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AN EVALUATION OF GASKETS, SEALS, AND JOINTS FOR AEROSPACE HARDWARE

by F. M. Henson

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FOREWORD

A significant portion of the research described herein, conducted by Aerojet-General Corporation, Liquid Rocket Operations, was performed under NASA Contract NAS 3-2555 with Messrs. R. G. Willoh and J. M. Ladd, Chemical Rocket Division, NASA Lewis Research Center, as Technical Managers. Some of the data presented herein was generated as part of other Aerojet-General Programs. The report was originally issued as Aerojet-General Report No. 8800-39, 30 June 1966.

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ABSTRACT

This report is a compilation of surveys and investigations of static and dynamic seals as well as joints that were performed over a seven-year period. The report is divided into three basic sections: static seals; small tubing joints; and dynamic seals.

The static seals section contains pertinent and often forgotten design information for static joints, required installation and handling procedures for the seals and flanges, and the results of tests conducted with different types of gaskets and joints. It deals primarily with lightweight joint designs larger than 1.00-in. sizes.

The small tubing joint section deals primarily with tubing connections of 1.00-in. size and smaller. Welded, brazed, swaged, as well as the proprietary and military standard connections are discussed. The results of tests conducted with these connections are included.

The dynamic seal section discusses sliding and rotating shaft and piston seals for running speeds of 1000-in. per minute or less. Test results are included for the seal designs tested.

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I. SUMMARY

This report presents the results of surveys, analysis, and evaluation tests conducted with static and dynamic seals as well as joints over a span of approximately seven years. The work was performed in conjunction with various liquid rocket engine contracts for the Air Force and NASA as well as company-funded projects. This report was written under a NASA contract (NAS 3-2555) for the M-1 Engine. Also, much of the evaluation testing included was accomplished under this NASA M-1 contract.

The report is subdivided into three basic sections; static seals, small tubing joints, and dynamic seals. Each section contains information pertaining to good joint design, data to assist in selecting the proper seal and joint configurations, some pertinent and often forgotten "do's" and "don'ts" in joint design, and the results of evaluation tests with many types of gaskets, seals, and joints.

A. STATIC SEALS

The basic requirement for a static seal (gasket) is plastic (or elastic) flow of the gasket material to fill the asperities of the mating surfaces to the extent necessary to satisfy the allowable leakage of the system fluids at all operating and environmental conditions of the joint. Gasket selection is based upon the leakage requirement, fluid compatibility, fluid permeability, possible electrolytic corrosion problems, as well as other factors. Then, the flanges and fasteners are designed and constructed to make the best use of the gasket. The selection factors for the gasket, the variables encountered in the joint design, gasket capabilities, and loading requirements are detailed in this report. The requirements for proper joint assembly procedures as well as the necessary handling and packaging to prevent damage to the gasket and mating flange surfaces are also discussed.

The results of evaluation tests of many different types of gaskets and joints indicate that some proprietary designs have lower leakage rate capabilities and better flange deflection capabilities than others for specific applications. However, the best gasket selection for aerospace service is one that will provide less than the allowable maximum leakage rate of the system fluid with the lightest weight joint design. The lowest leakage rates for a given application can be obtained with a flat gasket design, but the weight penalty is nearly always excessive, particularly with the larger joint designs. Thus, the need for the "light-loading" proprietary gasket designs is well established. However, the problem of damaged or dirty mating flange surfaces is greatly aggravated by the use of these gaskets because of the narrow width of the sealing interface.

The relative helium leak rate capabilities determined from the tests at different operating and environmental conditions are presented for the joint designs using the proprietary gaskets.

B. SMALL TUBING JOINTS

The small tubing joints are treated as a separate entity (although they are static sealed joints) because they are usually threaded coupling joints and these present their own unique problems. The small tubing joints are more easily welded, brazed, or swaged where "zero" leakage is required and they should be used if at all possible. These welded and brazed joints are lighter in weight and can provide the lowest leakage rates of any of the small tubing joints.

Where threaded connections are necessary or desirable, the basic requirements for sealing are the same as for the static sealed joints. The proprietary threaded connections offer one important advantage over the flared and flareless tube connections in that they contain only one sealed joint (rather than two) for each connection. The disadvantages of the proprietary connections are that each half of the connection (each flange) must be welded, brazed, or swaged to the tube or component and usually, the male flange (with exposed male threads) must be mounted on the component. A more detailed discussion of the advantages and problems of the various types of small tubing connections is presented in the text.

The relative helium leakage rate capabilities of the proprietary and standard ("AN" and "MS") connections tested are presented in the text.

