing in various environments including chroniate solutions, slurries, oil, and water. Conclusions regarding effect of seal pressures, resin treatment and selection of proper mating materials are pertinent to selection of carbon-graphite seals for commercial applications.

Supp. 065
Roy, A.; Hagen, F. A.; Belleau, C.
<u>Chrysler Gas Turbine Car Materials Requirements</u>
SAE Paper 777C for Meeting Jan. 13-17, 1964, 15 p.

Conventional and newly developed materials played an important role in developing Chrysler gas turbine engine: new low cost materials, developed by Chrysler Metallurgical Research, include -- series of irons has super alloys having equivalent or superior life to aircraft type alloys, heat resisting iron aluminum alloys for elevated temperature service under low stress application, and long-life rubbing seal materials that operate satisfactorily from ambient temperature to 1200°F and above, materials meet all present or immediate high temperature alloy requirements for competitive mass produced automotive turbine.

Supp. 066
Anonymous
<u>Composite Materials Self-Lubricating</u>
Sci. Lubrication, V. 15, N. 8, p. 22-24, Aug. 1963

Description of, and technical data on new metal (Cu and Ag) based on self-lubricating composities containing polytetrafluorothylene (film frames) and tungsten diselenide (dry lubricant); material is claimed to be able to carry loads as high as 1800 psi with little self-wear and negligible wear of surfaces against which they are sliding: application to cryogenic and high-vacuum conditions.

Supp. 067
Fulford, P. J.
<u>Performance of Mechanical Seals for Primary Coolant Pumps</u>
Atomic Energy of Canada, Ltd., Toronto, 15 p. (CONF-9-14)

The experience of AECL with high pressure pump-shaft seals is reviewed, and in this light important operating parameters and several variations in design are discussed. The requirements for such seals are outlined.

Supp. 068
Cottrell, W. B.,Ed.
<u>The Aircraft Reactor Experiment No. 1 is Described and Discussed</u>
Oak Ridge National Lab. Contract W-7405-eng-26, 179 p., Apr. 1953

Tests of valves, seals, pumps, etc., are described.

Supp. 069
Gneen, J.
<u>Elastomers for Liquid Rocket Fuel and Oxidizer Applications</u>
Indus. and Eng. Chem. - Product Research and Development V.2, N.2, June 1963, p. 126-133.

It is desirable to prepare seals, gaskets and flexible connectors from elastomeric polymers, resistance of commercially available and experimental polymers to hydrogen-type fuels, nitrogen tetroxide and fluorine-containing oxidizers was investigated, on basis of static and dynamic tests several materials have been recommended.

Supp. 070
Fogg, Warren E.
<u>An Explosive-Operated Valve</u>
A.D. 291578

The object of this investigation was to develop a method for rapidly stopping the flow of gas through a tube and permanently sealing the tube against further leakage. A "valve" has been developed which will close an aluminum tube (0.5 in. O.D. by 0.065 in. wall) in approximately 25 microseconds, giving a hermetic seal thereof. The closure is obtained by radially collapsing the tube wall under an explosive load.

Supp. 071
Paxton, B. R.
<u>Testing High Speed Seal Carbons</u>
ASLE Trans. V.5, n.2, Nov. 1962, p. 308-314

Test method used to guide development of carbon graphite suitable for main shaft seals in jet aircraft is described; simplified seal assembly was used; rubbing speeds up to 16,700 FPM and face loading to 18 psi were attained. The apparatus and sample preparation methods, wear rates, air leakage rates past face seal, and seal temperature rise due to friction were measured, data for four carbon graphites are included to show effects of carbon type, face pressure, rubbing speed, and mating face material.

Supp. 072
Schweiger, F. A.
<u>Performance of Jet Engine Contact Seals</u>
Lubrication Eng. V. 19, n.6, June 1963, p. 232-238

Application of circumferential gas seals (type of contact dynamic seals) to main shaft positions of jet engines; description of seal design, development and evaluation; means for solving problems of excessive seal wear and some performance defects; maximum capabilities of this type of seal to date are 12,000 FPM rubbing velocity, 90 psi <u>pressure</u> differential and 750 F. ambient temperature.

Supp. 073
Abar, John W.
<u>Rubbing Contact Materials for Face Type Mechanical Seals</u>
Journal of the American Society of Lubrication Engineers
October, 1964, Lubrication Engineering

Silicon carbide, 85% alumina, 6% Ni tungsten carbide and alumina coating on 316 stainless steel have been investigated for their limitations as water lubricated seal face materials. The materials were tested rubbing against carbon graphite at rubbine speeds of 2,320 ft/min and face loads of 300 psi. PV limitations were found to be clearly defined by thermal failures which resulted in exaggerated carbon wear and in some cases seal failure.

Thermal stress factors (incorporating physical properties) were established as a material selection criteria based on correlation made with the same and limiting PV values at which thermal failures occurred. - (AUTHOR)

Supp. 074
Ambrose, D.
<u>A Simple Glass-to-Metal Pressure Tight Coupling</u>
Journal Scientific Instruments (GB), Vol. 40, No. 3, 129, March 1963

A glass-to-metal coupling based on the unsupported area principle is described. The principle is well known and widely adopted but the present application has the advantage of simplicity in its use of standard pipe couplings.

Supp. 075
Anonymous
<u>Seals - Reference Issue</u>
Machine Design, No. 14, Volume 36, June 11, 1964, pages 1 - 242

The Reference Issue contains useful design data as well as listings of commercial sources of supply. The Design Data Section contains technical chapters on seals and sealing devices for static and dynamic applications. Whenever possible the date is presented in the forms of tables, charts and graphs. The Product Directory contains eight sections presenting detailed information on types and manufacturers of seals, sealing devices, packings, gaskets and sealants.

Supp. 076
Anonymous
<u>Self-Tightening Pressure Vessel Seals</u>
Engineering V 199 n 5165, April 16, 1965, p. 498

New design of closure is based on use of endless metal seal ring of heavy section that is fitted into groove within bore of pressure vessel. About half of radial width of ring locates in groove, and end plate of vessel presses against inner half of ring. Bolt loading is only required to ensure initial sealing. Diagrams show joint arrangement for pressures up to 7000 psi and also up to 30,000 psi.

Supp. 077
Anonymous
Profiting by Mechanical-Seal Advances
Chemical Engineering, Vol. 71, Part 1: 83-90, March 2, 1964

Discussion of sealing problems pertinent to the chemical process industry. The information is a summary of informal talks at an ASME technical session covering the selection, design and operation of mechanical seals. Although no specific data or information is presented, the discussion is informative.

Supp. 078
Apple, F. C.
Leakage of Water From Gasketed Joints Proposed for the HWCTR - Part II and Pump Mechanical Seal Vapor Leakage
AEC Research and Development Report - Engineering and Equipment, DP-611, dated August 1961, Savannah River Laboratory, Aiken, South Carolina

This report presents leakage data taken on typical gasketed joints for the HWCTR (Heavy Water Components Test Reactor) vessel. Leakage was 0.3 pound of water per year per linear inch of gasket during tests that consisted of about 100 cycles with deionized light water at a maximum pressure of 1500 psig and a maximum temperature of 250°C. Spiral wound gaskets of stainless steel and asbestos and four pipe-to-tubing fittings were tested. In addition, two conventional quick-disconnect couplings (Grayloc and Conoseal) were leak tested for use on the bayonet fuel housings of the isolated coolant loops.

Tests on the centrifugal pump operating at 3600 RPM showed that the average loss of water vapor from the conventional mechanical seal was only 1.4 #/yr during a five-month test. The water leakage was 910 #/yr, however, this leakage is considered to be recoverable. The pumps circulated deionized water at 260°C and 850 psig while seal water temperature was maintained at 45°C by cooling.

The test data is very well organized and is presented in considerable detail.

Supp. 079
Apple, F. C., Burns, D. L., Sedes, J. E.
Leakage of Water From Pump Mechanical Seals Technical Division
AEC Research and Development Report (Savannah River Laboratory), DP-666, November 1961
(Available from the Office of Technical Services - Washington, D.C.)

Both vapor and liquid leakage measurements were made in bench-scale tests on mechanical seals from three manufacturers. Pump shafts diameters were 2.625 and 3.563 inches diameter. Speeds up to 3600 RPM and pressures up to 1000 PSIG were considered. Seal cooling was employed and the leakage of D_2O was recovered and therefore not considered to be an economic loss in an operational cycle. The report presents considerable leakage test data.

Supp. 080
Armstrong, D. C., and Blais, Normand
Large Rotary Vacuum Seal
Review Scientific Instruments 34,440 - (1963)

Successful rotations of large circular sections or ports in the walls of a vacuum chamber was permitted by the simple seal described. Friction arising during rotation was small and no mechanical drives were required to turn ports up to 2 ft. in diameter. Ultimate pressures of 3×10^{-7} mm were obtained in the "mol" beam chamber tested, and the pressure rise when the lid was turned during operation was $< 2 \times 10^{-8}$ mm.

Supp. 081
Arnold, W. C, Eser, W. J., Jr.
Materials and Designs for Nonlubricated Seals
Mechanical Engineering, December 1964, p. 38-43

This article discusses the results of dynamic wear rate tests of various "filled" teflons and a special polymer. The mating surface for these tests was cast iron, chromium, stainless steel and aluminum. Data on characteristic properties is presented for all plastic materials tested. Considerable detail on piston ring design is presented.

Supp. 082
Beaves, L. C, Steward, D. I.
Two Ultrahigh-Vacuum Radio Frequency Feedthroughs
Review Scientific Instruments, 34, 449-50 (1963)

The feedthroughs presented used materials having low vapor pressures even at high temperature, could withstand repeated bakeout without developing leaks, could conduct high current at radio frequency, were small and strong, and were suitable for use in ultrahigh-vacuum systems.

Supp. 083
Bealkowski, L. S., Stachiw, J. D.
Metallic Seals for Deep Submergence Underwater Vehicles
Marine Technology, V 2, n 2, April 1965, p. 116-24
(also ASME Paper 64WA/UNT-7 presented November 29-December 4, 1964 meeting)

Underwater manned or instrumented vehicles for exploratory or defense missions require static and dynamic seals of axial or radial types to seal joints against high, external, hydrostatic pressures. Metallic knife-edge seals can do many of the sealing functions with superior performance in terms of pressure capability, low dynamic friction, controlled unit loading, low breakout force, self cleaning and reusability. Theoretical and practical design consideration and experimental data generated to date are discussed.

Supp. 084
Blow, C. M.
Development and Testing of Elastomeric Materials for Fluid Sealing Applications
Aircraft Eng., V 36, n 7, July 1964, p. 208-12

Problems associated with the development of rubbers for use with fire-resistant phosphate ester hydraulic fluid Skydrol 500A are presented. The use of rubbers as seals is discussed.

Supp. 085
Boebel, C. P., Mackie, N. A., and Quaintance, C. C.
Outgassing Studies of Space Materials
Trans. Nat'l. Vacuum Symposium 9, 307-10 (Pub. 1962)

Outgassing and permeability studies of several polymeric organic materials useful in space vehicles are described. An automatic recording vacuum balance combined with a residual gas analyzer was used. Samples were held at 160°F and approximately 1×10^{-6} mm for 3 days, although degassing was usually complete in < 10 hours. Detailed results are given for castable elastomeric materials intended to function as a low temperature, low permeability adhesive seal.

Supp. 086
Brown, P. F., Gordan, N., and King, W. J.
A Test Method for Evaluating Gas Turbine Engine Seal Materials
ASLE Preprint No. 65AM3C5, presented at the 20th ASLE Annual Meeting, May 1965

The severity of the temperatures, pressures and speeds encountered by shaft seals in today's turbojet engines has required the establishment of an experimental rig development program to provide the optimum materials. Such a program has been developed and, in cooperation with several carbon manufacturers, has resulted in several superior carbon grades. Several stainless steel seal plates with machining and corrosion advantages also came from this program.

The method of testing employed in the early years of this material evaluation program was the conventional full factorial approach. Economics dictated that a more efficient test method be developed. A test method employing a statistically planned experiment approach has proven capable of reducing the number of tests needed to successfully screen seal material combinations. A description of the old and new test methods is presented and a comparison made of the ability of each method to screen materials. - (AUTHOR)

Supp. 087
Carson, Ralph S., Hendricks, Charles D. (Univ. of Illinois, Urbana)
Vacuum Seals
Review of Scientific Instruments, 34, 118-119, 1963

Vacuum-tight plugs and feed-throughs for electrical connections through the base plate of a vacuum bell jar system were produced from brass fittings machined to accommodate an O-ring. Seals to 10^{-7} mm. were made without the use of wax or varnish.

Supp. 088
Clark, Murl W.
Sealing Problems in Mobile Hydraulic Systems
National Conference on Industrial Hydraulics, Volume XVII, 19th Annual Meeting, October 17, 18, 1963, Chicago, Illinois, p. 46-53

In applications to earthmoving equipment, hydraulic components are subject to the severest environment. The number of these applications has been increasing rapidly. Leaks account for a large portion of equipment failures and require special consideration. Testing and design of improved piston packings are described here. Test results on hoses demonstrated the unpredictability of hose life and thus the need for a joint development program between hose manufacturers and users.

Supp. 089
Courtney, W., et al
Sealing Techniques for Rotation in Vacuum
Astronautics and Aeronautics, V 2, n 2, February 1964, p. 40-4

Various types of low-speed hermetic seals for either high or low torque transmission at room temperature are discussed. For low-torque transmission, magnetic drives offer solutions up to 10,000 rpm and about 400 C.

Supp. 090
Crawley, D. J., de Csernatony, L.
Degassing Characteristics of Some "O" Ring Materials
Vacuum, V 14, n 1, January 1964, p. 7-9

Limitations imposed by using organic "O" rings in valves, flanges and other vacuum connections where metal gaskets may be inconvenient are discussed. Degassing rates of nitrile and Viton A "O" rings measured both at ambient temperature and after baking at various temperatures are compared with typical metal degassing rate. "O" ring cleaning techniques are briefly discussed. Contamination may result from baking at too high temperature.

Supp. 091
Darcey, Owen A.
Fail-Safe Stressed in DC-8 Hydraulic Changes
Hydraulics and Pneumatics, Vol. 16, 88-92, August 1963

This article presents actual operating experience in overcoming some troublesome leakage and control problems in hydraulic systems.

Supp. 092
Decker, Arthur L.
Unique Way to Seal Rotating Shafts
Chemical Engineering, Vol. 71, No. 15, July 20, 1964

Describes several applications of spring-loaded continuously lubricated packing arrangements for shaft sealing. The device has been used on low speed agitator shafts and centrifugal pump shafts with considerable success. One of the main disadvantages is the leakage of lubricant from the packing box.

Supp. 093

Faber, G., Heise, M.
Use of Silver and its Alloys for Seals in Oxygen Compressors
Brown Boveri Review, V 50, n 6-7, p. 430-4, June-July 1963

It is shown how additions of aluminum oxide to silver improves heat resistant properties of metal, which is not affected by oxygen. Comparitive tests carried out with pure silver and silver copper alloy, and silver hardened with Al_2O_3 showed superiority of hardened silver in temperature range from 200 to 300 C.

Supp. 094
Genz, Orville F.
Plastics - Proven as Hydraulic Components
National Conference on Industrial Hydraulics, Volume XVII, 19th Annual Meeting, October 17, 18, 1963, Chicago, Illinois

Describes various seals and seal materials. Presents test data on life tests for various rod packing materials both lubricated and dry.

Supp. 095
George, J. C., Silverman, B
Seal Design For Space Station Usage
Rubber World V151, N3, December, 1964, p. 51-9

Design of static and dynamic seals for manned space stations with 200-to-300 nautical mile orbits, minimum life of one year, and possible life of five years are presented. Effect on seal design of temperature, vacuum, radiation and meteoroids are reviews. Also covered are configuration, concepts, repair of seals and evaluation of metals, epoxies, polysulfides, silicones, silicone rubber, polyurethanes, polyesters, polyvinylidene chlorides, polyvinyl fluoride resins, polychloroprenes, phenolics and butyl rubber for seals. Study was made by Lockheed - California Company for NASA Manual Spacecraft Center, Houston, Texas, under Contract No. NAS 9-1563.

Supp. 096
Gifford, F. E. and Dolenga, A.
Direct Glass-Metal Seals
Review of Scientific Instruments (USA) Vol. 35, No. 5, 591-3, May, 1964

Direct glass-metal seals have been developed for use in tubes containing hot alkali vapors. Six types of glasses (Corning code nos. 7052, 7056, 7720, 3320, 1720, 7740) in conjunction with three types of metal (tungsten, molybdenum and Kovar) were employed in this study. Combinations of glasses and metals previously considered incompatible by other sealing methods were successfully united by utilizing this technique. Tubes made with these seals have been thermally cycled and they remained vacuum tight.

Supp. 097
Gunn, A. J.
Seals Tighten Grip on Machine Maintenance
Can.Machy and Metalworking V75 N7, July, 1964

A summary is given of new developments in seals and how they can be best applied in machinery. A discussion of high pressure Viton oil seals supported by a Teflon lip is presented. This combination greatly improves performance in terms of high PV factor.

Supp. 098
Hanlon, Paul
Lubrication of Reciprocating Compressor Seals
ASLE, Preprint No. 65AM3C2; presented at the 20th ASLE Annual Meeting, May, 1965

Most large compressors built today are packed at the piston and rod with floating metallic seals. This paper discusses the mechanism or theory of the "micro" aspects of lubrication as well as the practices of "macro" lubrication of these seal rings.

Steadily increasing pressures and speeds compound what are already poor conditions for good lubrication. How these conditions affect packings and piston rings and what means are used to overcome these are also discussed. (AUTHOR)

Supp. 099
Heyn, W. O.
New Machine Improves Testing for Shaft Seal Reliability
SAE Journal, Volume 73, No. 1; January, 1965

Briefly describes a shaft seal testing unit built to meet the specifications of the Oil Seal Division of the Rubber Manufacturers Association. The unit features a 3-HP electronically controlled variable speed drive, tapered arbor to provide dynamic eccentricity and a cast aluminum bronze test head designed for 1200 psi internal pressure. No performance data is supplied.

Supp. 100
Hodgson, J. N.
Designing Molecular Pump as Seal-To-Space
ASME Paper 65-GTP-15 for meeting February 28 - March 4, 1965

The molecular pump has been shown to be a valuable seal-to-space element. A mathematical model has been developed which accurately describes molecular-pump performance. For space applications, operation can be between molecular and continuum flow. A computer program was used to allow solution of an optimum molecular-pump configuration for minimum leakage to space. The test program has given confirmation of the mathematical theory and has demonstrated exceptional leakage-restricting capability of the molecular pump.

Supp. 101
Hort, F. D., Zorowski, C. F.
Onset of Mechanical Separation In Bellows-Supported Rotary Face Seals
ASME Paper 65-GTP-4 for meeting February 28 - March 4, 1965, p. 8.

Problem of incipient dynamic and mechanical separation in face seals of gas turbines is analyzed assuming bellows to be represented by series of distributed springs and dampers supporting rigid seal carrier. Expression is developed for distribution of contact forces between seal and mating rings generated by coupled effects of shaft pulsation and mating-ring wobble; this expression is used to determine operating conditions which will produce onset of separation. Equations and graphs define minimum initial compression necessary to maintain continuous contact.

Supp. 102
Horvath, G. E.
TFE Fluorocarbons For Gaskets and Packings
Machine Design V36 N23 September 24, 1964, p. 166-72

Up-to-date guide to proper specification and use of filled and unfilled TFE gasket and packing material; available forms, mechanical properties, friction and wear characteristics, and service limits are listed for pure TFE and numerous TFE compounds.

Supp. 103
House, Philip A. (Wright Patterson Air Force Base, Ohio)
Resilient Seal Materials For Liquid Rocket Propellants
Symposium On Materials For Space Use, Seattle 2, 9 pp. (1963)

In O-ring tests, Teflon (I) was the best sealant for N_2O_4, allowing no loss in 30 weeks and 0.25% loss of retained contents in 40 weeks from a test jig. I was better than CaF_2-filled I, butyl rubber (II), Viton B (III), ethylene-propylene rubber (IV), and polythethylene (V). For ClF_3, unfilled and filled I allowed no loss after 50 weeks in contrast with 1 week vor V. V was best for H_2O_2, showing 0.25% loss in 1 week and 2.25% in 40 weeks. V was better than I, III, silicone rubber, and II. A 50:50 NH_2-NH_2-Me_2NNH_2 mixt. was held best by V (no loss in 40 weeks), as compared with I, II, IV, and polybutadiene. Among the elastomeric O-rings coated with 0.001, 0.005, and 0.01 in. of metal, the ones coated with 0.01 in. metal were the best. Ag/Pb (0.008/0.002 in.) and Cu/Pb (0.008/0.002 in.) appeared to seal better than Pb 0.010 in. with N_2O_4. Expulsion bladders made with foil-coated elastomer resist penetration by N_2O_4 for at least 4 weeks. Best adhesion is with Al and II. Pinholes in the foil did not destroy the effectiveness of the laminate. -(AUTHOR)

Supp. 104
Huang, T. S.; Swift, G. W.; Kurata F.
High Pressure, Low Temperature, Glass to Metal Seal
Review Scientific Instruments (USA) Vol. 35, No. 5, 637-8 (May, 1964)

A technique is described for sealing Pyrex 7740 glass to AISI 316 stainless using Hysol adhesive to form a joint capable of withstanding pressures in excess of 600 atm. in the temperature range -77 °K to 300 °K.

Supp. 105
Kamphausen, H. A.
Method For Making Multiple Glass-Tungsten Rod Seals
Journal Scientific Instruments (G.B.) Vol. 40, No. 12, 605, December, 1963

A simple method for sealing suitable metal rods into a glass envelope is described. This technique is especially recommended for multiple seals used for experimental purposes.

Supp. 106
Kelly, Robert W.; Wood, Glenn M.; Manfredi, Daniel V.
Cermet Face Seals For Inert Gas Environments
ASLE Preprint No. 65AM4C4, presented at the 20th ASLE Annual Meeting, May 1965

Cermet face seals operating in inert gas environments were utilized to effectively seal the shafts of centrifugal liquid metal pumps employing conventional oil lubrication systems. These seals were used to isolate the oil lubricated bearings from the liquid metal region of the pump and from the ambient air which cooled the drive motors. Development problems for these seals were intensified by the need for liquid metal compatibility and for maintenance-free operating lifetime in excess of 10,000 hours with very low oil and inert gas leakage rates being permissible.