C. DYNAMIC SEALS

While the different types of dynamic seals are discussed, this section deals primarily with rubbing "contact" seals (sliding and rotating) for surface speeds of less than 1000-in.-per-minute.

The basic requirement for a "zero" leakage dynamic seal is the same as the static seal (complete plastic flow or deformation of the seal to the mating surfaces), except that this condition must be maintained with relative motion between the seal and mating surface. However, this is not possible because the mating surface would have to be perfectly smooth ("zero" RMS surface finish) to prevent shearing of the sealing interface at either the start of motion or during motion. Therefore, the dynamic joint design is the result of compromises between permissible leakage rates, joint life, and many other factors that affect these two variables. "Zero" leakage can be attained using sliding shaft seal designs with bellows or diaphragms. The requirements, variables, and compromises necessary for a dynamic joint design are discussed in detail in the text.

The results of tests with two types of dynamic (sliding and shaft) seals are also presented.

II. INTRODUCTION

"Leaking joints" and "leaking seals" are the predominant problems frequently remaining with fluid systems after all development work with the aerospace hardware is completed and the components are ready for production. There appears to be two basic reasons for this condition. Sealed joints have never been required to do so much with so little weight (Boiler Code flanges eliminate many of the variables encountered in lightweight flanges because of their heavier construction). Secondly, there is insufficient joint design data available concerning the selected gasket, seal, or joint for the designer to do a thorough study of the joints during the initial design stage.

The intent and purpose of this report is to provide current joint design information for the various gasket and seal designs tested. Considering the number of variables involved in joint designs (there are approximately 1.0×10^{22} possible variables and combinations of variables), it would seem impossible to provide all the necessary design information. However, by standardizing certain variables, such as groove machining requirements (i.e., depth tolerances, diameter tolerances, surface finish and flatness), the variations in other areas (i.e., allowable flange deflections and distortions), and gasket loading can be determined. With this type of information available, the designer is able to make the best gasket selection for a particular application. He is also able to design the other parts of the joint to meet the requirements for the specific gasket or seal.

Although the task of determining gasket capabilities is made easier by standardizing some of the variables, it is still not simple. Each supplier has seals with many different variations (part numbers), different inherent characteristics, various materials and platings, as well as different dimensional configurations. Also, there are numerous suppliers for seals, gaskets, and joints. All configurations from each of the suppliers cannot be evaluated by any single user because the time and expense involved would be prohibitive. All data and information presented in this report are for seals, gaskets, or joints with specific part numbers from a particular supplier and are noted as such in the data. Also, most testing of the seals and gaskets was conducted in grooves or glands to a standardized configuration developed with the assistance of many of the suppliers and which were designed for the particular type of seal or gasket. The type of data presented will be useful as a guide for requesting or determining the necessary design information required for other configurations of seals and gaskets. The seal or gasket manufacturer should be capable of determining (by test and analysis) individual gasket capabilities and providing the designer with the necessary requirements for designing other parts of the joint.

There is no specific method for comparing the capabilities of the various gasket and seal designs because there are numerous considerations as well as conditions for the many joint applications and designs. A gasket design may work well in one application and be completely unsuitable for another application and design condition. However, general comparisons can be made to help the designer make the best selection(s) for a particular application and design condition.

There are two basic types of seals; static (gaskets) and dynamic (packings). Each of these types are further subdivided into different classifications depending upon their function or method for providing a seal (i.e., face seals, diametral seals, pressure energized seals). For the purposes of this report, the joints have been divided into three groupings; static seal joints, small tubing joints, and dynamic seal joints. Although the small tubing joints are basically static seal joints, they are separated because they are generally a threaded connection and present their own unique problems. These joints are usually for 1.0-in. or smaller tubing connections.

Many years of development and fabrication experience have gone into the small tubing connection designs and some types have been standardized. It would be normal for a designer to use the standard designs in any new applications if previous applications have been successful. However, for the exotic system fluids and gases being used in aerospace designs, the standard connections are not adequate. In most aircraft systems, leakage of a few drops of oil or hydraulic fluid was not considered a serious problem, but in the current rocket engine systems, leakage of oxygen, hydrogen, nitrogen tetroxide, or similar propellants poses very serious problems. Such leakage can cause the whole system to malfunction as a result of external fires or external corrosion of the mechanical and electrical components. It can also be a serious personnel hazard. The tests conducted with these connections were designed to determine the suitability of the connector for use with various aerospace system fluids.