The highlights of a development program for various shaft face seals with cermet rubbing faces operating in dry inert gas and in inert gas blanketed oil environments are described. Pertinent seal manufacturing techniques, performance data, and experimental techniques evolved during this program are presented. A total of over 60,000 hours of face seal testing was accumulated while satisfying the program objectives. Successful endurance evaluation tests of the cermet shaft face seals included a test of over 15,000 hours and several other tests for periods up to 10,000 hours. - (AUTHOR)

Supp. 107
Lake, G. J.; Lindley, P. B. and Thomas, A. S.
High-Vacuum Sliding Seal
Journal Scientific Instruments 40, 40 (1963)

This seal was developed to allow direct transmission of reciprocating motion into a high vacuum chamber, for experiments in connection with the fatigue of rubbers in the absence of oxygen.

Supp. 108
Lessley, R. L., Hodgson, J. N.
Low-Leakage Dynamic Seal to Space
ASME Paper 65 GTP-14 for meeting February 28 - March 4, 1965, p. 20

A space power system, turbine-shaft seal capable of restricting leakage of mercury vapor and of oil to space environment is described. The seal was designed to permit use of oil-lubricated bearings in turbine-alternator assembly of mercury rankine cycle. The seal uses a slinger-pump, a visco pump and a molecular-pump element. The role and limitations of each element are discussed. Tests results are presented. Work was part of SNAP-8 power-conversion system and the authors believe it to be the first successful development of a functional dynamic seal to space.

Supp. 109
Lewis, Robert Bee
Predicting the Wear of Sliding Plastic Surfaces
Mechanical Engineering - October, 1964, p. 32 - 35

The two most important criteria in the design of unlubricated plastic bearings, piston rings and seals are PV-limits and wear. The purpose of this article is to show that simple economical tests can form the basis for wear prediction. The wear of a plastic surface sliding without lubrication against another surface is predictable if:

1. The temperature rise due to frictional heating added to the ambient temperature, results in a surface temperature below the critical value associated with the plastics PV-limit.
2. The mating surface is defined in terms of material, hardness and roughness.
3. The environment is defined and free of contaminants.

Wear rate data on various plastics is given as a function of pressure. Limitations on material surface combinations, finish and hardness are discussed, as well as cyclic operation.

Supp. 110
Lipson, H. G. and Bouthellette, L. O.
Epoxy Vacuum Seal For Introduction of Leads Into Cryogenic Equipment
Vacuum 13 (4) 145 (1963)

A satisfactory cement for introducing thermocouples into evacuated vessels such as Dewaro consists of a mixture of 3 parts of Hysol 6020 resin and 1 part Hysol 6020 hardener.

Supp. 111
Llewellyn D. R. and O'Connor, C. J.
Convenient Break Seal For The Transfer of Gases Particularly Suited to Spectometer Samples
Chem. and Ind. p. 52 Jan. 11, 1964

Supp. 112
Mancebo, L.
A Cold High-Vacuum Seal Without Gaskets
Proceedings Institute Electrical Engineers (USA), Vol. 51, No. 9, 1250-1, September, 1963

A cold high-vacuum sealed enclosure is described in which the seal is obtained by evacuating the space between two sections mating at optically flat flanges. If required for re-use, a 1000 A Al film is evaporated on one flange prior to use. After 1-1/2 years shelf-life the system was still vacuum-tight.

Supp. 113
Matt, Richard J.
High Temperature Metal Bellows Seals For Aircraft and Missile Accessories
Transactions of the ASME-Journal of Engineering for Industry, August, 1963.

This is a general presentation on various aspects of metal bellows of the type used in conjunction with dynamic face seals. Information on bellows construction, pressure limits and enviromental considerations are reviewed. Seal ring flatness and seal ring retention techniques and mating ring "cam hop" are discussed as well as vibration, weldability and fatigue. It should be noted that this type seal need not be restricted to high temperature service.

Supp. 114
Memsforth, L. H.
Operating Experience With Mechanical Shaft Seals On a High Temperature Water Circulation Pump
Canadian General Electric Limited - Civilian Atomic Power Department - Report-R60CAP53, September 30, 1960

This report presents data on a circulating pump for a reactor component test loop operating at 1075 psig and 500 °F. The centrifugal pump mechanical face seal has a 304 stainless steel (with stellite overlay) rotaing element and a carbon type mating ring. The seal is cooled by an external heat exchanger. Considerable test data is presented on long time seal performance.

Supp. 115
McHugh, J. D.
Adiabatic, Laminar Flow in Concentric Seals
ASLE Preprint No. 65AM4C3, presented at the 20th Annual Meeting, May, 1965

For calculations of flow and power loss in concentric, cylindrical sleeve seals such as are commonly used as flow restrictions on rotating shafts, a time-consuming iterative procedure is normally required if viscous dissipation effects are taken into account. Making some simplifying assumptions, however, it is possible to correlate a dimensionless temperature rise and ratios of adiabatic to isothermal flow rates against a single dimensionless number containing easily determined seal variables. The sleeve seal designer is therefore provided with a rapid method for assessing the effect of viscous dissipation upon flow, power loss and fluid temperature rise for laminar flow. (AUTHOR)

Supp. 116
Munchhausen, H. von, Schettko, F. J.
Investigation of Outgassing Process of Silicone Rubber
Vacuum V 13, N 12, December 12, 1963, p. 549-53

Study on silicon rubber as a gasket material; outgassing is reduced by two orders of magnitude through heat treatment; optimum condition of pretreatment is determined; experiments show that outgassing is based on diffusion mechanism.

Supp. 117
Papirov, I. I.
Ultrahigh Vacuum Demountable Seals and Valves That Can Be Heated
Prebory i Tekhnika Experimenta (USSR), 1962, No. 2, 5-15 (March-April).
English translation in: Instruments and Experimental Techniques (USA), No. 2, 225-33 (March - April 1962; published November, 1962) - Instrument Society of America, Penn-Sheraton Hotel, 530 William Penn Place, Pittsburgh 19, Pennsylvania

Supp. 118
Parker, Elliot J.
Design and Selection of Packings and Seals for Hydraulic Components
National Conference on Industrial Hydraulics, Volume XVII, 19th Annual Meeting, October 17, 18, 1963; Chicago, Illinois

This paper is a non-technical discussion of packings and seals and the various factors which influence their design, selection and performance. These include mechanical design, installation, fluid, and environment.

Supp. 119
Peickii, V. L., Bowen, E. K.
Modern Materials and Designs for Rear Crankshaft Oil Seals
SAE - Paper 895B for meeting September 14 - 17, 1964, 8 p.

Paper is concerned with premium nonsautomotive engines and engines using one piece continuous circumferential lip type oil seals; emphasis is on diesel engines. Low temperature sealing problems are discussed and test results of present and new seal materials are given.

Supp. 120
Piehn, L. D.
The Application of Tungsten Carbide for Mechanical Seal Faces
ASLE; Preprint No. 65AM3C4; presented at 20th ASLE Annual meeting, May, 1965

Some applications where tungsten carbide has provided superior performance on mecahnical seal applications are discussed. Background information is given on the factors leading up to the standard use of tungsten carbide in mechanical seals with particular emphasis being placed on use in boiler feed service, high pressure, refinery and pipeline applications. Some work regarding corrosion resistant grades of tungsten carbide is mentioned and the effort to expand the use of carbides to a wider field are described. - (AUTHOR)

Supp. 121
Rasmussen, W. W.; Boink, R. V.
Influence of Oil Seal Design Parameters on Lip Opening Pressure
SAE - Paper 895A for meeting September 14 - 17, 1964, 7 p.

Evolution and significance of Lip Opening Pressure (LOP) inspection technique to permit rapid measurement of value that approximates radial load exerted by the seal on the shaft is presented. A force analysis shows LOP to be directly proportional to radial load and inversely proportional to shaft diameter and beam length. It is concluded that LOP is an adequate indicator of radial load consistency for small and average size seals.

Supp. 122
Rietdijk, J. A.
Positive Seal for Pistons on Axially Moving Rods
Engineer V218 N5666, p. 346-7, August 28, 1964

Seal comprises elastic rolling diaphragm mounted between piston rod and cylinder. The devices were originally developed for use on Philip-Sterling machines and is also considered suitable for other applications where oil penetration and gas leakage must be completely avoided.

Supp. 123
Roach, W. R., et al
Electrical Lead Seals for Cryogenic Applications
Review of Scientific Instruments, 35:634-6, May, 1964

Supp. 124
Rothberg, Samuel B.
Centrifugal Force Keeps Fluid in Groove of Liquid Shaft Seals
Product Engineering - November 11, 1963, pp. 85 - 87

This article discusses several patented ideas for sealing centrifugal type seals at zero speed. A magnetic device is also discussed.

Supp. 125
Scott, P. A.
Plutonium Ricycle Test Reactor Primary Coolant Pumps
ASME - Paper 63-WA 234. Hanford Laboratories, Richland, Washington

The Plutonium Recycle Test Reactor primary coolant pumps use face-type mechanical shaft seals and incorporate a large flywheel for emergency flow during power outages. The test and development work preceding and during initial PRTR operation revealed several major problems. This report discusses these problems and their solutions. Mechanical seal life for PRTR was extended to several thousand hours.

Supp. 126
Shvets, I. T.; Dijban, E. P.; Khaven, V. Yu
Heat Exchange in Labyrinth Seals of Gas Turbine Rotors
(Teploobmen V Labirintovykh Uplotneniyakh Rotoror Turbin)
Energomashinostroenic N12, December 1963, p. 8-11.

Results of experimental investigation of influence of clearance under labyrinth seal cam, cooling air flow rate, and rotor speed on heat exchange and hydraulic resistance of seal.

Supp. 127
Smoley, E. M.
Design Criteria for Sealing Gasketed Joints
Machine Design, V15, N27, November 21, 1963, p. 174-7

Criteria for seal designing are represented by relationships between minimum sealing stress and such variables as internal pressure and viscosity of sealed fluid, surface roughness of flange, ratio of gasket width to uncompressed thickness (shape factor) and gasket material; above relationships are known for rubber and cork/rubber compositions, and can be applied to other available gasket materials for simple gaskets and flange design purposes.

Supp. 128
Stephens, C. A.
Seals and Lubricants
Machine Design, V37, N2, January 21, 1965, p. 172-6

Short term and long term effects of lubricant on rubber oil seals are described; additional tests for determining suitability of particular rubber compound for particular lubricant are presented; effect of lubricant formulation on seal-oil life is shown in table.

Supp. 129
Strach, L., Depl-Ing
Contactless Piston Seal for Internal Combustion Engines
The Engineer, Vol. 217, May 29, 1964, pp. 969-970

The four cylinder, four stroke carburettor engine, known as the boxer configuration, was designed to eliminate the need for an oil film at the cylinder walls. Sealing is accomplished by a sharp edge at the piston periphery which serves as a high restriction device to the leakage of combustion gases. As a result of this design, increased combustion temperatures and correspondingly increased efficiencies are predicted.

Supp. 130
Sussman, S.
Causes and Cures of Mechanical Shaft Seal Failures in Water Pumps
Heating and Piping, 35:141-8; September, 1963. Discussion 36:108, January, 1964

Supp. 131
Sussman, S.
Does Water Treatment Affect Seal Performance
ASHRAE J, V6 N11, November 1964, p. 48-52

Discussion refers to performance and causes of failure of mechanical shaft seals in chromate treated waters.

Supp. 132
Taplin, John F., Fobian, G. W.
Compressor Control Seals With Rolling Diaphragms
Hydraulics and Pneumatics, 16:107-7, November, 1963

This article discusses a typical application of a deep rolling, fabric reinforced elastomer which prevents leakage between a moving piston and a cylinder. A high degree of sealing effectiveness is achieved.

Supp. 133
Teichman, O. E., et al
Experimental Techniques in Testing Liquid Sealing Face Seals
Journal of the American Society of Lubrication Engineers, September, 1964, pp. 345-357

During an investigation of 2 inch diameter face seals sealing aqueous liquids under pressure from 100 to 2000 psi and running at speeds up to 3600 rpm, a cradled seal tester was developed that permits accurate measurement of seal torque. To determine from the torque readings the coefficient of friction between the mating surfaces, the closing force of a "zero balance seal" was measured as a function of seal pressureand interpreted in terms of coefficient of friction and pressure distribution between the seal faces.

Supp. 134
Vaisnys, R. J., Montgomery, P. W.
Materials for Ultrahigh Pressure Sealing in Bridgman Anvil Devices
Review of Scientific Instruments, V35, N8, August, 1964, p. 985-9

Epoxide, Eccobond 104 and reconstituted mica, 6S Isomica, are found to be suitable substitutes for conventionally used pyrophyllite for use in sealing rings of 100 kbar anvils; materials are superior to pyrophyllite in case of fabrication, electrical resistance and ability to retain thick (0.020 to 0.026 in.) samples; epoxide is sufficiently transparent to X-rays to permit its use as gasket in high pressure diffraction studies.

Supp. 135
Watkins, S. Jr.
Honeycomb Rotor Seals
Product Engineering, V36, N1, January 4, 1965, p. 61-5

Several methods of applying stainless steel, Inconel and Hastelloy foils fabricated into honeycomb and brazed to stator of compressor or turbine are detailed. Seals are applicable especially where temperature and speed are high. Honeycomb is easily deformed to provide a non-contacting seal. Applications to shaft seals and lobes of displacement compressors are illustrated.

Supp. 136
Watson, R. D. and Roche, K. W.
A Comparison of Operating Characteristics of Flat-Face and Hydrodynamically Lubricated Rotating Mechanical Shaft Seals in High Pressure Water
Atomic Energy of Canada Limited, Report No. CRE-1104 dated August, 1962 - Chalk River, Ontario

Describes tests on a tapered mechanical shaft seal for high-pressure nuclear applications. The tapered faces are designed to provide a hydrodynamic film to aid in providing a broad range of operating conditions. The results are presented in graphical form for various speeds and pressures. Future work is planned in certain problem areas.

Supp. 137
Watt, P. Ridgway (Vitamins Ltd, Tadworth Engl)
An Elastomer Seal for Glass Vacuum Desiccators
Chem. Ind. (London) 1963, 567-8

A molded elastomer ring is described, that provides an effective seal for glass vacuum desiccators and protects both flanges from accidental damage. The molding is in the form of a channel that snaps over the upper flange and carries an annular undercut, sealing lip on its lower face. A heavy rim around the outside of the molding projects downward below the level of the sealing lip.

Supp. 138
Wheatley, J. C.
Epoxy Resin Seals to Copper and Nylon For Cryogenic Applications
The Review of Scientific Instruments, Volume 35, Number 6, June, 1964

Detail description covering the use of epoxy resin in making vacuum and pressure seals to copper and nylon for low temperature work. The technique provides an electrically non-conducting, non-magnetic seal at temperature as low as 1 ^{o}K. No test data is presented. The work was supported by the U.S. Atomic Energy Commission.

Supp. 139
William, John C.
Shaft - Seal Systems for Large Power - Reactor Pumps
Nucleonics, Vol. 23, No. 2, February, 1965

The economic and technical considerations favoring the use of shaft-seal main coolant pumps are reviewed. Descriptions of the face-contact and radial-clearance seals are presented along with design details and expected performance. The role of reliability and its significance to seal life is estimated for a 90% probability factor. Also considered are the "adjustable film" concepts and fixed film pressure breakdown designs. Operational data for typical mechanically sealed pumps in nuclear service are presented. A total of seventeen references are cited.

Supp. 140
Young, T. R.
Ultrahigh Vacuum Leak Sealing Material
Review Scientific Instruments (USA), Vol. 35, No. 1, 116, January, 1964

GE SR-82 silicone resin can permanently seal metal and glass systems and withstand bakeout to 400 °C. Pressures of $2 \cdot 10^{-10}$ torr can be obtained without any evidence of hydrocarbon contamination due to the seal.

Supp. 141
Rathbun, Forrest O., Jr.
Metal to Metal and Metal-Gasketed Seals
Machine Design, August 5, 1965

Static seals for extreme environment applications are reviewed for both metal to metal and metal-gasketed seals. The fundamental requirements for effective sealing are presented analytically and correlated with an extensive graphically presentation based on basic experimental investigations by the author. The experimental work was performed under NASA sponsorship. (See also SAE Paper No. 650312, "Metal-to-Metal and Metal-Gasketed Seals," presented at the Aerospace Fluid Power Systems and Equipment Conference, Los Angeles, May, 1965; Conference Proceedings pp 156 to 169.

Supp. 142
Shepler, Paul R.
Seals for Fluid Applications
National Conference On Industrial Hydraulics, Volume XVII, 19th Annual Meeting, October 17-18, 1963, Chicago, Illinois

The paper discusses various types of "split ring seals" for internal combustion engines, reciprocating pumps and actuators. The effect of conformability, axial loading and step height on leakage are presented. Also shown is a monogram for leakage of reciprocating split ring seals.

Supp. 143
Aleksandrov, A. A. Eng. Ind. October 1966
Leaktight Seal in Agitator Devices
Chemical and Petroleum Engineering (English translation of Rhimicheskoe i Neftyanoe Mashinostroenie), Vol. 1, January 1965, p 61-62.

New design of packless seal operating under h-p (25 atm and higher) or in high vacuum (2×10^{-6} m m Hg) and at any shaft rpm makes it possible to mount drive shaft in region adjacent to lid of machine, on sleeve bearings or ball bearings; it requires no stuffing box, labyninth seal, or flanged seal on shaft; test results performed for solution of rosin in alcohol and hydrogen at 20 atm showed no leakage from autoclave.

Supp. 144
Alford, J. S., Lawson, G. W. Eng. Ind. July 1966
Dimensional Stability and Structural Integrity of Labyrinth Seals
S.A.E. Paper 660048 for meeting January 10-14, 1966, p31

Seal most used in aircraft gas turbines is labyrinth type, grouped into straight and stepped tooth configurations; dimensional stability and structural integrity are compromised if certain mechanical criteria are not incorporated into design; factors to consider and design features to minimize heat generation and distortion resulting from rub are outlined with reference to fatigue failures and failing modes; means to protect seals from fatigue caused by self-excited vibration are indicated; analytic approach to mechanical design of labyrinth seals for aircraft gas turbines is also presented.

Supp. 145
Anonymous ASM 18-47013.66
How to Reduce Corrosion of Seals and Shafts
Material Design Engineering, Vol. 63, No. 3, March 1966, p74-75.

Corrosion failures of seal materials are believed to be due to electrochemical attack in more cases so that a primary requirement of seal materials is low concentrations of water-soluble components which can produce an electrolyte with moisture absorbed from the atmosphere. Several tests for evaluating the corrosion resistance of seal materials are given.

Supp. 146
Anonymous Eng. Ind. September 1966
P.T.F.E. Cord for Gasketing and Sealing
Australian Plastics and Rubber Journal, Vol. 21, No. 248, March 1966, p35

Advantages of flexible cord of unsintered polytetrafluoroethylene for sealing valve stems and flange joints; cord withstands vibration, operating temperatures from -200 to 250C, working pressures over 500 psi, and attack of most chemicals; it seals against steam, gases, oils, gasoline, liquid oxygen, acids, alkalis, and solvents.

Supp. 147
Belevtsev, B. A., Freitag, V. A. Chem. Abs. Vol. 65, No. 11
Immobile Seals at High Pressures
Khim. i Neft. Mashinostr. 1966 (8), 11-13 (Russ)

Two types of seals, viscoelastic and elastoplastic, used in high pressure hydraulic tests are described. The viscoelastic seals are made from an assortment of rubbers and plastics. The elastoplastic seals are a combination of a viscoelastic seal with a bellow insert.

Supp. 148
Belitsky, M. E. ASM 18-40024.66
Investigation of Chemical Stability of Solid Lubricants at High Temperatures
Poroshkovaya Met. No. 4, April 1966, p 40-44

Oxidation resistance and thermal stability of MoS_2, ZnO, BN, silver graphite and mica were tested from 100 to 1250C in air, A, H_2 and N_2. The stability of the above materials was tested for use as solid lubricants and seals (on Ni base) for airplane turbine engines. It is determined that boron nitride filler (lubricant) can be effectively used for working temperature 800 to 900C; for temperature over 900C., the phloyopite mica can be used.

Supp. 149
Billington, I. J. Eng. Ind. 1965
Experimental Assessment of Validity Pressure Measurements in Fluid Film Between Face Seals.
Brit. Hydromechanics Research Assn. - Int. Conference on Fluid Sealing, 2nd - Proc. 1964 paper F3, p F29-44, F81-96.

Feasibility of measuring static pressures in thin fluid film between face of rotary contact seals was assessed experimentally with the use of large seal models; investigation was conducted using static pressure taps whose diameters ranged from 0.5 to 64 times thickness of fluid film; experiments established feasibility of measuring seal interface fluid pressures; measurement errors to be expected over large range of potential seal applications are shown to be negligible and conditions where larger errors should be expected are identified.

Supp. 150
Billington, I. J., Smillie, W. D., Stonehill, B. C. Eng. Ind. April 1966
Shaft Seals for Liquid Metal Applications in Space Environment
Can Aeronautics & Space Journal, Vol. 11, No. 9, November 1965, p347-65

Problem of shaft seals contain liquid metals in rotating mechanism and examination of phase diagram for liquid metal which shows that problem resolved itself into that of containing liquid and that if containing vapor, which evaporates from liquid vapor interface; it is necessary to employ two seals in tamden , one to contain liquid and the other to restrict escape of vapor and maintain adequate back pressure on liquid seal; operating principles and geometries of seals are developed; vapor seals considered include restrictive gap, and centrifugal seal; liquid seals include centrifugal and modified hydrodynamic seals.

Supp. 151
Bremner, G. F. Eng. Ind. 65
Experimental Study of Balance Conditions, Leakage Rates and Pressure Profiles for Stationary Radial - Face Seals
Brit. Hydromechanics Research Assn.- Int. Conference on Fluid Sealing, 2nd - Proc. 1964 paper F2, p F13-28, F81-96.

Seals tested were operating in pure viscous flow regime; using SAE 30 oil as working fluid and inlet pressure up to 100 psig, measurements were made for different upper seal materials, e.g., stainless steel and brass; majority of lists were conducted with fixed separating distance of 0.003 in., some results also were obtained for pressure profiles without controlled separation distance.

Supp. 152
Brown, P. F., Gordon, N., King, W. J. Eng. Ind., Sept. 66
Test Method for Evaluating Gas Turbine Engine Seal Materials
Lubrication Eng., Vol. 22, No. 1, January 1966, p7-16

Experimental rig development program was carried out by Pratt and Whitney Aircraft Division to provide optimum materials; program, in cooperative with several carbon manufactures resulted in several superior carbon grades; several stainless steel seal plates with machining and corrosion advantages were also obtained; test method employing statistically planned experiment approach is capable of reducing number of tests needed to screen seal material combination; description of old and new test methods is presented and comparison made.