A similar analogy can be made concerning static and dynamic seal joints. The ASME boiler code has been used for years as the standard for designing pressure vessels and piping systems that include flanged joints. Where weight is not a special consideration, this method of design is still satisfactory. The code provides safety factors of 4 to 10 and at these stress levels, deflection and distortion of the members are so small that they become negligible. With the flight-weight hardware and the temperature extremes of the aerospace systems, where the materials are subjected to 60% or more of the yield strength, much greater deflections and distortions of the members are experienced. This requires a greater detailed analysis of joint members and gasket capabilities. Usually, the limiting factor in the joint members is deflection (strain) or distortion (permanent set) rather than stress. The data compiled in this report present the gasket and seal capabilities with respect to sealing surface deformation, flange deflection capabilities (a "measure" of how light the flanges may be), groove machining requirements, as well as other environmental and fluid compatibility characteristics.

A. STATIC SEALS

There are many types of static seals, but two basic types are predominant; face gaskets (squeezed between two flange faces) and diametral gaskets (squeezed between flange diameters). These can be further subdivided into pressure energized types, crush gaskets, internal and external pressure types, combination face, and diametral types to mention but a few. In this

report, the static seals have been grouped according to their more common usage; flat gaskets, pressure energized face seals (internal and external pressure), diametral seals, and special groove types. Many other groupings were considered but the ones presented herein are considered best for the designer's requirements.

All of the different types of gaskets that are discussed were not tested. Only those gaskets selected by engineering evaluation for application in various rocket engine systems were tested. However, since the completion of these tests, some manufacturers have modified or revised their gasket configurations and it is not known what improvements, if any, have resulted. New configurations should be thoroughly tested and analyzed with respect to the desired application before attempting to use them in actual hardware.

B. SMALL TUBING JOINTS

There are three basic types of small tubing connectors; threaded, brazed or welded, and swaged. A separate and complete study could be made of only the various types of threaded connectors. However, the flared and flareless tube threaded connections are the ones most widely used and they are also the most versatile. In addition, the standardization of the various components has done much to encourage the use of these connections. Tooling is available for the female bosses (usually used in component hardware) which makes these connections more desirable from the fabrication and production aspect. The standardized connectors are satisfactory for use in many systems, depending upon such considerations as the system fluids, temperatures, pressures, and vibration and shock loads. However, to be satisfactory for use in most aerospace systems, many improvements and changes are required. A detailed discussion of these desirable changes has been included. Each of the standardized connectors contains two joints; the boss-to-fitting and the tube-to-fitting joints. Some type of seal is required for the boss-to-fitting joint and the seals suitable for this joint are discussed. Sealing in the fitting-to-tubing joint can be obtained in two different ways:

1. The tubing is flared and makes intimate contact with a mating cone on the fitting when the connector is tightened (the flared tube connection).

2. A hardened ferrule is provided that "digs into" the outside diameter of the tubing when the connector is tightened (the flareless tubing joint).

A discussion of the relative merits of these connections is included.

There are other types of threaded small tubing connections available which are similar to the standard connections. These differ in some respects and it is purported that they are an improvement over the standard connection or that they have been designed for a specific application. Some

of these connections (or parts of them in instances where they are actually interchangeable with the standards) were tested and some were evaluated from an engineering aspect only.

The types of threaded connections that are purchased as a unit and welded to the tubing (or component) are also discussed. These units may provide somewhat better sealing capabilities than the standard connections but they present a more difficult production problem. Also, most of these units require that the male threaded flange be provided on a component. The more-easily damaged thread and sealing surface could cause the loss of a component as a result of improper handling. However, in some applications, the use of these connectors may be more desirable than the use of standard connections.

C. DYNAMIC SEALS

There are two basic types of dynamic seals; the face seals and the diametral seals. Face seals are most commonly used in high-speed, rotating shaft applications where a highly-polished shoulder is provided on the shaft and the seal is mounted in a body or housing to make sealing contact with the shoulder. The high-speed, rotary face seals are not included in this report because they are usually designed for a specific application. The diametral seals are subdivided into rotary seals (low-speed rotating shaft) and sliding seals. These two types are normally used in controls hardware (i.e., valves) and are limited to surface speeds of approximately 1000-in.-per-minute, or less. These diametral seals were evaluated from an engineering aspect and some were tested in particular applications. Although the data and information presented is limited, it provides a good basis for further study and evaluation.