Supp. 153
Cerness, Joseph, F; Strong, Douglas, H. Chem. Abs., Vol. 4, No. 13
Hard Ferrous Alloy Resisting Wear and Corrosion for Sliding Seals at High Temperatures
U. S. 3,096,565 (Cl. 29-149.5), July 9, 1963, Appl. Dec. 21, 1956 and April 10, 1959, 4pp.

Such material useful for sealed sliding service against itself or other hard materials such as Cr plate, nitrided iron, cast iron piston rings, etc. at temperatures up to 1200°F with or without lubrication, has the approximate composition. C.0.87, Mo8, Cr4, V1.9, Mn and Si each 0.25% and balance Fe. It should be useful for gas turbine oil seals, and a suitable design for such a seal of a rotating shaft bearing is described in detail.

Supp. 154
Chzhu, Go-Khua; Rumyantseu, O. V. ASM 65-08 MM-25885
Experimental Investigation of Air Tightness of Elastic Cone Seals for High Pressure Apparatus
Khim Neftyanoe Mashinostv, No. 3, 1965, p13-16

Study of specific pressure necessary to produce air-tight conditions in bevel type seals of 45 carbon steel; reference to pressure medium (air helium), hardness of seal surfaces, surface finish of contacting surfaces, bevel angle of the seals and their diameter and width. Application of the cone type seals for high pressure apparatus and gas compressors. 4 refs.

Supp. 155
Coopey, Walter — Chem. Abs. Vol. 63, No. 13
New Seal for Super-pressure Shafts
Chem. Eng. 72 (20), 153-4 (1965) (Eng.)

A new method for sealing rotating shafts against very high pressure involves the use of thin lubricant films between balanced-pressure packing elements.

Supp. 156
Dega, R. L. — Eng. Ind. August 1966
Selecting Materials to Cope With Corrosive Properties of Seal Applications
ASME Paper 66-MD-9 for meeting May 9-12, 1966, 7p

Corrosion is major factor of wear of lip type sealing element and shaft; this corrosion is electromechanical in nature occurs in presence of high humidity conditions, and can be accelerated by heat; almost all elastomers and other organic seal materials produce corrosion in varying degrees, and process is only slightly inhibited by application of lubricants to sealing areas; numerous product failure occur in storage since they corrode at sealing face and will be immediate failures when put in use; various known causes for this corrosion phenomenon are discussed in detail.

Supp. 157
Dolman, R. E. — IAA 66-30407. 16-2711
Some Special Consideration Necessary to Achieve Optimum Seal Performance
American Society of Lubrication Engineers, Annual Meeting, 21st, Pittsburgh, Pa., May 2-5, 1966, Paper 66AM2C2. 13p.

Examination of the hazards that may contribute to seal failure, with special emphasis on those too often overlooked or not recognized. These include dirt, misalignment, distortion, shaft deflection, imperfections, excessive face load, and dry operation. Considerations that are important to achieve optimum seal performance are cleanliness, careful alignment of the fixed face, cooling lubrication, gasketing, shaft deflection and end play, balance, and arrangement. The advantages and limitations of external and internal seals are reviewed.

Supp. 158
Fedorchenko, J. M., Denisenko, E. T. and Miroshnikov, V. N. — ASM65-08, MO9-26682
Study of the Change in Mechanical Properties of Sealing Materials in the Oxidation Process, Pt. 2

Poroshkovaya Met., No. 4, 1965, p57-60

A study of the effects of oxidation on two mechanical properties and running-in of sintered sealing material on a Ni base showed that the hardness of the material should not exceed 60 units H_B. Oxidation substantially affects the hardness and strength of the materials. The rise in hardness during oxidation is directly connected with the degree of oxidation of the material and the change in strength depends on the nature of the oxidation.

Supp. 159
Ferreira, L. E., Briggs, D. D.
Aluminum Oxide and Beryllium Oxide Ceramics - Seal Materials of the Future
Society of Automotive Engineers, 1965, p153-155

Description of the temperature dependence of various physical properties of alumina and beryllia ceramics along with some important chemical and nuclear consideration. Properties in the cryogenic temperature regions are stressed. The data presented will be a composite of information gained in the laboratory and from literature sources on cryogenic properties of alumina and beryllia material.

Supp. 160
Field, W. E. — IAA 66-40285. 22-3927
Selection of Optimum Seals and Sealant for Spacecraft
Joining of Materials for Aerospace Systems; Society of Aerospace Material and Process Engineers, National Symposium, 9th, Dayton, Ohio November 14-16, 1965, Papers

This paper is concerned with the joints and connections in a spacecraft or spacecraft booster, which must be separable for access, repair, or other purposes. Aerospace vehicles are designed with a minimum number of such joints because of the following problems: (1) the consideration involved in selecting seals and sealants which will reliably prevent leakage during exposure to spatial environments, and (2) the undesirable weight factor associated with flange and reinforced joints. Presented is an analytical approach to the selection of spacecraft sealing system. The materials and design considered lead to high sealing efficiency to weight systems which are ideally suited to perform in a space environment.

Supp. 161
Froessel, W. — Eng. Ind. December 1966
Untersuchung von Gewindewellendichtungen
Konstruktion, Vol. 18, No. 4, April 1966, p152-9

Investigation of threaded seals; theoretical and experimental study is made of threaded seals used for high speed shafts; seals with rectangular, trapezoidal and tapered threads of various depth, width and clearance are considered and effect of these parameters on seal thighness is examined; advance effect of oil-entrained air bubbles on seal tightness; means are suggested for preventing oil foam formation at high shaft speeds. In German.

Supp. 162
Goldstein, Marcy, B. — IAA 65-35191.23-3449
1000 + C Ceramic - Metal Seal for Spaceborne Reactors
Space/Aeronautics, Vol. 44, September 1965, 100-2-4-6-8

Development of a ceramic seal that can provide continuous, reliable operation at more than 1000°C, resist high temperature corrosion in cesium vapor, and resist severe thermal cycling. It uses columbian as its metal component, a 97% aluminum oxide as the ceramic component, and pure vanadium as the brazing filler metal. It was found that the theoretical "rule" in ceramic-to-metal seal design, that a metal member must be selected to the thermal characteristics of which closely match those of the ceramic member, need not be applied. Instead, it was found that a low yield metal is preferable; hence the choice of columbium. Vanadium brazing techniques are discussed, and it was found that brazing can be done at a higher temperature than that at which the ceramic was sintered.

Supp. 163
Golubiev, A. I. — Eng. Ind. January 1966
Studies on Seals for Rotating Shafts of High-Pressure Pumps
Wear-Usuer-Verschleiss
Vol. 8, No. 4, July-August 1965, p270-88

Results of studies of mechanical seal at water pressure of 100 Kg/SQ. cm. and shaft speed of about 3000 rpm carried out at Soviet Research Institute for Hydraulic Machines are described: design of seal ensured hydraulic balance. Best friction couple was established, friction couple parts were protected against deformation, and optimum finish of friction surface was found; results of development and investigations of screw-type labyrinth seals intended for shaft rotating at high speeds are reported; hydraulic calculation method of screw type labyrinth seals is presented and results of calculations compared with results of experimental studies of such seals.

Supp. 164
Goree, W. S., McDowell, B., Scott, T. A.
Seals For Low Temperature High Pressure System
Review of Scientific Instruments, 36, 99-101, January 1965

Description of: (1) plug seal in which the mating surfaces are wedge-shaped with a layer of indium metal between them. (2) electrical seal in which tubing is used with epoxy. (3) seals for 10k bar piston - cylinder apparatus, using phosphor bronze, half hard beryllium copper and indium.

Supp. 165
Hamilton, D. B., Walowit, J. A., Allen, C. M. — IAA 66-14244.04-0527
A Theory of Lubrication by Microirregularities
American Society of Lubrication Engineers and American Society of Mechanical Engineers, Lubrication Conference, San Francisco, California, October 18-20, 1965, Paper 65-Lub-11, 9p, 10 refs.

Descriptive of a theory of liquid lubrication applicable to parallel surfaces such as the surfaces of a rotary-shaft seal. The lubrication mechanism presented is based on surface microirregularities and associated film cavities. Closed-form analytical solutions are obtained giving load capacity as a function of speed, viscosity, and surface asperity dimensions. The theoretical results agree qualitatively with load capacity determined experimentally for three asperity distributions.

Supp. 166
Heinz, Heineniann; Kraus, John, W., Marshall, L. — NSA66, Vol. 20, No. 5209
Method of Sealing Refractory Vessel Containing Radioactive Wastes
U. S. Patent 3,213,031. October 19, 1965

A method is described for sealing a refractory vessel containing a solid fixation mixture of radioactive component and non-radioactive components. The fixature mixture is fed through an entrance port of a vessel. After the entrance port is sealed, the vessel is submerged in a sealant mixture contained in a second larger container. The sealant mixture is made up of oxides of Cr, Co, and Fe and an elemental metal selected from a group consisting of Al, B, Zr, and Si. The sealant mixture is heated to its ignition temperature to provide a seal around the vessel that is resistant to erosion and thermal degradation.

Supp. 167
Hudelson, John, C. — IAA66-30408.16-2711
Dynamic Instability of Undamped Bellows Face Seals in Cryogenic Liquid
N.A.S.A. Lewis Research Center, Cleveland, Ohio. NASA-TN-D-3198
American Society of Lubrication Engineers, Annual Meeting, 21st, Pittsburgh, Pennsylvania, May 2-5, 1966, Paper 66AM2C3, 20p

The primary motions of an undamped bellows face seal are described along with the displacements and frequencies of the motions experienced during dynamic instability. Subminiature accelerometry were attached to the nose piece of a small bellows seal and the accelerations and frequencies recorded on magnetic tape for analysis at an expanded time factor. The results of the tests indicated that dynamic instability will occur in undamped bellows face seals operating in a cryogenic environment and be of such magnitude as to damage the sealing surface causing excessive leakage. Variations in the pressure difference across the seal and in the contact load between the seal nosepiece and the mating surface did not change the characteristics of the dynamic instability. The primary motions of dynamic instability are diametral rocking and torsional oscillation.

Supp. 168
Jackson, B. S.; Akirman, H. E.; Rigg, A.; Welch, M. J. — Chem. Abs. Vol. 65, No. 12
The Effect of Atomic Radiation on Mastic Sealing Compounds
Rubber J. 148 (8), 48-50 (1966) (Eng.).

The constant rate penetrometer is used to determine the effect of atomic radiation on mastic sealing compounds. Sealants were exposed to 10^6r./hr. of γ-radiation at $\leq 25^\circ$ and compared with controls. Bitumen-based sealants were the most resistant to radiation. Oil-based mastics showed highly accelerated aging. Butyl-based mastics showed a viscosity decrease but polybutene mastic showed a viscosity increase.

Supp. 169
Jackson, B. S. Chem. Abs. Vol. 65, No. 12
Making Use of Rubber-Like Materials in Building Sealants
Rubber J. 148 (9), 46-7, 50-2, 54 (1966) (Eng.).

The specifications of sealants are reviewed and the advantages and disadvantages of rubber-like materials are discussed. The types of sealants and auxiliary materials are discussed as well as use of sealants in curtain wall construction.

Supp. 170
King, Alan E. IAA65-31122. 19-2817
Screw Type Shaft Seals for Potassium Lubricated Generators
IEEE Transactions on Aerospace Vol. AS3, June 1965, Supplement, p471-479. 7refs.

Development of liquid and vapor screw seals to separate the bearing and rotor cavity in a potassium-cooled and lubricated space electric power generator to the point where ground test application is possible. Individual tests were made in potassium-simulating fluids to determine seal performance at laminar and turbulent conditions. Liquid seal performance was verified in 375°F potassium. Vapor seal performance concurred with analysis of a simplified flow model. Remaining problems associated with adaption of these seals to an actual generator are discussed.

Supp. 171
Koulnik, Rodnick J. Chem. Abs. Vol. 65, No.12
Ball Valve with Teflon Seals for Cryogenic Fluids
U.S. 3,276,737 (Cl. 251-171), October 4, 1966, Appl. September 30, 1963, 4pp.

Cryogenic fluids can be controlled by a ball valve containing frustoconically shaped Teflon seals.

Supp. 172
Lambert, Robert L., McKeirnan, William R. Chem. Abs. Vol. 64, No. 9
Self-Supporting Films for Vacuum-Tight Seals
U.S. 3,227,591 (Cl. 156-89). January 4, 1966, Appl. April 26, 1963; 4pp.

A self-supporting film is prepared from a suspension consisting of an organic binder and a binder plasticizer dissolved in a mixture of volatile

organic solvents in which powdered particle additives are homogeneously dispersed. The additives are chosen for their ability to bond to the designed surface; they must be either sinterable, fusible, or able to flow when heated, preferably at below the deformation temperature of the materials which are to be bonded, and must solidify and bond upon cooling. The process is suitable for any of the usual types of seal depending on the aditives dispersed in the film. It may be used to bond ceramic to metal, glass to metal, metal to metal, and ceramic to ceramic.

Supp. 173
LeFave, G. M. Chem. Abs. 63-5422
Elastomens For Use In Cryogenic Environments
Material Symp., National SAMPE (Soc. Aerospace Mater. Process Engineers) Symp., 7th, Los Angeles, 1964 (13), 14pp.

The tensile strengths of polyester urethans between 70° and 250° K. is less than that of Teflon, nylon, or Mylar. Polyester urethans have higher impact strengths than polyether urethans.

Supp. 174
Lyubchanskaya, L. J.; Tsopina, N. A.; Kuzminskii, A. S.; Chem Abs. Vol. 64 No. 1
Relations Between Chemical Relaxation of Stress and Loss of Seal Efficiency
Kauchuk i Rezina 24(9), 13-16(1965) (Russ)

An apparatus was constructed for the study of the dependence of the contact stress on the hermeticity of a vacuum system. The effect of various factors on the critical contact stress was also studied. Elastomers differing in the type of polymer, vulcanizing groups, fillers and degrees of filling were tested.

Supp. 175
Manferdi, D. V. NSA 65-40718
Summary of Shaft Face Seal Development Program
Pratt and Whitney Aircraft, Middletown, Connecticut. Connecticut Advanced Nuclear Engineering Lab, June 30, 1965. Contract AT(30-1)-2789, 78p.

A program for the development of a sealing system with a minimum operating life of 10,000 hours for shafts of pumps used to circulate Li, K, on NaK at 1600°F is described. The shafts supported by oil-lubricated ball bearings were to operate up to speeds of 8500 rpm. The purity requirements of the cover gas above the liquid metal were 2 ppm or less of O_2 or moisture. The sealing system consisted of two oil cermet face seals, one dry gas cermet face seal, a centrifugal dynimac seal, and two sweep gas systems. Seal manufacturing techniques, performance data, and experimental techniques evolved in the program are reported. A total of over 150,000 hours of testing was accumulated in the overall program.

Supp. 176
Mauri, R. E. IAA65-22746
Materials for Sealing Applications
Space Materials Handbook, Addison-Wesley Publishing Co., Inc., 1965, p321-366. 26 ref.

Discussion of the properties of, and effects of the space environment on, elastomers and plastics used for seals in space vehicles. The use of such seals are described and include the sealing of cabins, fluid storage and distribution systems, and electronic-component housings. The environment effects on elastomer, plastic, and composite sealing materials are reviewed, and include effects associated with liquid propellants and hydraulic fluids, penetrating radiation, temperature extremes, and high vacuum.

Supp. 177
McDowell, Donald J. IAA66-13781
Oil Seals - Performance and Materials
Annual National Conference on Fluid Power, 21st, Illinois Institute of Technology, Chicago, Illinois, October 21-22, 1965. Paper. 11p

Examination of the performance characteristics and construction materials for the lip seal, a radial load sealing device for rotating shafts. The three basic components of a typical lip seal--metal core, spring, and sealing element--are reviewed, and the major requirements for each are discussed. According to the modern concept of how a lip seal performs, a thin rubber element moves in response to shaft eccentricity and frictional drag and is separated from the shaft by a thin film of oil. Four categories of material are considered: NBR or Buna N elastomers, polyacrylates, silicone elastomers, and fluorocarbons. Stress-strains data considered to be significant in lip performance are graphically presented.

Supp. 178
McGrew, J. M.; McHugh, J. D.
Analysis and Test of the Screw Seal in Laminor and Turbulent Operation
ASME Trans. Vol. 87, Series D, Nov. 1965, pp 155-62

An analytical and experimental study of the screw seal in both laminor and turbulent operation is presented. The various scattered analyses of laminor screw seal operation are evaluated in comparison with one another and with a simplified flow model. The basic operations are extended to cover turbulent flow. Experiments are described in which the screw seal was tested in both laminor and turbulent flow regimes. Finally, the breakdown of sealing capability under certain conditions is examined and reasons postulated for its behavior.

Supp. 179
McKannan, E. C.; Gause, R. L. NSA66 Vol. 20, No. 23794
Effects of Nuclear Radiation and Cryogenic Temperatures on Nonmetallic Engineering Materials
J. Spacecraft Rockets, 2: 558-64, (July-August, 1965).

Engineering data on the changes produced in the physical properties of nonmetallic materials by exposure to the various combinations of cryogenic temperatures, nuclear radiation, and vacuum are being sought. To date, 80 materials from the following functional categories were evaluated in one or more of these combinations of environments; structural adhesives, structural laminates, thermal insulations, electrical insulations, potting compounds, seals, sealants, lubricants, and thermal control coatings. The effects of cryogenic temperature and radiation on some of these materials are discussed.

Supp. 180
Miller, Forbes M. Chem. Abs. Vol. 65, No. 10
Porous Compressible Metallic Powder Bodies for Sealing Clearance Spaces
U.S. 3,268,997 (Cl. 29-420) August 30, 1966, Appl. May 14, 1963; 6pp

Such powder compacts suitable for preventing leakage between the moving and stationary parts of pumps and gas turbines are formed as coatings sintered on a roughened firm metallic backing and are composed of any alloy powder such as bronze, inconel, or stainless steel. A compatible liquid binder is used for spraying the powder mixture on irregular surfaces. The compact can be machined to fit the clearance space which is to be sealed.

Supp. 181
Murphy, John C. IAA66-30455.16-2712
Design and Construction of 10' Diameter Bakeable UHV All Metal Seal
Institute of Environmental Sciences, Annual Technical Meeting, San Diego, California, April 13-15, 1966, Proceedings

Description of a large-diameter metal seal configuration designed for ultrahigh vacuum service and incorporating bakeable characteristics. The seal described is distinguished by the fact that of the two mating flanges the head flange has considerable stiffness and essentially remains constant after fabrication while the vessel flange, comparatively weak in bending, conforms to the shape of the head flange by the lead tongue provided and essentially rounds itself to mate to the head flange.

Supp. 182
Novell, G. L. NSA65 Vol. 19-40702
Water Tests of High Speed Centrifugal Dynamic Shaft Heads
Pratt and Whitney Aircraft, Middletown, Connecticut, Connecticut Advanced Nuclear Engineering Lab., June 23, 1965, Contract AT(30-1)-2789.

Current higher speed liquid metal pumps under development require smaller diameter seals with minimum power consumption. Four models were tested in water at shaft speeds up to 13,000 rpm. A 36 vane dynamic seal, designated DS-11, with a vane height of 0.200 inch, an axial vane clearance of 0.010

inch, and two circumferential tang slots exhibited the most stable liquid-to-gas interface at high speeds and provided optimum values of vane and torque coefficients.

Supp. 183
Paxton, R. R., Shobert, W. R. IAA65-28033.17-2481
Carbon-Graphites for Aerospace Seals
Society of Automotive Engineers, 1965, P 149-152

Discussion of the special requirements for carbon-graphite seal components in aerospace applications. Carbon-graphite materials are divided into five major groups with the unique properties of each group listed. The influence that environment, operating conditions, and seal design have on the proper grade selection is described. Special materials are required to obtain reliable seal performance with high vacuums, cryogenic fluids, and high temperature gases. Suggestions for handling and using these materials are included.

Supp. 184
Penslee, R. L. Eng. Ind., September '66
Compressible Metal for Turbine Seals
Metals Eng. Quarterly, Vol. 6, N 1, Feb. 1966, p 55-8.

Discussion of Nicroseal coating designed to provide seal between moving and stationary parts that would also permit dimensional changes of rotating members without eventual destructive contact; seal is diffusion welded coating used to seal clearance between rotor and adjacent stationary parts; examples of applications in automotive electric power and aircraft industries are given.

Supp. 185
Powers, J. V., Jr. IAA65-32125.20-2968
A Rotatable Sealing Coupling for Cryogenic Uses
Advances in Cryogenic Engineering, Vol. 10, Sections A-L. Proceedings of the Cryogenic Engineering Conference, University of Pennsylvania, Philadelphia, Pa., August 18-21, 1964. New York, Plenum Press, 1965, p 458-463

Description of a new type of rotatable sealing coupling. The coupling permits full 360° rotation at varying rotational velocities in a single plane. Employing this coupling allows the test engineer to supply, as an example, liquid nitrogen to a spacecraft while it is rotating and under test. It also allows the design of thermal conditioning shrouds with 360° rotatable shutters and spacecraft positioners with thermally controlled, movable parts. The seal is effective under pressures of less than 10^{-6} ton.

Supp. 186
Prince, William A.
Bimetallic Seal Solves Cryogenic Sealing Problems
Hydraulics & Pneumatics, Vol. 18; 105-9, Nov. 1964

Maintaining seal compression gets tougher at extreme temperatures. The bimetallic seal uses dissimilar materials with different contraction coefficients to solve this problem. The bimetallic seal combines the principles of the ring spring and the temperature-actuated seal. Spring, pressure, and temperature vectors for the seal can be assigned according to needs.

Supp. 187
Reitdijk, J. D. et al
Rolling-Diaphragm Seals for Large Pressure Differences
Mechanical Engineering, Vol. 88; 54-55, Oct. 1966 (from Engineers Digest, England)

Rolling diaphragm showing method of attachment to a piston and cylinder wall for high-pressure seals by supporting the diaphragm on a cushion of fluid and with a stepped system of fluid support.

Supp. 188
Shkurupii, G. I.; Mamon, L. I.; Lokshin, M. A. Eng. Ind., Sept 1966
Investigation of Hydrodynamic Pressure in Clearance Between Friction Elements
Chem. & Petroleum Eng. (English translation of Khimicheskoe Neftyanoe Mashinostroenie) N 3, March 1965, p 215-18

Tests at various pressures due to sealed kerosene and oil to study causes and distribution of hydrodynamic pressure as function of liquid pressure and specific pressure; distribution curves for clearance between two friction elements show that in certain service conditions, liquid film is retained and prevents escape of working liquid; thickness of this film depends on viscosity of liquid.

Supp. 189
Snapp, R. B. Eng. Ind. 1965
Theoretical Analysis of Face-Type Seals with Varying Radial Face Profiles
ASME Paper 64-WA/LUB-6 for meeting Nov. 29-Dec. 4, 1964, 8 p.

Analytical study of thin fluid films between sealing surfaces of face type seals is presented for six types of radial profiles--parallel, linear converging, linear diverging, and thiner types of parabolic faces. Equations are developed to analyze effect of these radial contours on pressure profile, fluid force on seal face, film thickness, and leakage rate; graphic comparisons of numerical solutions are presented for cases analyzed.

Supp. 190
Symons, J. D. Eng. Ind., Sept. 1966
Seal Design Parameters and Their Effect on Seal Operating Temperature
ASME Paper 66-MD-10 for meeting May 9-12, 1966, 12 p.

Aim of study was to determine optimum design parameters for radial oil-seal lip; effects of variation in trim diameter, radial lip force, eccentricity, spring position, contact width, flex-section thickness, and material on

seal operation temperature were studied by means of fractional factorial experimentation; in general, reduction in temperature as a result of change in given variable also resulted in reduction in seal leakage; seal performance and life can be improved by controlling various seal design parameters so as to obtain lower operating temperature.

Supp. 191
Thayer, J. H. Eng. Ind., Oct. 1966
Construction of Mechanical Seals
Matis Protection V 5, N 3, March 1966, p 67-70

Construction of mechanical seals with emphasis on selection of proper construction materials for various seal parts in relation to operating conditions to which they are subjected; materials commonly used for component parts of seal are brass, bronze, steel, stainless, steels, monel, and hastelloys; selection of materials depends on their wear resistance and their ability to stand against mechanical, thermal, chemical and radiational forces.

Supp. 192
Watson, R. D. NSA65-19-26489
Effect of Seal Ring Deflection on the Characteristics of Face-Type Mechanical Shaft Seals in High Pressure Water
(AECL-2242) Oct. 1963 18 p. Atomic Energy of Canada Limited

The effect of seal ring deflection on the characteristics of face-type seals was investigated by assuming that the seal rings deflected in the following ways: the deflection varied uniformly across the face or the deflection at any point across the seal face was proportional to the pressure at that point. Theoretical curves were developed, which show the variation in balance ratio and leak-rate with film thickness for different deflections of the seal ring. The effects of seal ring size and fluid viscosity on the seal characteristics were also considered. Symmetrical deflection of the rings of a face-type mechanical shaft seal is necessary to provide stability. To be useful, the deflection must allow a wedge of water to form between the faces. Seals with absolutely flat and parallel faces are unstable. Wear and erosion are believed to be partly responsible for seal failure since they produce an out-of-balance condition, which in turn leads to over heating and more wear.

Supp. 193
Weinand, L.; Wroblewski, R. C. Eng. Ind., Dec. 1966
Principles of Hydrodynamic Sealing
Rubber World, V 154, N 1, April 1966, p 79-83

Newly designed "hydrodynamic" seal combines pumping action with sealing lip to stop virtually all leakage while allowing more tolerance in manufacturing and design variables; hydrodynamic seal generate positive force in balance. Leakage flow forces are capable of sealing high pressures, have infinite life since shaft and housing do not contact, consume less power than contact seals, are low in cost, easy to manufacture, and ideal for high-speed applications.

Supp. 194
Williams, John G.
Shaft-Seal Systems for Large Power-Reactor Pumps
Nucleonics, Vol. 23, No. 2, pp 49-53, February 1965

Most new reactors will have shaft-sealed pumps that are 10-15% more efficient and 40% less expensive then canned pumps. Many kinds of shaft-seal systems are competing for this application.

Supp. 195
Wills, Stanley Charles Frederick NSA66-20-12949
Improvements In or Relating to Sealing Means for Providing a Seal Between Two Relatively Movable Parts
British Patent 1,013,683, December 15, 1965

A sealing means is provided for use between two relatively movable parts that are subjected to a pressurized fluid tending to escape between the parts, e.g., the assembly of elements on a reactor standpipe. The sealing means includes a piston and cylinder assembly with mutually opposed steps in their cylindrical surfaces. A sealing ring groove is located on one of the cylindrical surfaces so that a ring located in the groove makes a face seal when brought into contact with the apposed mutual step by relative movement of the piston and cylinder and a piston seal with the surface of the cylinder when not in contact with the step on the cylinder.

Supp. 196
Schnetzer, E. (ed) NSA66-20-18881
Two-Stage Potassium Test Turbine
General Electric Co., Cincinnati, Ohio, Quarterly Progress Report No. 13, May 8-August 8, 1964, Contract NAS5-1143, 142 p. (NASA-CR-54211) CFSTI

Two-stage test turbine being developed for operation in a saturated potassium vapor at 1600°F underwent its first potassium vapor test. Efforts were continued to identify and eliminate the oil leakage problems. This was successfully accomplished by minor changes in component design and improvement of static seals. Materials support involved testing of sealing ring configurations, testing the potassium resistance of the fluorinated hydrocarbon sealing material, Viton.

Supp. 197
Takamori, Takeshi; Tamozowa, Minoru
Behavior of Interlayers of Glass-to-Tungsten Seals
American Cermanic Society Journal, V. 48:405-9 Aug. 1965

The relation of the stability of glass-to-tungsten seals to glass composition was studied. Colors of interlayers in the seals depended on alkali content of the glasses. The interlayers were identified, by X-ray diffraction, as WO_2, W_{18}, O_{49}, and alkaline tungsten bronzes. Differences in the thermal expansion of these substances were considered with respect to the seal strength. Sodium and potassium-free glasses did not form tungsten bronze at the interface, and seals between such a glass and tungsten were more durable than seals between an ordinary alkali-containing glass and tungsten. An accelerated procedure for detecting potential leaks in seals was tested. Effects of heat treatment on colors of sealed specimens were also observed.

Supp. 198
Yeremenko, V. N.; Lesnik, N. D. ASM 65-09,09-29000
Possible Use of Sintered Packings With Low-Melting Liquid Filler For Vacuum Systems
Paper from "Trudy VII Vses Nauchntakkn Konf Po Por Met Erevan" 1964 p 207-213

Value packings of porous metal liners were made from Fe, Ni, Cu, Mo, W, and stainless steel powders by impregnating with an alloying filler with a melting point from 400 to 650C. Specimens with 67-70% porosity were sintered in vacuum or in H_2 for 1 hour at 1100 to 1200C and impregnated in vacuum at 650 to 700C for 3-5 minutes.

Supp. 199
Zadorskil, V. M. Eng. Ind., July 1966
Pump Shaft Seals
Chem. & Petroleum Eng. (English translation of Khimicheskoe Neptyaoce Mashanostroence) N 3, March 1965, p 231-2

Development of various synthetic rubber end seals for pump delivering water divinyl, methylstyrenes, and corosive and abrasive products. Introduction of Teflon provides as filler, reduces swelling of rubber wedge gasket; replacement of stuffing gland of pump by combined end-helical packing in which sealent along shaft is tapered Teflon ring, provided much longer service; internal end seal using Teflon membrane gives excellent results in service in abrasive medium.

Supp. 200
Anonymous NSA21-67 3225
Loads to Deform Elastomer O Seals in Grooves and Between Plain Flanges
Atomic Weapons Research Establishment, Aldermaston, (England), October 1966, 27 pp. (AWRE-0-82/66)

The loads necessary to deform O seals in various types of grooves and between plain flanges have not been extensively investigated. As there is a recurrent demand for this information, a series of practical tests was carried out. The results of 210 tests are presented on a nomogram.

Supp. 201
Anspach, W. F. Eng. Ind., July 1967
Hydrofluorocarbon High Temperature Integral Fuel Tank Sealants
Rubber Chem. & Technology, Vol. 39, No. 4, Pt. 2, September 1966, pp. 1200-1214

Interim solution to critical fuel tank sealant problem in March 3 aircraft; high temperature sealant based on mixture of high and low molecular weight hydrofluorocarbon polymers has solids contents of 85% and cures at moderate temperatures; it is one-component system stable for several weeks at ambient temperatures; sealant exhibits good room temperature and high temperature. Mechanical properties both before and after aging in hydrocarbon fuel and fuel vapor at temperatures up to 500F adhesion to metal substrates is good.

Supp. 202
Appleton, B. Eng. Ind., January 1967
Design Considerations for Polyurethane Sealing Devices
Nat'l Conference on Fluid Power, Proc., Vol. 19, 1965, pp. 79-83

Designer must examine carefully what is ultimately required of seal and what fringe benefits seal material can yield to product to which it is to be fitted; chemical and chemical properties required in seal, cost of material and cost of processing must be considered. Types of seals made from polyunethane resins include: O-rings, U-cups, V-rings, flange packings, cup packings, rod wipers, scrapers, bellows, diaphragms, and special balljoint and bearing seals.

Supp. 203
Bazzarre, D. F., Spain, R. G., Withers, J. C. Eng. Ind., July 1967
Metal Clad Elastomeric Cone Seals for Liquid Rocket Propellants
Products Finishing, Vol. 31, No. 1, October 1966, pp. 66-77

Development of seals resistant to storable liquid rocket propellants was motivated by two facts, i.e., (a) that current storable propellants differ widely in composition and necessitate elastomeric vulcanizater to secure reasonable compatibility, and (b) that propellants are new fluids and existing elastomers are eliminated rather quickly as elergetics of these fluids advance; fabrication and preliminary evaluation of limited quantities of metal clad O-rings, using variety of metals in thickness of 1 to 10 mils; determination of techniques for preparing conductive surface on O-ring and application of metals by electroplating; evaluation of metal clad O-rings in N_2O_4 and in acetone at 75F.

Supp. 204
Bellanca, C. F., Salyer, J. O. Eng. Ind., July 1967
Effect of Liquid Rocket Fuels and Oxidizers on Elastomeric O-Ring Seals
Rubber Chem. and Technology V. 39, No. 4, Part 2, September 1966 pp. 1215-1221

Tests of selected elastomeric and compliant materials were made at 73F with N_2O_4, ClF_3, and 90% H_2O_2 (oxidizers) and with mixed hydrazines, pentaborane and hybaline A-5 (fuels); at 73F only polytetrafluroethylene was resistant to all fluids; butyl and ethylene propylene rubber appear satisfactory for long-term storage in presence of mixed hydrazines and hybaline A-5; fluorocarbons appear best suited for use with pentaborane and hydrogen peroxide.

Supp. 205
Billington, J. J., Toronchuk, J. P. Eng. Ind., December 1967
Characteristics of Self Energized Hydrostatic Seals for Rotating Shafts
Eng. Inst. Canada Trans. Paper EIC-66-MECH 7, May 1966, 10 pp.

Theoretical and experimental study of controlled leakage shaft seals with fixed gap maintained between rotating and stationary members is described, fixed gap is by self-energized hydrostatic actuators; effect of various geometrical parameters on seal performance was investigated theoretically; good correlation was obtained between experiment and theory when operating in low viscosity fluid (water); effects of pressure-induced seal face deformation and effects of thermal transients were studied.

Supp. 206
Billington, J. J., Fitzsimmons, T. E. NSA67-21 17432
Final Report on Shaft Seal Development

A coolant pump shaft seal development program is summarized; particular reference is made to the final endurance test series on hydrostatic self-energized, finite clearance seals. The endurance program resolved the seal geometry and materials.

Supp. 207
Bupara, S. C., Walowit, J. A., Allen, C. M. IAA67-22705 10-15 1658
Gas Lubrication and Distortion of High Pressure Mainshaft Seals for Compressors
B.H.R.A., Int. Conf. on Fluid Sealing, 3rd; Cambridge, England, April 1967, Paper F3, 24 pp., 7 ref.

Theoretical analyses and static experiments have been performed for high-pressure, gas lubricated face seals of the type used for mainshaft seals for jet-engine compressors. The analyses consider the combined effects of seal-element distortion and gas flow through the seal interface.

Supp. 208
Davies, A. R., O'Donoghue, J. P. Eng. Ind., June 1967
Lubrication of High-Pressure Face Seals
ASME-Paper 66-WA-LUB-7 for Meeting November-December 1, 1966, 8 pp., 12 Refs.

Theoretical analysis of lubrication conditions of hp face seal is presented which uses very simple treatment for estimating effects of elastic and thermal distortion. Agreement between theory and experimental results obtained for carbon seal is good; from results obtained, it is concluded that there is inherent stiffness in seal and this enables leakage to be controlled by load clamping seal faces together,. Effect of speed on temperature and problems of hydrodynamic pressure generation under seal face are discussed; it seems that from order of pressure used (1000 lb. ft./sq. in.) hydrodynamic effects are not of major importance.

Supp. 209
Decker, O. Eng. Ind., September 1967
Advances in Dynamic Seal Technology
ASME-Paper 67-DE-50 for Meeting May 15-18, 1967, 16 pp.

Need for advanced seal concepts has been brought about by increasingly severe requirements imposed by applications such as supersonic transport, dynamic space power machinery, nuclear applications cryogenic machinery, and liquid-metal technology; conventional sealing techniques prove inadequate to meet challenges improved by such applications; recent trends and technological developments are solving advancement sealing problems; some dynamic seals are described in applications employing liquid metals, gas, and move standard organic fluids for both ground, aerospace and space turbomachinery.

Supp. 210
Decker, O. IAA67-28881 14 2328
Advances in Dynamic Seal Technology
ASME, Design Engineering Conf. & Show N.Y., May 1967, Paper, 67-DE-50, 17 pp., 21 Refs.

Description of recent trends and technological developments in solving advanced sealing problems. A variety of dynamic seals is described in applications employing liquid metals, gas and more standard organic fluids for ground, aerospace and space turbomachinery.

Supp. 211
Dickenson, R. W. NSA21-67 14356
Bearing and Seal Requirement for Large Liquid Metal Cooled Reactor Systems
Atomics International, Canoga Park, Calif., MTI-66TR66, pp. 8, 1, 1-6

The requirements of bearings and seals in pumps for high power output liquid-metal cooled reactors are outlined. Present and potential uses of sodium lubricated bearings are described. The need for a reliable, economical seal for mechanisms in sodium reactors is emphasized.

Supp. 212
Eibert, P. NSA21-67 8002
Nuclear Reactor Coolant Circulator with a Liquid Shaft Sealing
To Siemeas-Schuckerverke Aktiengesellschaft German Patent 1, 218, 076, June 2, 1966, Filed September 29, 1962

The sealing comprises an annular gap portion between the shaft and the housing filled with a sealing liquid, and a second gap portión filled with a sealing gas, the latter portion being located adjacent to the interior of the reactor vessel. In the middle of the annular portion which is filled with gas there is a chamber connected to a storage tank containing clean reactor coolant gas. The liquid sealed portion and the gas sealed portion are separated by another chamber connected to the interior of the housing, into which the sealing liquid and the sealing gas escape and which is provided with an exit through which the gas-liquid mixture is drawn off.

Supp. 213
Gerlach, T. Eng. Ind., January 1967
Self-Adjusting Seal with Fluid Friction
Polska Akadenia, Prace Instytutu Maszyn Prezeplywouych, No. 22, 1965, pp. 15-33

Self-adjusting seal, with constant clearance between seal elements independent of pressure and fluid viscosity, enabling frictionless transfer of dynamic loads in bearirg test rigs; equation of equilibrium of sealing element is analyzed; dynamic properties of seal are examined for constant load and for periodically changing load occuring in test rig operation; sufficient conditions for attenuation of vibrations are determined. Results of experimental tests verify theoretical assumptions. In Polish.

Supp. 214
Green, J., Levine, N. B., Sheehan, W. Eng. Ind., July 1967
Elastomers Resistant to Rocket Propellants
Rubber Chem. & Technology, Vol. 39, No. 4, Pt. 2, September 1966, pp. 1222-1232

Resistance of polymers to hydrazine - type fuels, hybaline A-5, nitrogen tetroxide, and fluorine - containing oxidizers; butyl and ethylene propylene rubber are found compatible with hydrazine; carboxy-nitrose rubber is recommended for long-term exposure to N_2O_4 at 165F; fluorocarbon plastics and silicone rubber are most suitable for use with oxygen difluoride, chloryl fluoride-tetrafluoro-hydrazine mixture; teflon, Kel-F81, and uncured nitrose rubber gum are suitable for limited applications in chlorine trifluoride.

Supp. 215
Halliger, L. Eng. Ind., June 1967
Sealing of Anti-Friction Bearings
T2 Fuer Praktische Metallbearbeitung, Vol. 60, No. 4, April 1966, pp. 207-218

Important features and properties of common shapes of protective seals are described and factors to consider in their selection indicated; development of grease-charged bearings with built-in seals for maintenance-free bearings; examples of seal combinations which proved satisfactory under plant conditions. In German.

Supp. 216
Hamaker, J. Eng. Ind., January 1967
New Materials in Mechanical Face Seals
Nat'l Conference on Fluid Power, Proc., Vol. 19, 1965, pp. 58-65

Fundamentals of seal operation, basic seal types, available and advantages and limitations inherent in seal's design and materials of construction are discussed; special emphasis is given to materials introduced in recent years; materials for secondary seals and seal rings.

Supp. 217
Hawkins, R. J. NSA21-67 8007
Improvements in or Relating to Sealing Means
To Babcock and Wilcox, Ltd., British Patent 1,045,910, October 19, 1966, Filed September 11, 1964

A method is described for making nuclear reactor seals by means of which a flow path for coolants, from the reactor core to heat exchangers, is formed. Ducts which provide for circulation of coolants are given a sealing contact by sliding blocks of graphite. The blocks are held in place, against movement caused by thermal expansion, by springs, pins, or toggle arms, but may be moved vertically or horizontally.

Supp. 218
Hazard, R. E. Eng. Ind., September 1967
Testing Seals for Aerospace
Fluid Power Int., Vol. 32, No. 373, April 1967, pp. 38-40

New sealing technique developed in view of increasingly higher temperatures and pressures required to work aerospace hydraulic equipment; design of testing machine at British Aircraft Company at Stevenage, England, for evaluating dynamic performance of seals and to control temperature and pressure of hydraulic fluid, fluid leakage part seals and piston movement; types of seals investigated were metal, plastic and combined metal and plastic.

Supp. 219
Hooke, C. T., Lenis, D. J., O'Donaghue, J. P.
Elasto Hydrodynamic Lubrication of O-Ring Seals
IME Proc., Vol. 181, Pt. 1, No. 9, 1966-1967, pp. 205-210

The paper presents a theoretical solution to the ideal problem of an O-ring seal constructed of an isotropic elastic material moving at a constant axial velocity along a cylinder. The net leakage past the seal, which is the difference between the fluid flow across the seal when it is "pumping" against the pressure and that when the seal is "monitoring" or moving with the pressure, is shown to increase with sealed pressure.

Supp. 220
Hummel, B. L. - Editor IAA67-22610 09-15 1509
Seals: Reference Issue (3rd Edition)
Machine Design, Vol. 39, March 9, 1967, 219 pp.

The Design Data section contains completely revised and updated information which clearly and concisely describes how to select and apply seals and sealants. The all-new Product Directory is set up so that the reader in search of information about manufacturers of seals and sealants can easily find exactly which company can fill his needs.

Supp. 221
Johnson, D. F. Eng. Ind., January 1967
Controlled Gap and Segmental Seals
Nat'l Conference on Fluid Power, Proc. Vol. 19, 1965, pp. 39-43

Performance materials and construction of controlled gap and segmental carbon shaft riding type seals are discussed; seals can operate to 1400F for short periods of time and to 1200F for long periods of time; they are easy to install over shaft and have capability to handle axial tolerance or shaft movements.

Supp. 222
Kinsbury, J. E. IAA67-15788 04-18 0642
New and Proposed Cryogenic Temperature Resistant Plastics
Society of Automotive Engineers, Aeronautics and Space Engineering and Manufacturing Meeting, Los Angeles, Calif., Oct. 3-7, 1966, Paper 660638, 5 pp.

Discussion of the need for cryogenic-temperature-resistant plastic material in the space program; the selection of a plastic material for cryogenic application in dependent on the cryogenic fluid with which it will come in contact. Problem areas discussed are insulation of cryogenic materials, adhesives used on plastics, use of seals and gaskets and the expulsion of cryogenic propellant in a zero gravity environment. It is believed that although the research being done in this field at present appears to have limited applications in the aerospace industry, it may be of great significance for the future.

Supp. 223
Koehler, H. Eng. Ind., January 1967
Apparatus for Sealing in Plastic Forming Machines
U. S. Patent 3,238,566, March 8, 1966

Seal for sealing space between two members having sheet of plastic material interposed there between when one of the members is moved into engagement with sheet; continuous, seal receiving groove around outer edge of one of members and yieldable sealing member mounted in groove, sealing member being formed upwardly angled flat surface with projects above upper edge of said groove, and centrally disposed, continuous passage provided in one side wall of seal member.

Supp. 224
Lawrie, J. M., Summersgill, J. Eng. Ind., March 1967
Design Specifications for Hydraulic Seals
Fluid Power Int., Vol. 31, No. 368, November 1966, pp. 330-334

Information required for seal design specifications; categories considered include applications, fluid to be sealed, temperature, pressure, relevant dimensions, additional technical information, special, life, function limitations, permissible leakage and form of lubrication.

Supp. 225
Lindsey, M. H.
Mechanical Seals; Carbons Key Role
Chemical Engineering, Vol. 74, No. 5, February 27, 1967, p. 160

Carbon's unique qualities have made carbon sealing ring an indispensable part of the mechanical seal. Now, a denser, more impervious carbon is widening mechanical seal applications, and a new type of components is reducing costs. Carbon - because it is self-lubricating, resists chemical attack excellently, withstands large and rapid temperature change, and can be machined readily by conventional methods - finds wide application in the petroleum and petrochemical industries which handle many gases and liquids that cannot provide the lubrication that is necessary between moving surfaces.

Supp. 226
McManus, S. P., Piken, S. Eng. Ind., July 1967
Elastomeric Seals for Army's Lance Missile
Rubber Chem. & Technology, Vol. 39, No. 4, Pt. 2, September 1966, pp. 1233-1246

Missile propulsion system requires dynamic and static seals which must operate under extremely severe conditions for short periods; resin-cured chlorobutyl rubber meets requirements for low and high temperature operation and retains properties on exposure to propellants; comparisons with vinylidene fluoride-hexafluoropropylene copolymer (Viton). 19 Refs.

Supp. 227
Moore, B. C., Bergquist, L. E., Clements, P. R. Eng. Ind., July 1967
Cryogenic Low-Pressure Seal
J. Vacuum Science & Technology, Vol. 4, No. 1, January-February 1967, pp. 45-46

Communication to Editor describes development of simple, reliable metallic seal aluminum or stainless steel flanges at low pressures and cryogenic temperatures; self metal wire gasket is retained in rectangular groove in aluminum spaces.

Supp. 228
Mowers, R. E. IAA67-31990 17 2864
Static and Dynamic Seals Used on Rocketdyne Engines
Soc. Auto. Eng. Aerospace Systems Conf., Calif., June 1967

The paper reviews typical sealing concepts and materials used throughout the propellent feed systems and pneumatic and hydraulic control systems of liquid propellent rocket engines. Emphasis is placed on the sealing concepts utilized to compensate for extreme conditions.

Supp. 229
Murray, F. S. NSA21-67 14345
Bearing and Seal Materials for Liquid Metal Fabrications
Mechanical Technology Inc., Latham, N.Y., MTI-66TR66, pp. 6.3.1-26

Flame and plasma-sprayed hard coatings for bearing and seal components of process-fluid lubricated bearing systems are evaluated. The effectiveness of some of these coatings in protecting the surfaces against sliding damage in inert environments has been demonstrated in a number of practical applications. All of the test results show that these coatings can be used effectively for liquid metal-lubricated systems. The coatings simplify the design and fabrication of the bearing and seal components, but have the disadvantage of lack of adequate quality control. Necessary quality control procedures for the use of coatings must be developed to ensure a reliable product.

Supp. 230
Nau, B. S. IAA67-27889 14-15 2324
Hydrodynamic Lubrication in Face Seals
B.H.R.A. Int. Conf. on Fluid Sealing, 3rd, Cambridge, England, April 1967, Paper H3, 48 pp., 54 Refs.

The literature of fundamental studies of face seals is reviewed in detail with the emphasis on the conditions in the interface region. Attention is concentrated on hydrodynamic face seals. Among the subjects covered are the effects of surface roughness; wear-rates; leakage and inward pumping; and film thickness measurements.

Supp. 231
Paxton, R. R. Eng. Ind., December 1967
Carbon and Graphite Materials for Seals, Bearings, and Brushes
Electrochem Technology, Vol. 5, No. 5-6, May-June 1967, pp. 174-82

Developments of past ten years in carbon and graphite materials for seals, bearings, and electric motor brushes are briefly reviewed; properties of wide range of currently available materials are compared, and variety of applications are described indicating type of carbon or graphite which has been found suitable for each; bibliography of recent literature is included. 242 Refs.

Supp. 232
Ruhl, F. F., Wendt, A. B., Dalenberg, P. N.
A New Approach to Face Type Sealing
Lubrication Eng. 23(6), pp. 241-244, 1967, England

The following 17 major factors affect face type seal wear: design, materials, speeds, temperatures, pressure, alignment, lubrication or lack of it, cooling, mechanical forces, chemical potential, abrasive foreign material, care in assembly, the medium being sealed, secondary mechanical factors, secondary chemical factors, possible electrical effects, and the interactions of all factors. These factors are appraised in relation to each other and in relation to the overall problem.

Supp. 233
Schnetzer, E. NSA21-67 14353
Alkali Metal Bearing and Seal Development at Space Power and Propulsion Section
General Electric Co., Cincinnati, Ohio, Space Power and Propulsion Section, MTI-66TR66, pp. 6.2.1-46, Includes Appendices A through E

Three major programs associated with the development of Rankine cycle power conversion technology for space applications were undertaken; low viscosity bearing stability investigation, development of dynamic seals, and preparations for a liquid metal bearing test, simulating the conditions of a space generator. All programs were supported by a corresponding analytical effort, presently geared towards predicting bearing-rotor response. Experimental results are expected in 1966 and 1967. Screw seals, rotating channel and slinger squeeze seals were investigated in the dynamic seal program. Interface instability was identified as one of the major problems. It was resolved for the case of the slinger squeeze seal, which was successfully tested in potassium.

Sur 234
Shevckenka, R. P. IAA67-22537 09-28 1560
Shaft, Bearing and Seal Systems for a Small Engine
Society of Automotive Engineers, Automotive Engineering Congress, Detroit, Mich., January 9-13, 1967, Paper 670064, 10 pp.

Discussion of the key mechanical difficulties inherent in designing shaft bearing and seal systems for three high-speed turbine engines. A review is made of the bearing, shaft and seal system of a family of large turbine engines which has demonstrated its high reliability, high efficiency, and low cost per unit of power as the thermodynamic, material and structural component technology has been improved and as the design power requirement has been reduced. Three different solutions to the problems discussed are presented; two for sophisticated, efficient engines for weapon system consideration, and the third for a potential industrial, low cost, high-efficiency, moderately heavy engine.

Supp. 235
Smaley, E. M. Eng. Ind., March 1967
Sealing with Gaskets
Machine Design, Vol. 38, No. 25, October 27, 1966, pp. 171-187

Design guide presented shown best practices for creating seal in gasketed joints and insuring seals longevity; two sets of parameters are surveyed, those that create seal and those that affect life in service, e.g., flange movement; gaskets considered are cork composition, cork and rubber asbestos fiber and rubber cellulose fiber and rubber, or rubber above; flat and grooved - face types of flanges are discussed.

Supp. 236
Stair, W. K., Hale, R. H. IAA67-22706 10-15 1659
The Turbulent Visco Seal - Theory and Experiment
B.H.R.A. Int. Conf. on Fluid Sealing, 3rd, Cambridge, England, April 1967, Paper #2, 32 pp., 12 Refs.

Experimental evaluation of the laminar viscosial for a number of seal geometrics has shown that the Boon and Tal's analysis is in better agreement with experiment than any other analytical approach available. It has been observed further that the seal performance improved and became dependent upon Reynold's number as turbulent conditions were encountered.

Supp. 237
Stair, W. R. IAA67-15356 04-15 0630
Effect of Groove Geometry on Viscoseal Performance
ASME Winter Annual Meeting and Energy Systems Exposition, New York, N.Y., November 27-December 1, 1966, Paper 66-WA/FE 28, 8 pp., 19 Refs.

An experimental investigation of the effect of groove geometry on the performance of the grooved-shaft viscoseal is being conducted. Using distilled water as the sealed fluid, the concentric and eccentric performance of ten seals has been determined under laminar and turbulent conditions. An approximate method is devised to predict the empirical factors in the turbulent sealing equation. The findings reveal that the optimum seal geometry for laminar operation is not optimum for turbulent operations. The seal eccentricity has slight effect on performance, and the phenomenon of air injection may present a major problem in the application of the viscoseal.

Supp. 238
Ward, J. R. Eng. Ind., June 1967
Piston Seal Development for Oil-Free Compressors
Naval Engrs. J., Vol. 78, No. 6, December 1966, pp. 995-1002

Report of experimental work at U. S. Navy Marine Engineering Laboratory leading to design of nonlubricated seal for air compressors intended for shipboard use, which successfully operated 1500 hrs. at discharge pressure of 4500 psi; seal consisted of sleeve made of 25% glass-filled TFE and follower made of polyimide material; long term operation occured in chromium-plated cylinder bore; in later evaluations, follower material was charged to metal impregnated carbon-graphite.

Supp. 239
Wasil, T. J., McCleary, G. P.
Sealing Corrosive Materials
Lubrication Engineering, Vol. 23, No. 6, June 1967, pp. 234-240

This paper discussed tungsten carbide, tungsten titanium carbide, and titanium carbide as applied to axial mechanical sealing materials for sealing corrosive fluids. It states that the corrosive resistance of the carbides is dependent mainly upon the compatibility of the corrosive media with the carbide binder material. The tungsten carbide grains are inert to most corrosive fluids. The general mechanism of the carbide corrosion is explained. Galvanic corrosion, concentration cell corrosion, erosion corrosion, direct chemical attack and oxidation are discussed in detail.

Supp. 240
Zuck, J., Strom, T. N., Ludwig, L. P., Johnson, R. L., IAA67-28797 14-15 2327
Convective Inertia and Gas Ingestion Effects on Flow Regimes of the Viscoseal - Theory and Experiment
ASLE, Annual Meeting, 22nd, Toronto, May 1967, Paper, 61 pp., 39 Refs.

An experimental and visual investigation of the viscoseal operating regions was conducted. Importance of convective inertia forces and changes in the functional relations between variables in the super laminar flow regime are discussed. Correlation between modified Reynolds number and sealing capacity improvement is found.

Supp. 241

Adams, M. L. and Colsher, R. J.
An Analysis of Self-Energized Hydrostatic Shaft Seals
Paper No. FICFS-38, May 1969

An analysis of a self-energized hydrostatic shaft seal operating with incompressible fluid in the laminar regime is presented. Dimensionless results are optimized for maximum stiffness. Comparisons are made between full annulus and sectored annulus recess configurations. It is shown that translational and torsional stiffnesses are critically dependent upon orifice size.

Supp. 242

Allen, C. M. and W. G. Rieder
Performance of a Modified Packed-Gland Seal
ASME Paper No. 67-WA/Lub-5, Winter Annual Meeting, Nov. 1967

An experimental investigation of the mechanics of hydrodynamic lubrication and heat generation and dissipation of packed-gland rotary shaft seals was performed on a somewhat novel design. The design differs from conventional stuffing-box seals in that the packing was inverted so that it rotated with the shaft.

Supp. 243

Allen, E. T.
Crankshaft Rear Seals
Third International Conference on Fluid Sealing, April 1967, Paper D2

The limitations of scroll seals and split rubber seals without spring loading are discussed in relation to motor vehicles. Consideration is given to the design requirements of crankshaft wear bearing seals and other sources of apparent rear end oil leakage are reviewed. An account of the development of satisfactory sealing devices is given, a spring load lip seal is found to meet the exacting requirements of a crankshaft rear bearing seal.

Supp. 244

Anderson, H. H.
Self-Seal for Very High Pressures
ASME Paper No. 67-WA/PVP-5, November 12-17, 1967

New closure embodies end plate within cylinder held captive and leakproof by endless metal ring partly within groove in cylinder bore and partly engaging plate; ring, which at first sight cannot enter cylinder bore, is inserted by ovalling within its elastic limit; single ring has held 16,000 psi, and, aided by segmental shear ring, can hold up to 100,000 psi.

Supp. 245

Anno, J. N.; Walowit, J. A.; and Allen, C. M.
Microasperity Lubrication
Third International Conference on Fluid Sealing, April 1967, Paper No. E2

Large load-supporting capability has been observed between parallel surfaces of a rotary-shaft face-seal when one of the surfaces contains microasperities in the form of circular cylinders with 12×10^{-3} inch diameter and 100-microinch

height, covering about one-third of the surface. A theory is presented in which the load support is attributed to small tilts on the tops of the asperities.

Supp. 246

Anno, J. N.; Walowit, J. A.; and Allen, C. M.
Load Support and Leakage from Microasperity-Lubricated Face Seals
Paper No. FICFS-21, May 1967

Observations of load support from various shapes of microasperities placed on the surface of a rotary-shaft face seal indicate that all are comparable in this respect and generally follow the theory developed previously for cylindrical asperities. Both positive and negative asperities produce similar load support.

Supp. 247

Anno, J.; Walowit, J.; and Allen, C.
Microasperity Lubrication
ASME, WA, 67-WA/Lub-1

(See preceding abstract)

Supp. 248

Anon.
New ASTM Work on Gaskets, Seals and Sealants
Materials Research and Standards, Vol. 8, February 1968, P. 36-7

Gasket and seal requirements are steadily becoming more severe as higher loads, exotic fluids, more attention to emissions and longer warranties press on performance. As the difficulty of maintaining the seal under new conditions increases, the education of the designer and fabricator becomes more important. Five ASTM committees serve as national forums for gaskets, seals, and sealants, their measurement and performance.

Supp. 249

Anon.
Oil/Seal Compatibility Test Method
Rubber J v 150 n 9 September 1968

Method of ensuring compatibility of mineral hydraulic oils in hydraulic equipment with commercially produced nitrile rubber seals; percentage volumetric change of this rubber in oil constitutes AHEM Nitrile Seal Compatibility Index of oil.

Supp. 250

Ashworth, T.; Bunting, J.; and Smith, S.
Cryogenic Lead Seals
Cryogenics Vol. 3, June 1968, P. 167

Electrical leads-through are a necessary evil in every cryostat in which a vacuum space is used to provide thermal insulation. It is concluded after testing, that the bonded lead seal is applicable to systems in which small d. c. signals are to be measured and in which the leads themselves have to support a temperature gradient.

Supp. 251

Aston, M. W.; Fletcher, W.; and Morrell, S. H.
Sealing Force of Rubber Seals and Its Measurement
Paper No. FICFS-7, May 1969

The behaviour of rubber seals depends on the sealing force exerted by the compressed rubber under service conditions. When the temperature is increasing, the sealing force can be predicted approximately from the Gough-Joule effect and the expansion of the rubber relative to the metal housing. Descriptions are given of three pieces of apparatus for measuring sealing force.

Supp. 252

Austin, R. M. and Fisher, M. J.
A Hydrostatic Seal for Boiler Feed Pumps
Second International Conference on Fluid Sealing, April 1964, Paper No. E4

This paper is concerned with the design of a hydrostatic seal for boiler feed pump applications. A small test seal was operated satisfactorily under some conditions and pressure losses in the seal recesses were found to have a marked effect.

Supp. 253

Bailey, B. J. and Kellner, K.
A Static High Pressure Low Temperature Seal
J. Sci. Instrum., Vol. 44, November 1967, P. 962

A method of producing a demountable seal between a thin walled tube and a pressure vessel is described which will contain pressures between .1 torr and 500 gf cm^{-2} over the temperature range 90-300°.

Supp. 254

Balakrishnan, K., and Kar, S.
A Theoretical and Experimental Investigation of Mechanical Seal
Paper No. FICFS-17, May 1969

The leakage through a mechanical seal under full-film conditions in the seal gap is predicted by a theoretical equation developed from the basic equations of fluid mechanics. Experiments were conducted to compare the leakage rate predicted using different seal material combinations. The designed seal performance were found to be satisfactory with very small leakage rates for all combinations.

Supp. 255

Balfe, M. P.; Gill, G. E.; Harrison, J. M.; and Roper, K. M.
Leather as a Sealing Medium
Second International Conference on Fluid Sealing, April 1964, Paper No. B5

The physical structure and chemical composition of leather are summarized. A description of the manufacture of sealing elements from leather is given. The advantages which leather, as a sealing medium, derives from its fibrous structure are described.

Supp. 256

Baron, H. W. and Enting, J.
The Development of a Flexible Seal for Reciprocating Motions
Third International Conference on Fluid Sealing, April 1967, Paper No. C3

During the development of a piston engine for generating cryogenic cold, wherein a gas space which should remain absolutely oilfree and a crankcase which must be sealed gastight from the gas space, it seems that the current sealings for reciprocating rods did not satisfy with the high rod velocities occurring with this engine. The principal imperfections of these sealings were: limited life, gas leaking and some oil transport from the crankcase to the gas space.

Supp. 257

Batoh, B. A. and Iny, E. H.
Pressure Generation in Radial-Face Seals
Second International Conference on Fluid Sealing, April 1964, Paper No. F4

Measurements of the thickness and pressure of the fluid film between the faces of a radial-face mechanical seal have revealed a twice shaft speed fluctuation in both thickness and pressure, the pressure increased as the gap decreased and vice-versa, indicating the hydrodynamic lubrication conditions existed within the film.

Supp. 258

Beerbower, A. and Dickey, J. R.
Advanced Methods for Predicting Elastomer/Fluids Interactions
A. S. M. E. and A. S. L. E. Lubrication Conference, October 8-10, 1968
ASLE Paper 68 LC-21

The method of predicting seal swell previously reported by this laboratory has been extended to include three-dimensional solubility parameter concepts. Techniques for analysis and calculation of the partial solubility parameters in consistent dimensions have been formulated for pure liquids, binary mixtures of known composition, and multicomponent mixtures of unknown ingredients.

Supp. 259

Berens, A. S. and Van Deven, D. E.
Low Temperature Evaluation of Rear Axle Seals
ASME Paper No. 67-WA/Lub-22, Winter Annual Meeting, November 1967

Tests are described for observing the effects of low temperatures on the operation of automobile rear axle seals. Results showed increased wear on both seal and shaft is likely to occur at low temperatures and this wear is increased by lubricants which become extremely viscous at these temperatures.

Supp. 260

Bernd, L. H.
Survey of the Theory of Mechanical Face Seals
A.S.L.E. Lubrication Engineering, Vol. 24, October 1968

A brief review is presented of various theories and approaches that have been followed seeking to explain the functioning of face seals, and improve their operation. The known, essential characteristics of a radial face seal are summarized. The performance and inherent limitations of a seal are discussed in terms of the axial forces on the seal faces and the character of the lubrication between the mating faces.

Supp. 261

Van Beukering, H. C. J., and van der Aa, H. H. M.
A Rolling Diaphragm Seal for High Pressures
Third International Conference on Fluid Sealing, April 1967, Paper No. G4

The oil-supported rolling diaphragm provides a hermetic seal for rods and pistons and is suitable for very high fluctuating pressures and very high speeds. Research into various types of rubber for the diaphragm has led to the use of a special polyurethane elastomer with a previously unheard-of life provided that the correct conditions, such as dimensions, pressure difference and type of oil, are carefully selected.

Supp. 262

Bertouille, R.
Boot Seal Technology Boom
Rubber World v 157, November 1967

Evolution of protective seals in form of rubber products for automotive suspension and steering systems and variously called dust covers, sliding covers, shields, or boot seals; material requirements to manufacture boot seals range from 5 to 5.5 million lb of rubber-nearly all neoprene; seal design, materials and typical configurations described and illustrated.

Supp. 263

Billet, A. B.
Hydraulic Sealing in Space Environments
Second International Conference on Fluid Sealing, April 1964, Paper No. C2

A number of new sealing configurations have been developed or are presently under development. These include modifications to carbon face seals to include a wedge type secondary seal. This development also includes extending the temperature ranges of previously used seals.

Supp. 264

Billington, I. J.
Experimental Assessment of the Validity of Pressure Measurements in the Fluid Film Between Seal Faces
Second International Conference on Fluid Sealing, April 1964, Paper No. F3

The feasibility of measuring static pressures in the thin fluid films between the faces of rotary contact seals has been assessed experimentally with the use of large scale models. The investigation has been conducted using static pressure taps whose diameters ranged from 0.5 to 64 times the thickness of the fluid film.

Supp. 265

Billington, I. J. and Fitzsimmons
The Spring Supported Hydrostatic Seal
68-WA/Lub-9

The performance of controlled leakage shaft seals is critically dependent, particularly at high pressure, upon the ability to achieve and maintain the correct clearances between stationary and rotating members. A concept in which both members of a seal can float, isolated from the backup structure, is described; theoretical performance is discussed; experimental verification of the performance of such a seal is presented and provides good correlation with the theory

Supp. 266

Bjerklie, J. W.; Cheng, H. S.; Wilcock, D. F.; Ludwig, L.; and Townsend, D.
Configurations for Gas Turbine Compressor End Seals
A.S.L.E., May 1968, Preprint 68 AM 4B-4

Modern high performance multistage axial flow compressors have several air leakage paths which reduce compressor performance. The most significant leakage occurs past the labyrinth seal at the high pressure end of the compressor. Since it was estimated that this leakage reduced the compressor efficiency by 1%, a project was undertaken to develop low-leakage-rate non-contact seals which would reduce compressor end-seal leakage by an order of magnitude. A large number of seal concepts and ideas were examined for suitability. Each concept was evaluated on the basis of leakage rate, tracking capability, gas film stability, thermal and elastic distortion, wear, and reliability.

Supp. 267

Blow, C. M.
The Development and Testing of Elastomeric Materials for Fluid Sealing Applications
Second International Conference on Fluid Sealing, April 1964, Paper H1

The significance of laboratory test methods now commonly applied to elastomeric materials is discussed in relation to service performance. Hardness, elastic recovery properties, heat resistance and liquid absorption are among the most important and experimental results are given to illustrate the effect of the various factors involved.

Supp. 268

Boucher, R. F., and McVeigh, J. C.
Some Experiments on the Reiner Centripetal Effect in Air
Second International Conference on Fluid Sealing, April 1964, Paper F1

Reiner discovered that a positive excess pressure existed between two flat coaxial discs when they were very close together and one of them rotated. Experiments with both steel and perspex discs were carried out and demonstrated the difficulties in attempting to verify the cause of this phenomenon.

Supp. 269

Bremner, G. F.
Experimental Study of Balance Conditions, Leakage Rates, and Pressure Profiles for Stationary Radial-Face Seals
Second International Conference on Fluid Sealing, April 1964, Paper F2

A limited experimental study has been made of the performance of stationary radial-face seals operating in the purely viscous flow regime. The re-

sults of the tests substantiated the use of the purely viscous flow relations in predicting the performance.

Supp. 270

Brewer, D.
Shaft Seals in Medium Pressure Low Cost Fluid Power Pumps
Third International Conference on Fluid Sealing, April 1967, Paper G2

Medium pressure fluid power circuits are widespread in industry today and a shaft seal leak can vary in effect from inconspicuous to disastrous. Within a price bracket appropriate to the market, various solutions are described. There are indicated practical operating limits which have been based on field operating data from production units.

Supp. 271

Brindle, B.
A Seal User's Experience of Fluid Sealing
Third International Conference on Fluid Sealing, April 1967, Paper D5

The subject of fluid sealing is wide and seals can be temperamental. In order to cover this in some depth within the time available, this paper has been restricted to two diverse automobile engine applications: a) Crankshaft oil sealing and b) Water pump coolant sealing.

Supp. 272

Brink, R.; Brady, J.; Daly, R.; and Rasonussen, W.
Seal Technology Assures Reliability
ASME Paper No. 67-WA/Lub-13

This paper portrays the development steps necessary to bring a line of oil seals from an idea to the market place. Emphasized are the design considerations, the test methods employed to verify the design parameters, and discussion of the performance characteristics as they relate to applications.

Supp. 273

Brown, J. M.
Oil Seal Quality Control - Important or Imperative
ASME Paper No. 67-WA/Lub-12

In 1957, a research program was initiated for the study of radial lip seals. Results of this study indicated that there was a lack of seal technology concerning the seal lip characteristics. From a program of planned experimentation, there emerged a series of controls which resulted in vastly improved seals being available for assembly and an inherent reduction in the sealing problems.

Supp. 274

Burton, R. A.
An Experimental Study of Turbulent Flow in a Spiral-Groove Configuration
A.S.M.E., November 12-17, 1967, Paper 67-WA/Lub-3

Pressure and velocity data are given for measurements in a large-scale experimental apparatus, designed to act as a model of a spiral-groove con-

figuration operating at turbulent Reynolds numbers, and using air as a working fluid. The results predict that inertial effects at the steps in film thickness will exert a dominant influence on flows in machine elements made in geometric similitude to the model.

Supp. 275

Carr, J. J.
Process Control and Testing of Elastomer and Elastomeric Compositions
ASME Paper No. 67-WA/Lub-10, Winter Annual Meeting, November 1967

The paper presents a discussion of test equipment and procedures used to control raw polymers and compounded rubbers. The data obtained during sampling stages and its value and subsequent application to monitoring production lots of elastomeric materials are presented.

Supp. 276

Chabosean, Jacques
Gas Bearings and Rotating Seals
Ninth Hydraulic Conference, Paris, June 1966 (in French)

Consideration of the problems of gas bearings and of tight seals for rotating machines which bundle gases of various types. Several types of seals and gaskets are described and illustrative diagrams are presented. Attention is given to the development of gaskets.

Supp. 277

Cheng, H. S.; Chow, C. Y.; and Castelli, V.
Performance Characteristics of Spiral-Groove and Shrouded Rayleigh Step Profiles for High-Speed Non-Contacting Gas Seals
A.S.M.E., Lubrication Symposium, June 1968, Paper 68-LubS-38

Current methods in gas lubrication were used to analyze the gas-film characteristics in a high-speed, face-type gas seal. Detailed analyses are presented for two different surface geometries, the spiral-groove-orifice seal and the shrouded Rayleigh step seal. The performance of a spiral-groove seal without the orifices is treated, considering the difference in performance when the grooves are located at the high pressure, low pressure, or both sides.

Supp. 278

Cheng, H. S.; Chow, C. Y.; and Wilcock, D. F.
Behavior of Hydrostatic and Hydrodynamic Noncontacting Face Seals
A.S.M.E., November 12-17, 1967, Paper 67-WA/Lub-19

The pressure generation and static stability of face-type seats are discussed, and an expression is developed to estimate the effectiveness of hydrodynamic action in these seals. Some design data are presented for the hydrostatic step seal, hydrostatic-orifice compensated seal, hybrid spiral-groove seal, and the shrouded Rayleigh step seal. The seal ring distortions due to initial imperfection, pressure, and thermal expansion are discussed.

Supp. 279

Cheng, H. S. and Snapp, R. B.
A Study of the Radial Film and Pressure Distribution of High Pressure Face Seals
Third International Conference on Fluid Sealing, April 1967, Paper E3

This paper presents a detailed study of the interaction between the elastic deformation of the seal surface and the hydostatic pressure in a typical, high pressure face seal for marine applications. In the analysis presented, the elasticity equations are used to calculate the local deformation and the ring and the face ring. Results are shown for several typical face seals for marine applications.

Supp. 280

Chiang, T. and Cheng, H. S.
An Analysis of Flexible Seal Ring Vibrations
ASLE Transactions, Vol. 11, July 1968

Vibration of the seal ring induced by the axial pulsation, wobbling, and initial distortion of the rotor surfaces in a face-seal assembly consisting of a seal ring and a carrier is analyzed. The analyses are divided into two parts - a rigid body analysis dealing with vibrations due to the axial pulsation and wobbling and an elastic ring analysis dealing with vibration induced by the initial surface waviness of the rotor. Results are obtained in terms of natural frequencies and influence coefficients characterizing the response of the seal ring to the rotor motion.

Supp. 281

Cieslik, W. J.
A Helium Face Seal Application In a Liquid Oxygen Pump
Paper No. FICFS-24, May 1969

The object of the work discussed in this paper was to develop a reliable helium gas shaft seal for use in an electric motor-driven, liquid oxygen pump on a space vehicle. The development effort covered tests on two basically different face seal designs, one with an attached carbon face and the other with a floating lap-fitted carbon face.

Supp. 282

Clarke, N. W. B.
Rubber Sealing Rings in Flexible Joints for Underground Pipelines
Use of Rubber in Engineering-Proc Conference, London, 1966

Rigid pipe lines require flexible joints in order to withstand forces imposed on them; such joints must permit angular and axial movement to take place; they must also be leakproof; rubber sealing ring is effective device for achieving these requirements; design and installation problems are described.

Supp. 283

Cnops, R. F.
The Friction of Elastomer Seals
Second International Conference on Fluid Sealing, April 1964, Paper B7

The experimental investigation of the friction of a cup seal shows the peculiar behaviour of elastomer seals under dynamic conditions. The

interpretation of the results is based on the investigation of the seal under static and dynamic conditions.

Supp. 284

Cook, D. V. and Guerreri, B. C.
Deflection in Gasketed Joints
Machine Design, Vol. 40, March 14, 1968, p. 149-150

Development of special equations which relate gasket compression mid way between fasteners, where compression is a minimum, to the maximum compression at the fasteners. The only values required to apply these equations are the moment of inertia, the modulus of elasticity of the gasketed members, and the gasket compression curve.

Supp. 285

Coopey, W.
Fresh Look at Springloaded Packing
Chem. Eng. r. 74, November 6, 1967

Spring loaded packinghas received increasing attention from sealing manufactures and chemical processors, because of its inherent advantages over conventional lantern-sized stuffing boxes in eliminating leakage and need for manual adjustment.

Supp. 286

Csernatony, L. and Crawley, D.
The Practical Application of Viton "A" Seals in a High Vacuum
Vacuum, Vol. 17, October 1967, p. 551

These findings indicate how Viton elastomers may be used to best advantage in high vacuum systems and how the pump-down performance may be predicted.

Supp. 287

Decker, O.
Dynamic Seal Technology-Trends and Developments
Mech. Eng. v 90 n 3, 4, 5, March 1968 April May Pt 1

Survey of dynamic seals covers mechanical (face or sleeve), floating (face or sleeve), labyrinth (face or sleeve), visco or screw, centrifugal or slinger, and combination of any of these. Pt 2. Developments in dynamic seals for advanced air-breathing propulsion systems; design, advantages, and limitations of orifice-compensated hydrostatic floating-face seals and self-aligning hydrodynamic seals. Pt 3. For Rankine-type turboalternator, centrifugal seal has been developed.

Supp. 288

Dega, Robert L.
Zero Leakage: Results of An Advanced Lip Seal Technology
Third International Conference on Fluid Sealing, April 1967, Paper D3

The advent of the "Helixseal", a type of hydrodynamic sealing element, to the automotive industry has provided a means for greatly reducing existing warranties with very little increase in production cost. The "Helixseal" is based upon hydrodynamic principles which allow the generation of controlled fluid forces. It consists of angular pumping vanes, applied

to the seal face to provide a viscous shear force that opposes leakage of the fluid film. Optimized design of the "Helixseal" can assure zero leakage in automotive seal applications.

Supp. 289

Donald, M. B. and Morris, C.
Effect of Flange Design on Gasket Performance In Narrow-Face Bolted Joints
Second International Conference on Fluid Sealing, April 1964, Paper A4

Photoelastic and other experiments were undertaken on a flange assembly. The stress and strain were fixed in the assembled joint and measured after it was dismantled. The following other parameters were examined: the outer diameter of the gasket, total bolt load, number of bolts used, and internal fluid pressure.

Supp. 290

Dondonor, N. T., et al.
Seals
Isobret., Prom. Obraztsy, 1968, 45(16)

Seals are manufactured from asbestos fiber, a suspension of Fluoroplast 4D and antifriction additives. Annular seals are formed by subjecting the woven packing to a pressure of 180-220 Kg/cm^2 for 20-60 sec at 25^0.

Supp. 291

Dowson, D. and Swales, P. D.
An Elastohydrodynamic Approach to the Problem of the Reciprocating Seal
Third International Conference on Fluid Sealing, April 1967, Paper F3

The results of a theoretical investigation of a lubricated contact between a rigid surface and a soft material are presented, and compared with those of other workers. A surprising conclusion is the extremely small initial preload or 'nip' which is theoretically required to initiate the sealing function. Preliminary experimental work suggests that it may be necessary to consider local elastohydrodynamic effects at inlet and outlet in addition to the overall elastic effect to explain the characteristics of practical seals.

Supp. 292

Dowson, D. and Swales, P. D.
The Development of Elastohydrodynamic Conditions in a Reciprocating Seal
Paper No. FICFS-1, May 1969

Attempts to determine experimentally the working conditions at the sealing face of a reciprocating seal have always been hampered by the difficulty of controlling adequately the experimental conditions. In this paper a rotating disc machine is described which simulates the action of a reciprocating seal undergoing an infinitely long stroke, under steady conditions. Some results have been obtained for seals of practical shapes which have surprising features, but in general conform qualitatively to practical observations of seal behaviour.

Supp. 293

Drutowski, R. C.
Contact Elasticity of Seal Elastomers
ASME Paper No. 67-WA/Lub-9, Winter Annual Meeting, November 1967

The calculation of Young's modulus of an elastomer is based on the optical measurement of the contact between a transparent spherical indenter and the elastomer. A critical comparison of this method is made with conventional hardness tests. Experimental results are presented.

Supp. 294

Drutowski, R. C.
Hertzian Contact and Adhesion of Elastomers
Paper No. FICFS-5, May 1969

The contact of a hard sphere with a flat elastomer is examined both analytically and experimentally when adhesive stresses are present. Use of a transparent spherical indenter enables continuous measurement of contact size while the samples are pulled apart. During separation, while the contact size decreases and the tensile annulus becomes a larger percentage of the total contact, the average tensile stress remains constant.

Supp. 295

Dukes, W. A.; Bryant, R. W.; and Long, J. V. (Mrs.)
Commercial Sealants for Parallel Threaded Joints
Second International Conference on Fluid Sealing, April 1964, Paper A3

Part 1 - Temporary seals: Desirable properties are formulated and discussed, and the relative efficiency of the materials in sealing parallel-threaded joints is assessed.
Part 2 - Permanent seals: The materials were tested in a series of sealing trials of increasing stringency. The proposed performance specification is critically examined in the light of the results obtained.

Supp. 296

Ehrich, F.
Aeroelastic Instability in Labyrinth Seals
ASME, March 1968, Paper 68-GT-32

Study of the stability problem associated with labyrinth seals in high-power, high-speed rotating machinery. A model is defined in which a pressure perturbation in the internal volume of the seal between the high-pressure and low-pressure teeth can cause an elastic rotation of the seal rotor or stator element about a virtual pivot point located on the high-pressure or low-pressure side of the seal. A stability parameter is then derived in terms of the high-pressure and low-pressure tooth clearances, the supply pressure to the seal, the seal width and height, and two parameters indicative of the elastic properties of the seal-its stiffness to radial motion and the location of its virtual pivot point.

Supp. 297

Elsworth, L.; Holland, L.; and Laurenson, L.
Bakable Vacuum Seals Using Aluminium Wire Gaskets
Third International Conference on Fluid Sealing, April 1967, Paper C3

Materials and techniques used to provide vacuum seals for bakable ultra high vacuum plant are discussed. After a review of work in this field the bakable seals developed by the writers are considered. These were made from aluminium wire gaskets clamped between plain stainless steel flanges. Such seals remain leak tight during and after repeated baking to temperatures up to 450°C providing the gasket either becomes bonded to the glanges or is under constant compression, during and after yield.

Supp. 298

Eudes, et. al.
Elastomer Seals for High-Pressure Carbon Dioxide
Nucl. Eng. 1968, 13(146)

Sealing problems encountered on the EL4 reactor had led to the search for an elastomer compatible with CO_2 under high pressure. A silicone-ethylene type was formulated which met the working conditions generally demanded.

Supp. 299

Evans, G. B.
Elastomers & Airframes
Aircraft Engineering, November 1967

Note on the possible uses of rubber moldings as static, dynamic, and general seals and as mountings and hoses in the hydraulic, fuel, and air systems of aircraft. The physical and technical properties of a group of nine elastomer types are given.

Supp. 300

Ewbank, W. J.
Dynamic Seals - A Review of the Recent Literature
ASME, November 12-17, 1967, Paper 67-WA/Lub-24

Review and discussion of all the significant papers printed since 1952 on rotary shaft seals, both of the flat face and of the circumferential type. Discrepancies between reported results are pointed out where noted, and suggestions are made for needed future experimental work.

Supp. 301

Findlay, J. A.
Cavitation in Mechanical Face Seals
ASME, November 12-17, 1967, Paper 67-WA/Lub-20

The characteristics of mechanical face seals operating with hydrodynamic films containing gas cavities were studied. Solutions to Reynolds' equation for hydrodynamic lubrication were obtained using both the short bearing approximation and numberical iterative methods. These two methods of solution satisfy continuity of flow throughout the seal film. Leakage rates and seal hydrodynamic loads were calculated.

Supp. 302

Findlay, J. A.
Inward Pumping in Mechanical Face Seals
ASME, Paper 68-Lub-2, October 8-10, 1968

Hydrodynamic characteristics which produce inward pumping in mechanical face seals are studied; solutions to Reynolds' equation for hydrodynamic lubrication are obtained using both short-bearing approximation and numerical iterative methods; it is found that pumping in face seals results from combined radial eccentricity and misalignment, or one circumferential surface wave; experimental results confirm theory. 18 refs.

Supp. 303

Findlay, J. A.
Measurements of Leakage in Mechanical Face Seals
Paper No. FICFS-19, May 1969

The characteristics of mechanical face seals operating with hydrodynamic films were studied experimentally and analytically. Experimental and theoretical results are given showing the effects of gas cavities in the film and inward pumping. Experimental results are also given for the two fluid case, i.e., where the seal separates two different fluids; and for the case where inertia effects are important.

Supp. 304

Fisher, C. F., Jr. and Stair, W. K.
On Gas Ingestion and Fluid Inertia Effects in Viscoseals
Paper No. FICFS-31, May 1969

A study of the stability of a dynamic gas-liquid interface between rotating cylinders was initiated to provide a better understanding of the mechanism of gas ingestion in viscoseals. Results of the study suggest that an acceleration of a portion of the interface toward the more dense fluid and/or a velocity of a portion of the interface toward the more viscous fluid are factors which cause a break to occur in an interface.

Supp. 305

Gardner, J. F.
Combined Hydrostatic and Hydrodynamic Principles Applied to Non-Contacting Face Seals
Paper No. FICFS-36, May, 1969

This paper deals with a combined method of achieving non-contacting seal operation. A curved face gap seal for use with gas, operating solely on hydrostatic principles. The gap is maintained when pressure is applied to the seal. For this reason a minimum pressure must exist when the seal is operated.

Supp. 306

Goldway, E. and Bounds, R. S.
Hydrostatic Seals for Water Turbine Shafts
Second International Conference on Fluid Sealing, April 1968, Paper E3

A seal suitable for use on large diameter water turbine has been developed.

The operating conditions made it necessary to devise a seal configuration which automatically maintained full film lubrication. This was achieved by balancing of all the forces acting on the stationary ring in such a manner that the seal gap is a function of the degree of balance.

Supp. 307

Golubiev, A. I.
On the Existence of a Hydrodynamic Film in Mechanical Seals
Third International Conference on Fluid Sealing, April 1967, Paper E1

The motion of a layer of liquid in the mechanical seal clearance is considered by using the equations of hydrodynamics and heat transfer. The hydrodynamic force appearing in the layer is explained by local thermal deformations of one of the walls of the friction couple.

Supp. 308

Golubiev, A. I.
The Development of a Three-Stage Screw-Type Labyrinth Seal
Paper No. FICFS-35, May 1969

An hydraulic theory was applied to the design of a three-stage screw-type labyrinth seal. The theoretical data were compared with experimental ones. The experiments confirmed the possibility of using such seals for the shafts rotating at high speeds.

Supp. 309

Greiner, H.
Mechanical Seals in Marine Industry
Lubrication Eng., V. 23 n. 6 June 1967

Current trends in ship automation create a growing requirement for mechanical seals on all rotating machinery. The range of applications to engine room auxilliaries, cargo handling and main propulsion machinery introduces problems in corrosion and abrasion resistance, reliability, maintenance and service. Factors and current practices are discussed with respect to seal selection and application engineering.

Supp. 310

Greiner, H. F.
Design and Test of An Unloading Gas Barrier Face Seal
Paper No. FICFS-26, May 1969

The gas barrier seal design discussed here employs a gas barrier system which will produce a varying unit loading on the conventional contact faces of a face seal. Discussion of geometric stability and an analysis of test results against theoretical flow yields some further insight into design requirements and variables affecting face seal performance.

Supp. 311

Hait, P. W.
The Application of Polyimide to Ultrahigh Vacuum Seals
Vacuum, Vol. 17, October 1967, p. 547-50

Until the present time, the only satisfactory seals that have been demonstrated to withstand the bakeout temperatures of ultrahigh vacuum systems

are those with metal gasket materials. This article describes measurements made on a new ultrahigh vacuum material called polyimide. It withstands intermediate temperature bakeout up to 300°C.

Supp. 312

Hamilton, J. and Walton, F.
Sealing Ring Assemblies
U. K. Atomic Energy Authority, November 1968

The described ring assembly for sealing at ungrooved flanges has inner and outer annular rigid guide rings and a sealing membrane between the guide rings.

Supp. 313

Harvey-Flint, G.
Use of Acrylic Materials as Locking Sealants in Engineering Assemblies
Production Engr. v 46 n 10, October 1967

Certain acrylic resins belong to group of materials, called anaerobic, which set in absence of air; presence of metals accelerates this process, while nonmetals can be made reactive by pretreatment with metallic ion coating; thus, these acrylics can be useful in locking and sealing components; securing threaded fasteners, nonsliding splines, bearings, and bush mountings; pipe sealing.

Supp. 314

Hershey, L. E.
Compare Cooling Methods for Mechanical Seals
Heating, Piping & Air Conditioning v 40, March 1968

Testing of five different cooling systems on same pump showed that cost of cooling for water and heat loss can vary widely when applied to equipment of different makes; heat losses and resulting annual costs are compared for various systems tested.

Supp. 315

Hershey, L. E.
Time Rate of Wear Testing Mechanical Seal Carbons
Lubrication Eng. v 24 n 1, January 1968

Relatively short time test for mechanical seal carbons that indicates long term wear results to be expected is described; comparison of wear rate of carbon after series of one day tests and after series of four day tests of same items is made and results for variety of different carbons against three different counterface materials are tabulated.

Supp. 316

Hershey, L. E.
Evaporative Cooling Applied to Mechanical Seals
Paper No. FICFS-11, May 1969

The cooling of mechanical seals using the heat of vaporization from boiling water is demonstrated as a feasible system. The effect of the higher flush temperature on seal cage temperature, seal wear, pump bearing temperature, and cooling water consumption are discussed. It is shown that the seal environment is only slightly changed by wide variations in product temperatures and pressures and that the seal wear need not be excessive when using this method of cooling.

Supp. 317

Heyn, W. O.
Shaft Surface Finish Is an Important Part of the Sealing System
ASME Paper No. 67-WA/Lub-6, November 1967

Current specifications for shaft finishes to be used with shaft seals are inadequate. A new shaft finish specification is proposed which specifies a process to obtain a satisfactory sealing surface. A special method for checking for machine lead, generated by plunge grinding, is recommended.

Supp. 318

Hirabayashi, H.; Kato, Y.; and Ishiwata, H.
Excessive Abrasion of Mechanical Seals Caused By Salt Solutions
Third International Conference on Fluid Sealing, April 1967, Paper B1

The present investigation is concerned with the problem of excessive abrasion of mechanical seals caused by salt solutions. The cause of excessive wear of a carbon ring is evidently abrasion by crystallized particles of salt between sealing faces. The conclusions of this investigation are as follows: the progress of excessive abrasion is mainly influenced by hardness of salt.

Supp. 319

Hirabayashi, H.; Oka, K.; and Ishiwata, H.
The Relationship Between Ringing, Heat Transfer and Sealing Condition
Paper No. FICFS-13, May 1969

It is observed that mechanical seals sometimes generate a "ringing" sound under certain operating condition. The present investigation is concerned with a fundamental approach to "ringing" from the view-points of dynamic instability, heat transfer and interfacial phenomena. The conclusions are summarized as follows: the "ringing" or vibration of a mechanical seal is caused by the sealed liquid boiling between, or near, the sliding surfaces.

Supp. 320

Hirano, F.
Dynamic Inverse Problems In Hydrodynamic Lubrication
Third International Conference on Fluid Sealing, April 1967, Paper F1

The author solved the inverse problems in the hydrodynamic lubrication considering the wedge and squeeze terms simultaneously. The indeterminate solution is found to be unstable one. The unique stable solution is given by the critical profile which occurs under the pressure distribution with a point of inflexion on its ascending side. As examples of the dynamical problems the variations of the film profiles under typical pressure distributions were numerically calculated.

Supp. 321

Hirano, F. and Kaneta, M.
Dynamic Behavior of Flexible Seals for Reciprocating Motion
Paper No. FICFS-2, May 1969

The inverse problems of hydrodynamic lubrication for reciprocating motion are dealt with assuming three typical pressure distributions: the parabolic distribution, the Gaussian curve, and the distribution with zero curvature at the centre. It is pointed out that there is a critical value of the ratio of the width of contact to the stroke to establish stable periodic profiles.

Supp. 322

Hirano, F.
Starting Torque of a Mechanical Seal
Second International Conference on Fluid Sealing, BHRA, April 1964, Paper D2

This paper deals with the problem of the starting torque of mechanical seals used for centrifugal pumps of small power. The mechanical seal composed of stainless steel and carbon shows an excessive starting torque after a standstill for long time, when its internal space is filled with water.

Supp. 323

Hodgson, J. W. and Milligan, M. W.
Visco-Type Gas Sealing
Paper No. FICFS-34, May 1969

The performance of visco-type gas seals operating in the continuum flow regime has been investigated both experimentally and analytically. Experimental values of the sealing coefficient agree with a laminar flow analysis within experimental error(10%).

Supp. 324

Holan, Darel
Sealing In Engineering
Second International Conference on Fluid Sealing, April 1964, BHRA Paper E5

After a general discussion of clearance seals and contact seals the theory and design of screw-type clearance seals is treated in detail. The main part of the paper is then devoted to a hydrostatic seal design with calculations of the flow of liquid through the seal and the friction losses.

Supp. 325

Hooke, C. J. and O'Donoghue, J. P.
Elastohydrodynamic Lubrication of High Pressure Face Seals
Journal of Mech. Eng. Science, Vol. 10, February 1968, Page 59-63

Description of a method for the solution of the elastic problem in determining the stability of high pressure face seals. This method produces close agreement with the published results obtained experimentally, and demonstrates that the simplified treatment is valid only if the seal has a large face to thickness ratio.

Supp. 326

Hooke, C. J. and O'Donoghue
The Elastohydrodynamic Lubrication of High-Pressure Face Seals
Paper No. FICFS-16, May 1969

The paper presents results for square section high pressure face seals mounted on rigid housings. Three common forms of seal arrangement are investigated and it is found that in the case of one of these, there is a region of instability of operation where the stiffness can be either positive or negative. Design charts for the load and leakage given.

Supp. 327

Hooke, C. J.; Lines, D. J.; and O'Donoghue
A Theoretical Study of the Lubrication of Reciprocating O-Ring Seals
Third International Conference on Fluid Sealing, April 1967 Paper F4

This paper is a survey of solutions to the elastohydrodynamic lubrication problem of an O-ring sliding in a cylinder. The paper extends the theory presented in a previous paper by the authors in as much as the entrace effect to the contact zone is taken into account. Non-dimensional results for a range of nip values are given which enable a theoretical estimate of the side load, friction and oil film thickness to be made.

Supp. 328

Hopp, H.
Seals for Hydraulic Equipment with Fire-Resistant Fluids
Paper No. FICFS-4, May 1969

Fire-resistant fluids are applied in hydraulics to an ever increasing extent. FR-fluids demand better chemical resistance of seals. Long term development, not only in the laboratory and in the test field, but first of all in practice became necessary in order to find the best seals for all the fluids on the market.

Supp. 329

Huhn, D.
Carbon Ring Seals for Larger Shaft Diameters
Second International Conference on Fluid Sealing, April 1964, Paper E2

This paper reviews the development of some mechanical seals for hydroelectric power-station turbine seals. The particular problems met and the way in which these were overcome in successive seal developments are described.

Supp. 330

Humble, S.
Glandless Pumps: Types Available and Their Application
Second International Conference on Fluid Sealing, April 1964, Paper G4

This paper describes various glandless pumps and attempts to classify them. In each case the principle of operation is described, reference is made to the advantages and limitations of each type for specific applications.

Supp. 331

Ishiwata, H. and Hirano, F.
Effect of Shaft Eccentricity on Oil Seal
Second International Conference on Fluid Sealing, April 1964, Paper H2

Leakage through the oil seal is easy to occur owing to the wobbling caused by shaft eccentricity. The subject has been experimentally investigated. From this results, the followable limit of oil seal lip with the eccentric motion was made clear.

Supp. 332

Iwanami, S., Ueda, K., Imamura, T, and Tikamori, N.
Frictional Characteristics of O-Rings for a Rotating Shaft
Third International Conference on Fluid Sealing, April 1967, Paper G1

An O-ring is one of the simplest seals and widely used in industry to prevent oil leakage at reciprocating surfaces or fixed surfaces of machines. The pressure distribution, temperature of the contact part and frictional torque were measured and frictional characteristics of O-rings were investigated. Also, some running tests were carried out and 'seizure' phenomena are discussed.

Supp. 333

Jackson, C.
Practical Guide to Mechanical Seals
Hydrocarbon Processing v 47 n 1 January 1968

Seals described are used primarily on pumps, though same approach might apply to compressors, mixers, agitators, blowers, or extruders; actual plant practices from Monsanto Co in use of mechanical seals for many services and operating conditions are described; included are 23 seal arrangements (shown by diagrams) which Monsanto has found to work under most hydrocarbon processing plant operations.

Supp. 334

Jacobs, E.
How You can Improve Your Selection of Seals & Packings
Hydraulics & Pneumatics Vol. 21, April 1968

A summary of questions and answers between seal and packing manufacturers and users. Hints on design, use of materials and specific application suggestions are included.

Supp. 335

Jagger, E. T.
Positive Action Seals in Europe
ASME Paper No. 67-WA/Lub-2, Winter Annual Meeting, November 1967

This paper presents a historical survey of the development of the Positive Action seal and outlines the current situation. Reference is also made to the "Duo Positive" seal (positive action seals which operate equally effectively for both directions of shaft rotation).

Supp. 336

Jagger, E. T. & Halliday, G. W.
The Measurement of Radial Load Between Sealing Lip and Shaft
Second International Conference on Fluid Sealing, Paper H3, April 1964

The paper reviews the historical development of methods for measuring the radial loads between a sealing lip and its shaft - one of the most important criteria in seal design. The problems which arise are discussed, the shortcomings of existing methods are pointed out and a description of the authors' preferred method and apparatus is given.

Supp. 337

Jagger, E. & Walker, P.
Further Studies of Lubrication of Synthetic Rubber Rotary Shaft Seals
Instru. Mech. Engs-Proc V.181, 1966-67

From theoretical considerations of rubber asperity treated as elastohydrodynamic problem and from experimental investigation, it is concluded that an oil film known to exist between rubber sealing lip and shaft at point of contact is able to carry radial load by virtue of microasperity lubrication mechanism in which deformed asperities act as microbearing pads.

Supp. 338

Johansson, R. and Malmstrom, S. E.
The V-Ring, A New Shaft Seal
Third International Conference on Fluid Sealing, Paper D4, April 1967

The designer has many different types of seals to choose from, all having special advantages and disadvantages. The selection of the right type of seal is very difficult and mistakes have caused much trouble in the industry. The V-ring is a new type of seal. It consists of rubber, rotates with the shaft and seals with its flexible lip against a plane surface at right angle to the shaft.

Supp. 339

Johansson R. and Malmstrom, S. E.
The Perforated V-Ring A New Oil Seal
Paper No. FICFS-9, May 1969

This paper describes a new principle of oil sealing. The basic experiments have been carried out on standard V-rings, the lip of which has been perforated with a little hole. The task of the hole is to drain the leakage, if any, from the space between the V-ring lip and the sealing surface that the lip slides against.

Supp. 340

Johnson, R. L. and Ludwig, L. P.
Shaft Face Seal with Self-Acting Lift Augmentation for Advanced Gas Turbine Engines
Paper No. FICFS-27, May 1969

The need in advanced engines for shaft seals with improved pressure, temperature, and speed capabilities is pointed out. In particular, the high-speed requirement suggests that the seals must not have rubbing contact. It is shown that incorporation of self-acting lift pads to the conventional face seal prevents rubbing contact even though face deformation occurs.

Supp. 341

Johnson, R. N.
Spiral Edge-Wound Gaskets
Second International Conference on Fluid Sealing, Paper A1, April 1964

Flange sealing problems arising from the ever increasing pressures and temperatures of super heated steam in the marine and electricity generating industries are considered. The versatility of the edge-wound gasket and its use in other circumstances are described.

Supp. 342

Jones, R.
Some Aspects of Sealing in Gas Turbines
Second International Conference On Fluid Sealing, Paper C1, April 1964

Fluid sealing in gas turbine aero-engines is a field in which continuing development is required and is in fact being carried out. This paper covers two applications of elastomer seals, i.e. lip type and 'O' seal, where solutions to particular problems have been obtained.

Supp. 343

Kambayashi, H. and Ishiwata
A Study of Oil Seals For Reciprocating Motion
Second International Conference on Fluid Sealing, Paper B3, April 1964

The frictional characteristics of oil seals for reciprocating motion were investigated experimentally. The testing apparatus was newly made, and tests were performed with a variety of lip materials, shaft surface finishes, oil viscosities, pressures and rubbing speeds.

Supp. 344

Kamal, M. M
A High Pressure Clearance Seal
ASME Paper No. 67-WA/Lub-18

A simple method for estimating leakage and friction losses for a high pressure sleeve type low-clearance metal seal is presented. Elastic deformations in the shaft and seal and the change in viscosity due to the high pressures are accounted for and found to be of major importance.

Supp. 345

Kaufman, Warner B., Tischler, Richard F., and Breitwieser, Roland
A High Temperature, Electrically Insulating Cermet Seal Having Strength and Thermal Conductance
IEEE Conference, October 30-November 2, 1967, Paper

Description of high-temperature cermet seal developed for a research thermionic diode. The seal consists of one or more layers of alumina-coated niobium particles sandwiched between thick niobium pieces and bonded by isostatic hot pressing. The hermetic seal has a resistivity greater than 5MΩ at 1300°K, structural integrity to 1700°K, and exceptional thermal shock resistance to 1500°K.

Supp. 346

Kauzlarich, J. J. and Neuhauser, W. E.
Friction and Wear Characteristics of Hot Pressed MoS_2-Ni Composites in a Face Seal Configuration
Paper No. FICFS-23, May 1969

Experimental friction and wear tests for hot pressed MoS_2-Ni composites were run in air and results showed that increasing the percent MoS_2 in a composite decreased the coefficient of friction with a limiting value equal to that of a 100% MoS_2 film. However, the wear rate was greater than that for pure nickel.

Supp. 347

Ketola, H. N. and McGrew, J. M.
Theory of the Partially Wetted Rotating Disk
Third International Conference on Fluid Sealing, Paper H4, April 1967

Interest in the performance of the partially wetted rotating disk has recently increased due to the application of this device as a component in non-contacting dynamic seals for space power systems. This paper presents a theory which can be used to predict pressure, drag, and flow characteristics of the partially wetted rotating disk.

Supp. 348

Ketola, H. N. and McGrew, J. M.
Pressure, Frictional Resistance, and Flow Characteristics of the Partially Wetted Rotating Disk
ASME Paper No. G7-WA/Lub-4, Winter Annual Meeting, November 1967

A theory of the partially wetted rotating disk is described and experimental data presented which verify the application of this theory in practical applications. Experimental results are presented.

Supp. 349

Kibble, J. D. and Leahy, J. C.
Hydraulic Sealing Problems in Mining Equipment
Second International Conference on Fluid Sealing, Paper G2, April 1964

The paper outlines mining applications and describes successful solutions to design problems, mentioning relief valves and other valves, ram seals, hose joints and gaskets.

Supp. 350

King, A. L.
Bibliography on Fluid Sealing
BHRA, Essex, England, 1962

Supp. 351

Kitching, R. and Gleaves, J. S.
Static Friction Between a Cup Leather Seal and A Cylinder Wall
International Journal of Mechanical Sciences, Vol. 10, October 1968

If a hydraulic jack used in the vertical position has a cup leather seal and the pressure is released after being held for a length of time, there will be a significant static friction force required to move the ram in the cylinder. The cause of the friction force has been studied experimentally, as well as some of the variables which control its value. The subject is of considerable importance in the use of free-fall jacks as hydraulic pit props.

Supp. 352

Kojabashian, C. and Richardson, H. H.
A Micropad Model For The Hydrodynamic Performance of Carbon Face Seals
Third International Conference on Fluid Sealing, Paper E4, April 1967

A mechanism for the apparent hydrodynamic performance of mechanical face seals is advanced, based on surface micropads. Methods are described for observing the surface topography of carbon face seals using inclined light, polished sections, and Tallysurf traces. It is concluded that the micropad hypothesis deserves further study since it predicts the correct order of performance characteristics, is consistent with the fact that multiphase materials promote good performance, and permits performance to be stable over extreme ranges of wear and elastic deformation.

Supp. 353

Komotori, Kazunari
Leakage Characteristics of "Straight-Through" Labyrinths With Axial Motion
Third International Conference on Fluid Sealing, Paper H1, April 1967

This paper deals with the leakage characteristics of straight-through labyrinths, the throttling fins of which move perpendicularly to themselves. Problems dealt with under this heading appear in the seal mechanisms of labyrinth piston compressors, rotating piston machinery and the rotary type of regenerative heat exchanger. The leakage characteristics in these cases usually depend upon both the normal labyrinth effect and the displacement effect, which means that the fluid contained in the labyrinth pocket is carried over by the movement of the pocket itself.

Supp. 354

Kotte, G.
Axial-sliding Ring-seals for Idlers and Gearings of Crawler Vehicles and Their Functional Test
Foerdern u Heben v 18 March 1968

Various German foreign axial-sliding ring-seal systems are described and their more important features discussed; considering effects of precision-machined sealing faces, axial springing and radial tension, prospects for development of synthetic sliding rings appear to be particularly good. In German.

Supp. 355

Kuzma, D. C.
Theory of the Mechanism of Sealing with Application to Face Seals
Paper No. FICFS-18, May 1969

A theory of the mechanism of sealing is presented. This theory accounts for the load carrying capacity of the seal and allows a pressure gradient to be maintained across the seal with no leakage. The theory is applied to a simple face seal.

Supp. 356

Landgrof, E.
Poly (tetrafluorethylene) Seals
Plaste Kaut. 1968, 15 (7)

Various designs of seals and gaskets of poly-(tetrafluorethylene) I (teflon) and reinforced I materials and discussed.

Supp. 357

Lawrence, R. T.
Pneumatic Seal Friction
Third International Conference on Fluid Sealing Paper C1 April, 1967

An introduction to seal friction in industrial pneumatic equipment. The factors which determine friction and the effects upon reliability and performance are discussed. Mention is made of present and future seal materials and types, bore materials and coating and lubricants.

Supp. 358

Lawrie, J. M. and O'Donoghue, J. P.
The Mechanism of Lubrication in a Reciprocating Seal
Second International Conference on Fluid Sealing Paper B 6, April 1964

The lubrication and leakage mechanism existing in reciprocating seals used in automotive brake and clutch hydraulic circuits has been investigated using conduting rubber and high speed techniques.

Supp. 359

Lindley, P.
Compression Characteristics of Laterally-Unrestrained Rubber O-Rings
Instru. Rubber Industry, July-August 1967

Toroidal rubber rings are widely used in engineering as static seals, in particular for providing flexibility in joints of underground pipe lines. The effectiveness of the seal depends upon contract stress and coefficient of friction between rubber and restraining surfaces. Joint design must take these and other factors into consideration.

Supp. 360

Love, B. E. and Palmer, K. P.
Stress Relaxation Testing of 'O' Rings For High Temperature Sealing
Second International Conference on Fluid Sealing Paper H4, April 1964

Measurements have been made of the actual forces exerted by 'O' ring seals under conditions as near as possible to those in normal sealing practice, but in air only. Relaxation results for nitrile, silicone and Viton materials, are compared with compression set values.

Supp. 361

Lymer, A.
An Engineering Approach to the Selection and Application of Mechanical Seals
Paper No. FICFS-25, May 1969

A method of seal selection is described using the criterion of vapourisation of the liquid component between the seal faces. Heat generation by friction at the seal faces is considered in relation to the heat

dissipation characteristics of balanced and unbalanced seals of different sizes in different materials operating on different liquids.

Supp. 362

Lymer, A. and Robinson, R. P.
The Development of An Outboard Seal For Submersible Drive, Deep Well Drilling Units
Third International Conference on Fluid Sealing, Paper B4, April 1967

A recent development in the drilling of deep wells has been the introduction of submersible drive units on which the rock bits are directly mounted, thus avoiding the use of a surface drive. This has created the need for a rotary shaft seal which can exclude mud whilst retaining oil, with the minimum of leakage and operating under severe vibratory conditions. The development of such a seal is described.

Supp. 363

Lynch, D. P.
Gasket Materials
Second Internation Conference on Fluid Sealing, Paper A2, April 1964

Within the scope of this paper, an attempt will be made to describe the major classes of gasket materials together with explanation of the technical terms employed in this trade.

Supp. 364

McCleary, G. P.
Sealing the Cryogens with Hard Carbide Alloys
Lubrication Engineering, July 1968

This paper is a discussion of the date of the art of the carbides as used in seal rings at cryogenic temperature ranges. In this range, some materials retain their room temperature properties quite well while others become extremely brittle. Laboratory tests have been made to determine some of the properties of carbides at these cryogenic temperatures.

Supp. 365

McGowan, M. T. A.
The Development of a Rotary Seal For the Large CO_2 Circulation Used At The Trawsfynydd Power Station
Second International Conference on Fluid Sealing, Paper E1, April 1964

In the present series of nuclear power stations being built the CO_2 coolant is usually circulated by axial flow fans of large unit size. The use of this type of circulator which entails a high rubbing speed for the seal and the necessity of maintaining minimum oil leakage into the coolant to avoid contamination imposes severe limitations in the choice of seal. This paper describes the design and development of such a seal.

Supp. 366

McKibbin, A. H. and Parks, A. J.
Aircraft Gas Turbine Mainshaft Face Seals-Problems and Promises
Paper No. FICFS-28, May 1969

Current labyrinth and rubbing-contact sealing devices demonstrate limited growth potential in performance to withstand the increasingly severe environments imposed by advanced aircraft gas turbine. Various research programs show that major improvements in mainshaft seals can be made through the use of film riding seals.

Supp. 367

Mal'chikova, E. Y., et al
Increasing the Life of Seals Functioning In Contact With Oils
Kauch. Regina 1968, 27 (3)

Addition of various organic compounds including neozone D and acetoaniline to lubricating oils in contact with synthetic rubber seals increased the resistance of the rubbers to degredation.

Supp. 368

Mallair, F. R., Nelson, L. H. and Buckmann, P. S.
Evaluation of Wear Ring Seals for High-Speed, High-Pressure Turbopumps
Trans. ASME J. or Lub. Tech. 68-WA/Lub-1

This paper describes a seal evaluation program for application as pump impeller wear rings for high-pressure, high-speed liquid rocket turbopumps. The principal design requirements were to control the leakage to values consistent with efficiency requirements and to eliminate potential explosions resulting from high velocity rubbing at the required clearances.

Supp. 369

Mayer, E.
Design and Construction of Shaft Ring Seals
Konstruktion v 20 n 6 June 1968

Optimum construction and dimensions of ring seals depend on numerous operational factors such as leakage, service life, operation reliability, friction losses, space available, and economy of operation; these six factors and effects of their mutual superposition are discussed and illustrated by examples of latest developments in field of seals with reduced pressure, high-pressure seals, hydrodynamic seals, etc. In German.

Supp. 370

Mayer, E.
Thermohydrodynamics In Mechanical Seals
Paper No. FICFS-14, May 1969

Seal rings of inhomogeneous materials as leadbronze, alloys, phenolic- and metal impregnated carbons show hydrodynamic effects due to different heat conductivities and coefficients of expansion. Fundamental parameters of thermohydrodynamic mechanical seals with circular grooves and field experiences with 2000 seals and 40-50 millions service hours are discussed.

Supp. 371

Mentley, A. A., and Le Roy, W. W.
Identification of Elastomers Used in Hydraulic Sealing Applications
Lubrication Eng v 23 n 12 December 1967

Identification techniques for synthetic elastomers used in hydraulic sealing elements permit rapid examination of new and used elastomers; method utilizes internal reflectance attachment to infrared spectrophotometer; emission spectrographic analysis provides valuable corroborating data; sample preparation, equipment description, and data interpretation.

Supp. 372

'67 Product Progress Report-Materials for Sealing
Mill & Factory v 81 October 1967

New Materials for Packings (seals and gaskets) are polytetrafluoroethylene, polyurethanes, and fluoroelastomers which have temperature resistance of up to 450oF and will take hot oils; new anaerobic sealant hardens automatically in absence of oxygen; anaerobic sealants produce seals which are as strong as confining piping.

Supp. 373

Moran, E. F., Jr.
Some Limitations of Shaft Speed on Close-Clearance Seals
ASME and ASLE Lubrication Conference, October 8-10, 1968, ASLE Paper 68 LC-13

Results of a study of permissible shaft speeds on close-clearance seals. Some test data and analysis are presented to support the conclusion that high shaft speeds can effect an excessive change in fluid viscosity within the seal which causes seal contact and eventual damage. In order to overcome this speed limit, increased operating pressure is required with its attendant higher flow rate.

Supp. 374

Mottram, A. W. T. and Sunley, H. L. G.
Seals For Liquid Propellant Rocket Engines
Second International Conference on Fluid Sealing Paper C4, April 1964

Liquid propellant rocket engines depend heavily on seals for their safe and efficient operation. In this paper, the special conditions in rocket engines are described which affect the selection of seals.

Supp. 375

Muller, Heinz K.
Leakage and Friction of Flexible Packings at Reciprocating Motion With Special Consideration of Hydrodynamic Film Formation
Second International Conference on Fluid Sealing, Paper B2, April 1964

By sealing reciprocating rods with soft flexible packings, esp. O-Rings, the hydrodynamic fluid film formation is dependent on several parameters such as viscosity, velocity, pressure, hardness and form of sealing rings. It can be shown theoretically and by experiment that the thickness of the fluid film between the mating surfaces is also essentially dependent on the direction of motion relative to the direction of pressure acting on the seal.

Supp. 376

Muller, H. K.
Self Aligning Radial Clearance Seals
Third International Conference on Fluid Sealing Paper H6, April 1967

Balance, leakage and power loss of self aligning radial clearance seals are functions of the seal face geometry and of the by-pass flow into the clearance. The paper presents design data over a practical range of parameters. Working diagrams are given for a rapid calculation of the seal characteristics. Experimental results with a buffer gas test seal are discussed.

Supp. 377

Muller, R. K.
Externally Pressurised Barrier Shaft Seals
Paper No. FICFS-37, May 1969

Describing various design concepts for externally pressurized barrier shaft seals the paper deals with balance conditions, flow rates, power consumption and temperature rise, axial stiffness and secondary seal influence. Design directives and analysis is presented for seal geometries which are expected to have minimum distortion both from pressure and thermal stresses.

Supp. 378

Nau, B. S.
Hydrodynamics of Face Seal Films
Second International Conference on Fluid Sealing, Paper F5, April 1964

Various workers have shown that rotary face-seals normally function with a fluid film between the mating surfaces of the seal. In this paper mathematical analysis is presented to cover several alternative mechanisms which could be responsible for maintaining the fluid film.

Supp. 379

Nau, B. S.
Modified Packed Gland Designs
Third International Conference on Fluid Sealing, Paper A 4, April 1967

Factors affecting stuffing box performance are discussed in the light of previous experimental and theoretical work. The desirability of a more uniform contact stress distribution in the packing is established and means by which this may be achieved are considered, one design was selected for testing. There appear to be distinct advantages in using a radially loaded stuffing-box design, both for compactness and a number of other reasons.

Supp. 380

Nau, B. S.
Film Cavitation Observations In Face Seals
Paper No. FICFS-20, May 1969

With the aid of transparent seal faces the cavitation of face seal films has been studied experimentally and photographs obtained under various operating conditions are presented. The results provide evidence indicating the nature of seal film lubrication processes. It is found that both angular misalignment and surface waviness are prime causes of hydrodynamic pressure generation.

Supp. 381

Ng, C. C. W., McKibbin, A. H., and Parks, A. J.
Performance of the Orifice-Compensated Hydrostatic Face Seal Under Pressure and Thermal Loading
68-WA/Lub-6

Dynamic tests were run on a 7-india orifice compensated face seal design for an advanced aircraft gas turbine application. The test results were used together with an analytical model of the seal interface air passage to evaluate seal distortion. These distortion values were compared with seal distortion estimates produced by detailed thermal and structural analysis of the seal design.

Supp. 382

Orcutt, F. K.
An Investigation of the Operation and Failure of Mechanical Face Seals
Paper No. FICFS-22, May 1969

This paper describes some results of an investigation aimed at providing further insight into the mechanics of seal operation. A particular objective was to investigate the process whereby normal, effective sealing action breaks down so that the seal begins to leak.

Supp. 383

Osola, V. J.
Experiences in the Use of Mechanical Seals in Pumps and Agitated Pressure Vessels
Second International Conference on Fluid Sealing, Paper D3, April 1964

The paper describes some experiences in the application of mechanical seals to pumps and pressure vessel agitators in a chemical manufacturing company.

Supp. 384

Pape, J. G., and Vrakking, W. J.
Viscoseal-Pressure Generation and Friction Loss Under Turbulent Conditions
ASLE, May 6-9, 1968, Preprint 68AM 5B-1

A new theory is developed that predicts the sealing coefficient and the friction factor of a viscoseal under turbulent flow conditions. The results have been given in simplified formulas such that they are readily usable for design purposes. Existing theories on the turbulent behavior of the viscoseal are briefly discussed.

Supp. 385

Parry, J. S. C.
Development of Reciprocating Seals for Very High Pressures
Third International Conference on Fluid Sealing, Paper C6, April 1967

This paper describes the design and development of a reciprocating seal for a high pressure fatigue machine. Two seals were eventually developed for use at pressures up to 25 tonf/in^2.

Supp. 386

Pechko, M., and Khmel'nikes, V.
Elastic packing for High-Pressure Equipment Seals
Khim. Neft. Mashinostr., 1968 vol.(8)

The heat stability of materials useful as elastic fillers for high pressure equipment seals for work in the temperature range 650-700° was studied. The materials include asbestos, rubber, polymers, wools and fibers.

Supp. 387

Platt, A.
The Development of Glandless Pumping Systems
Second International Conference on Fluid Sealing, Paper G1, April 1964

Systems comprising glandless electric motors coupled to pumps were evolved originally for circulating high pressure, high temperature boiler water and for nuclear energy purposes. Further development into the chemical and process industries then followed.

Supp. 388

Quansah, W. A.
"A Fresh Approach Into The Problem of Compressed Asbestos Fibre Gaskets Used In Narrow-faced Bolted Joints"
Third International Conference on Fluid Sealing, Paper C5, April 1967

Cyclically varying internal pressures have been applied to a flange assembly which comprised of two B.S.S. 778 : 1951 table 13 flanges, made of mild steel and in which the compressed asbestos fibre (C.A.F.) gasket was entirely within the bolt circle. Among several parameters studied the following are reported: the Poisson's ratio, the gasket stress distribution across the gasket on initial tightening and the flange rotation coefficient.

Supp. 389

Quansah, W. A.
Prediction of the Penetration Pressure in Bolted Flanged Joints
Paper No. FICFS-10, May 1969

The present design techniques for flanged joints are based on mean ring loads acting on the seal which is usually made of compressed asbestos fibre. Also the investigations showed that certain phenomena during pressuration of the joint were inexplicable. An approximate elasticity solution has now been used for a limited number of tests to predict the stress at the inside radius of the gasket.

Supp. 390

Rabson, John
The Piston Ring - A Reciprocating Seal For Automotive Engines
Second International Conference on Fluid Sealing, April 1964
Paper B 4

The operating environment and functions of piston rings are discussed in some detail. Important factors affecting the design of piston rings are considered and special attention is given to control of oil losses past the piston.

Supp. 391

Rapson, J. E.
Research & Development
Aircraft Engineering, Vol. 40, May 1968, p. 15-20

Description of research and development methods used to construct cushion systems for hover craft. Testing facilities and procedures are presented, the aim of which is the achievement of the following requirements: (1) low drag characteristics of cushion sealing system, (2) durable & easily replaceable seals, (3) accessible seal components, & (4) the sealing system should be capable of being designed to provide predetermined cushion characteristics.

Supp. 392

Reshotko, F., and Rosenthal, R. L.
Fluid Dynamic Considerations in Design of Slinger Seals
Lubrication Eng. v 24 July 1968

Analytic approach is developed which treats flow in partially engaged slinger seal as portion of flow between two closely spaced infinite disks, one rotating and other stationary; analysis is for laminar flow; preliminary version of present formulation is extended to include best available information on laminar flow between infinite disks.

Supp. 393

Rhodes, Allen F.
An Experimental Study of Lip-Type Annular Elastomer Seals
Second International Conference on Fluid Sealing, Paper A 5
April 1964

In the production of petroleum, it is often necessary to make high pressure seals in the field around tubular goods manufactured to wide diameter tolerances. The rough surface of these tubes precludes the use of line contact seals such as O-rings. This paper gives the results of an experimental study of the effect of lip and ring design on performance of the seals.

Supp. 394

Richards, M.
The Frictional Characteristics of a Water-Lubricated Rotary Seal Employing A Soft Seal Ring in Eccentric Engagement With A Hard Counterface
Third International Conference on Fluid Sealing Paper A 1, April 1967

The paper gives an account of experimental work in support of the development of a rotary face seal for water. The seal employs a seal ring of a conformable material in place of carbon. A turn of gland packing and a ring out from a sheet of rubber were used. At lower speeds boundary lubrication was evident, with friction comparable with that found in carbon type face seals.

Supp. 395

Robinson, R. P. and Burton, R.
Temperature Gradients in Seal Rings: A Method of Analysis
Paper No. FICFS-12, May 1968

A numerical method, based on the well known relaxation technique, is described for determining temperature distributions in seal rings. The method is one which can be applied to seal rings of non-uniform cross sectional shape and is particularly suitable to digital methods of solution.

Supp. 396

Rogers, P. S., Butler, J., and Steele, B. C.
Technique for Sealing Molybdenum to Alumina or Platinum to Alumina by a Glass-Ceramic Seal
Jo. Sci. Intr. 1969, Vol. 2

A simple technique for sealing Al_2O_3 tubing or plates to Pt or Mo was devised which has excellent vacuum properties at 1000°. A glass to metal seal is first made and this is then converted to a glass ceramic seal by controlled heat treatment.

Supp. 397

Rosensweig, R. E., Miskolczy, G. and Ezekiel, F. D.
Magnetic-Fluid Seals
Machine Design, Vol. 40, March 28, 1968, p. 145-150

Description of a new type of shaft seal which uses a magnetic fluid (ferrofluid) contained by a magnetic field to effect the seal. Two seal configurations which use these fluids are illustrated, and their advantages and limitations are outlined. Properties of magnetic fluids and their performance characteristics are discussed. Possible future applications of ferromagnetic liquid shaft seals are considered.

Supp. 398

Roth, A.
Sealometry and Sealography
Third International Conference on Fluid Sealing, Paper C2, April 1967

Sealometry is a method developed by the author for expressing, measuring and specifying the sealing ability of gasket materials. Sealography is a technique derived from this method to record the history of the sealing process of actual seals. The present paper describes the principles of these methods and their measuring devices (sealometer, sealograph) and reviews the basic formulae derived to describe the sealing process and predict the performance of seals.

Supp. 399

Roth, A. and Inbar, A.
Analysis of Vacuum Sealing Process between Turned Surfaces
Vacuum v 18, June 1968

Technique developed to record and interpret history of sealing processes is applied to vacuum seals with surfaces having two different roughness profiles-radial and circumferential; comparison of cases of concentrically and helically machined surfaces; equations for characteristic curve of sealogram.

Supp. 400

Samoiloff, A. A.
Mechanical Seals - Longer Runs, Less Maintenance
Chem. Eng v 75, January 29, 1968

Survey includes basic components of mechanical seal, primary seal's dependence on sealing liquid, properties of sealing liquid, balancing inside and outside seals, double seals and lip seals, circulating ring for seal cooling, water jacketing and cyclone separators, functions for throat and throttle bushings, vacuum and crystallization problems, and thermal aspects of mechanical seals.

Supp. 401

Scannel, John B.
Specification of V-Packing Dimensions For Fixed Length Gland Applications
Second International Conference on Fluid Sealing, Paper B1, April 1964

The paper covers work done on formalizing methods of determining and specifying actual working height measurements for stacked molded packing rings. The relationship of working height (compression) to leakage and friction is outlined for a typical set of conditions.

Supp. 402

Schmitt, W. A.
Radial Load as a Lip Seal Design & Quality Control Factor
ASME, WA, 67-WA/Lub-14

Description of a new device for measuring the radial load in the laboratory and quality control. Equations derived for the evaluation of the radial load and the influence of materials and seal dimensions on the radial load is shown. Obtainable production tolerances for the radial load of a seal is discussed.

Supp. 403

Schnetzer, E. and Rossbach, R. J.
Development of a Liquid Dynamic Seal to Vacuum
Paper No. FICFS-40, May 1969

In the development of the advanced Rankine space power system technology the rotating machinery is to run on potassium-lubricated bearings. The problem investigated was the dynamic seal between the pressurized liquid potassium in the bearing cavity and the vacuum in the alternator rotor cavity. The objective was to keep the leakage rate at this interface below 10 lb/yr.

Supp. 404

Scholten, G. J.
A Novel Type of Bidirectional Helix Seal, The R-L Seal
Paper No. FICFS-6, May 1969

A hydrodynamic "Helixseal" for rotating shafts - or bearing bushings - suitable for both directions of rotation, is possible with the aid of hydrodynamic friction in the lubricating oil (or grease), the elastic properties of the rubber and a special groove pattern.

Supp. 405

Scibbe, H. W.
Bearings and Seals For Cryogenic Fluids
SAE, Aeronautic and Space Engineering and Manufacturing Meeting, October 7-11, 1968, Paper 680550

Description of the material requirements and lubricating techniques for rolling element bearings and face contact seals in cryogenic propellant turbopumps. Material compatibility and wear are the important factors for face contact seals in these cryogenic fluids. Carbon, normally used in seal nosepieces, has violent reactions when run in liquid fluorine, an extremely chemically active fluid. Wear at the rubbing contact is minimized when the seal is designed for positive face separation with acceptable leakage.

Supp. 406

Selfors, Henry J.
The Toroidal Rotary Regenerator For Gas-Turbine Engines
American Helicopter Society, May 8-10, 1968, Paper 213

The background of the problem of sealing regenerators for aircraft engines is presented. The principal design features of the toroidal rotary regenerator and the advantages of the design with flexibility are discussed. It is concluded that the toroidal rotary regenerator can be effectively sealed, is structurally durable, and offers the designer wide latitude in matrix selection.

Supp. 407

Shapiro, W.
Concepts for Deep Submergence Hydrostatic Shaft Seals
Lubrication Eng v 24, July 1968

Three main shaft seal system concepts for deep submergence vessels are presented; sealing is accomplished by balanced face seal buffered by fluid that is pressurized to value slightly greater than seawater regardless of operating depth; general configuration, theory of operation and performance characteristics of h-p, hydrostatic, increasing film thickness seal, h-p, hydrostatic, decreasing film thickness seal, and intermediate-pressure, hydrostatic seal; comparative advantages and disadvantages of each concept.

Supp. 408

Simpson, V. G.
Silicone Rubber in Automotive Radial Oil Seal Applications
SPI - Conference on Elastoplastics Technology March 23-24, 1966

Fundamental properties of silicone rubber and their applications to production of high quality radial oil seals for use on rotating shafts such as in automatic transmission and torque converters, gear boxes, internal combustion engines, and hydraulic systems; advantages of silicone rubber tip seals over those made of other materials; effect of environmental changes on silicone rubber performance has been rationalized, and direction of present and future investigations have been indicated.

Supp. 409

Smithers, B. C.
High Temperature Elastomeric and Metallic Seals For Aircraft Hydraulic Systems
Second International Conference on Fluid Sealing, Paper C3, April 1964

This paper concerns the problems encountered in the design and evaluation of present day elastomeric seals to cater for the increased operating temperature requirements of current aircraft hydraulic systems.

Supp. 410

Sneck, H. J.
Reversed Flow in Face Seals
ASME Paper 68-Lub-9 for meeting Oct. 8-10, 1968

Stream surfaces for laminar and turbulent flow in parallel surface seals are analytically determined for cases where fluid separates from one of solid boundaries; these stream surfaces are used to determine when there is exchange of external fluid across seal land as well as rate of these exchanges; criterion given by equations can be used to design face seal so as to avoid any tendency toward reversed flow and insure unidirectional leakage in two fluid case.

Supp. 411

Sneck, H. J.
Thermal Effects in Face Seals
ASME and ASLE, Lub. Conference, October 8-10, 1968, ASME Paper 68-Lub-14

The laminar and turbulent flow of an incompressible fluid between the rotating parallel surfaces of a face seal is investigated analytically to determine the influence of conduction, convection, and dissipation on the temperature distribution. A method of estimating the general temperature level within the seal is suggested.

Supp. 412

Sneck, H. J.
The Effects of Geometry and Inertia on Face Seal Performance
ASME, November 12-17, 1967, Paper 67-WA/Lub-16

The "short bearing" equation of lubrication theory, modified to include the inertial effects, is used to study the influence of geometric deviations from the ideal. The turbulent nature of the flow is described by an isotropic apparent viscosity and a power-law velocity distribution. It is found that geometric deviations from the ideal are less influential than in laminar flow.

Supp. 413

Sneck, H. J.
The Effects of Geometry and Inertia on Face Seal Performance-Laminar Flow
ASME, November 12-17, 1967, Paper 67-WA/Lub-15

The "short bearing" equation of lubrication theory, modified to include inertial effects, is used to study the influence of geometric deviations from the ideal. It is found that the centrifugal force could be responsible for hydrodynamic features of the leakage flow which are theoretically unexplainable in the absence of this inertial effect. There is good evidence that the theory and the results are applicable over the entire laminar range of operation, provided the nominal clearance is small compared to the nominal radius.

Supp. 414

Sneck, H. J.
The Misaligned, Eccentric Face Seal
Paper No. FICFS-15A, May 1969

The analytical investigation of a misaligned face seal shows that eccentricity will cause a component of the total leakage to be inward or outward depending on the phase angle between the eccentricity and the misalignment.

Supp. 415

Sneck, H. J.
The Eccentric Face Seal With A Tangentially Varying Film Thickness
Paper No. FICFS-15B, May 1969

It is shown analytically that face seal eccentricity combined with surface waviness can contribute an inward or outward leakage component to the total leakage flow depending on the phase angle between the eccentricity and the waviness.

Supp. 416

Sosonkin, V. L.
Dynamic Calculation of Annular Sealing Gaps
(Vestnik Mashinostroeniya n1 January 1967)

Generalized frequency characteristic of pressure variations in gap, and generalized transfer functions are obtained. An expression is derived for leakage rate. (In Russian)

Supp. 417

Strom, T. N., Ludwign, L. P. Allen, G. P. and Johnson, R. L.
Spiral Groove Face Seal Concepts - Comparison To Conventional Face Contact Seals in Sealing Liquid Solium (400 to 1000 DEG F).
ASME, November 12-17, 1967, Paper 67-WA/Lub-17

Conventional face contact seal performance was improved by incorporation of spiral-groove geometry. Both conventional face contact seals and seals with spiral grooves were used to seal liquid sodium at pressure of 20 psig (14.0 N/cm^2 gage), and a sliding velocity of 79 ft/sec (24 m/sec). In comparison with conventional face contact seals, seals with spiral grooves had negligible leakage. The wear and contact patterns indicated that the spiral-groove seal operated with separation of the sealing surfaces, which is necessary for long life.

Supp. 418

Symons, J. D.
Shaft Geometry - A Major Factor in Oil Seal Performance
ASME Paper No. 67-WA/Lub-7, Winter Annual Meeting, November 1967

Five seal variables, flex section thickness, lip length, trim diameter, material and lip force and two shaft variables, shaft out of round and number of lobes, were investigated by means of fractional-factorial experimentation to determine the effects of shaft out of round and no. of lobes on seal performance.

Supp. 419

Taylor, E. R.
Shaft Seal for Gas-Cooled Reactor Rotating Machinery
Lubrication Eng. v 24, January 1968

Second-phase shaft seal development program delineated necessary modifications to seal, developed for rotating machinery for Experimental Gas-Cooled Reactor; first-phase program was completed prior to establishment of all reactor operational criteria and to design of axial and centrifugal compressors into which seal was to be fitted; difficulties encountered in fitting multi-element seal into design of centrifugal machinery; use of visco-type seal elements in conjunction with, and as integral part of, floating bushings yielded solution.

Supp. 420

Taylor, E. R.
Preliminary Experiments with the Visco Seal-Pump Concept in Turbulence
Paper No. FICFS-33, May 1969

Perliminary experiments were conducted to determine the performance characteristics of large size visco seals functioning simultaneously as a gas seal and as a sealant fluid pump operating in turbulence. The visco seal-pump conceptual design, the use of the theory based on a static sealant, and the method of testing are presented with the results and evaluation.

Supp. 421

Theyse, F. H.
The Inverse Hydrodynamic Theory and Its Application in the Design of Controlled Leakage Seals Between Moving Parts
Third International Conference on Fluid Sealing, Paper F2, April 1967

A full lubrication film is known to exist between a dynamic flexible seal and its mating surface and this controls the leakage and wear of the seal. In this paper it is shown that leakage is entirely determined by the quantity $\bar{h}$, and the wear by h_{min}. Present day elastohydrodynamic lubrication does not provide information about these quantities. For a sufficiently flexible seal, however, the inverse approach to the lubrication problem made in this paper gives a solution to this problem.

Supp. 422

Thew, M. T.
Further Experiments on the Hydrodynamic Disc Seal
Paper No. FICFS-39, May 1969

The seal discussed is a form of centrifugal seal for application to rotating shafts. Experimental results on the rim torque absorption for thick discs are compared with predictions from the analogous pipe approach and data derived from long cylinders.

Supp. 423

Thew, M. T. and Saunders, M. G.
The Hydrodynamic Disc Seal
Third International Conference on Fluid Sealing, Paper H5, April 1967

The paper describes part of work in progress on a type of centrifugal non-contact shaft seal for use with liquids. Following an introduction giving the mode of action for the seal and a discussion of particular advantages or disadvantages, the main section is concerned with work to increase the sealing pressure whilst minimising the power absorbed, by systematically varying the geometry of the disc and housing.

Supp. 424

Upper, G. and Freudenberg, Carl
Temperature of Sealing Lips
Paper No. FICFS-8, May 1969

The life of a radial oil seal is very much influenced by the temperature in the contact zone of lip and shaft. Test results show how and why early failure will be caused by exceeding the temperature limit of the seal material or sealing fluid. The object of this paper is to determine the maximum temperature and the temperature distribution in a radial oil seal.

Supp. 425

Vilim, P.
Development of An Improved Face Type Mechanical Shaft Seal For High Temperature Pressurized Water Centrifugal Circulating Pumps
Third International Conference on Fluid Sealing, Paper B6, April 1967

Face type mechanical shaft seals used for high temperature pressurized water circulating pumps have been a source of problems which limited the continuous dependable operation of the pumps. Operating problems encountered with the conventional face type mechanical shaft seals as well as the underlying principles of operation and factors promoting failure are discussed.

Supp. 426

Vohr, J. and Chow, C. Y.
Theoretical Analysis of Spiral-Grooved Screw Seal for Turbulent Operation
Paper No. FICFS-32, May 1969

Various analyses of the operation of screw seals in the turbulent regime have been performed. In general, these have shown reasonable agreement with the experimental data obtained by their authors.

Supp. 427

Vreeburg, J. P. B.
Investigation of Laminar Flow Patterns and Pressure Generation in a Viscoseal Geometry
Univ. of Tennessee, Dept, of Mech. and Aerospace Engineering, March 1969

An approximate analytic solution is sought for the problem of the flow in an infinitely long viscoseal geometry under as few restrictions as feasible. The flow is assumed to be steady, laminar and fully developed. The fluid is considered to be of constant viscosity, incompressible and Newtonian.

Supp. 428

Walsh, R. H., Westcott, M. J. and Lydiard, W. G.
Metallic and Composite Seals for Jack Rods
Paper No. FICFS-3, May 1969

Work has been directed towards the development of dynamic gland seals for linear actuators capable of functioning at a temperature of 200°C which is beyond the capabilities of present day elastomeric seals. The results of the work indicate that metallic and loaded P. T. F. E. seals although somewhat inferior to elastomeric seals offer good prospects of usage under conditions which would be too severe for elastomeric materials.

Supp. 429

Ward, Ronald L.
Large Diameter Seals In a High Pressure, Sea Water Environment
Second International Conference on Fluid Sealing, Paper D4, April 1964

This paper deals with the evolution of propulsion shaft sealing systems in relation to extremes of fluid, mechanical, and operational environments. Differing design concepts as represented by face-type, circumferential, and buffered systems are compared and evaluated.

Supp. 430

Weinand
Helixseal - A Practical Hydrodynamic Radial Lip Seal
ASME Paper No. 67-WA/Lub-8, Winter Annual Meeting, November 1967

This report is a compilation of the work conducted in developing a practical hydrodynamic seal for automotive and similar applications. The initial investigation was limited to seals on a shaft that rotates in one direction and also applications which are semiunidirectional.

Supp. 431

Wilcock, D. F., Cheng, H. and Bjerklie, J.
Design of Floated Shoe Close Clearance Seals for Supersonic Jet Engine Compressors
ASME, November 12-17, 1967, Paper 67-WA/Lub-21

The increase in engine efficiency to be gained by substituting a floated close running air seal at the high-pressure end of large jet engine compressors, in place of the labyrinth seal, has prompted an intensive effort to design such seals for experimental evaluation. The paper describes the design of both a radial and an axial version in which multiple segmental shoes are floated 0.001 in. from the high-speed runner surface.

Supp. 432

Wilkinson, S. C. W.
The Application of Face Type Mechanical Seals to Superheated Water
Second International Conference on Fluid Sealing, Paper D1, April 1964

The paper outlines the problems encountered in the application of face type mechanical seals to superheated water, emphasizing the relative importance of stability at various positions in the vicinity of the seal. Design features which are essential to achieve ideal operating conditions are discussed.

Supp. 433

Wilkinson, S. C. W.
Large Diameter Shaft Seal Development
Third International Conference on Fluid Sealing, Paper B5, April 1967

This paper covers the development of large diameter radial face seals, and their application to marine propeller shafts.

Supp. 434

Winn, L. W., Thorkildsen, R. L., and Wilcock, D. F.
Design of One-Piece Jet-Engine Compressor End Seals
ASME Paper 68-LubS-14, June 1968

Performance of advanced air-breathing propulsion systems can be improved through use of effective sealing devices; one of areas in which immediate benefits are realizable is high-pressure compressor end area; designs of face-type flexible and of rigid seal for compressor end seal based upon cruise conditions expected in supersonic transport jet engine are described; seals, noncontacting during operation, operate on air films achieved through employment of hybrid gas-bearing geometries on seal faces; design procedure consists of selection of most applicable seal-face geometries force and moment balance and other factors which are outlined.

Supp. 435

Zuk, J. and Renkel, H. E.
Numerical Solutions for the Flow and Pressure Fields in an Idealized Spiral Grooved Pumping Seal
Paper No. FICFS-30, May 1969

A spiral grooved pumping seal model is formulated for laminar, incompressible flow for the limiting case of zero land clearance. Results are compared to classical models of the groove axial flow which neglect the coupling of the groove cross flow.

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