III. STATIC SEALS

A. TECHNICAL DISCUSSION

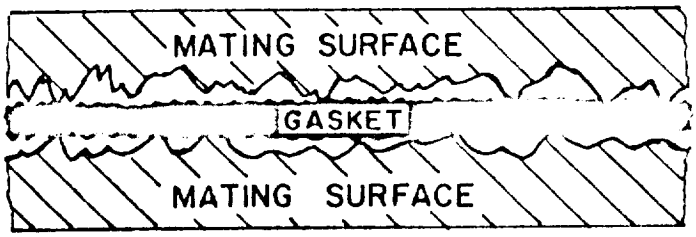
1. Basic Requirements

The perfect static seal is obtained when complete plastic flow or deformation occurs between the gasket and the mating-flange surfaces. The best sealing is obtained with an intergranular or intermolecular interface. A properly laid weld does this and it is recommended whenever possible. However, the necessity for easy maintenance of the system components eliminates this possibility in many cases; therefore, the static sealed joint must be used. To attain the initial seal, the replaceable gasket must provide plastic (or elastic) deformation or flow of the gasket material to fill all the asperities of the mating surfaces. This is most important because the extent of plastic (or elastic) deformation of the gasket sealing surfaces determines the degree of sealing obtained as shown in Figure No. 1. In general, plastic deformation applies to the metal gasket materials and elastic deformation applies to the elastomeric (rubber-like) gasket materials.

a. Leakage Criteria

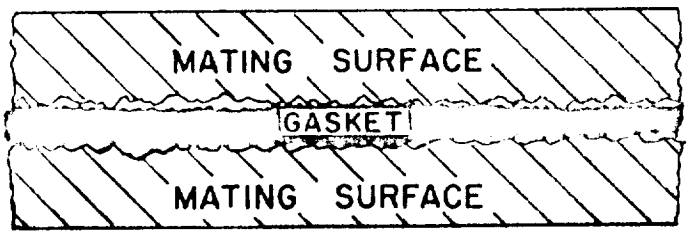
The allowable leakage rate is a basic consideration in the joint design. This one factor presents more problems to the designer than any other. To a great degree, the leakage criteria are determined by the fluids to be used. However, there is a great difference in designing a hydraulic fluid system joint for a leakage rate of 1 cc/min maximum as compared to 1 cc/year maximum. The difference is even more pronounced for pneumatic system joint designs.

The allowable leakage rate for joints in a particular system (commonly called the "zero leakage" value) must be a function of the system fluid (gaseous or liquid), the size and number of joints in the system, the use, and the application or mission required of the system. Other considerations include personnel hazards, explosion hazards, and external corrosion or erosion of the system hardware. Permissible leakage ("zero leakage" values) from joints in ground handling or support equipment could be relatively high with respect to deep space systems if the leakage does not present personnel hazards, explosion hazards, or other undesirable conditions. Ground equipment is easily refilled and tankage volumes can be adjusted during initial design to assure that the system performs its function before the fluids are depleted. Conversely, permissible leakage ("zero leakage" values) requirements for deep space system joints may be extremely low to prevent the depletion of system fluids before the mission is completed (possibly months long with a number of operating cycles), eliminate personnel or explosion hazards, and prevent external corrosion, erosion, or burning of the system hardware. Tankage volumes are very critical because of weight and size considerations as well as the impossibility of refilling the system during the mission.



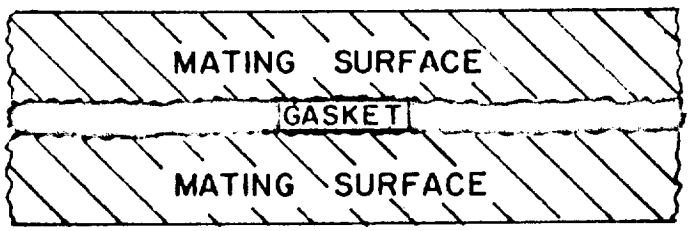
NO CONFORMATION
(LITTLE OR NO DEFORMATION)

UNSATISFACTORY FOR ALL FLUIDS AND SURFACE FINISHES



PARTIAL CONFORMATION
(SOME DEFORMATION)

MAY BE SATISFACTORY FOR SOME LIQUIDS DEPENDING ON SURFACE FINISH AND LAY



TOTAL CONFORMATION
(TOTAL DEFORMATION)

SATISFACTORY FOR ALL FLUIDS REGARDLESS OF SURFACE FINISH BUT NO DEEP SCRATCHES PERMISSIBLE ACROSS SURFACE

Figure 1

Sealing Surface Deformation

The type of leak detection equipment required to demonstrate that the joint is adequate also depends upon the leakage requirements. A good example of various leakage requirements for a gaseous helium system is shown on Table I.

TABLE I

LEAKAGE REQUIREMENTS FOR A GASEOUS HELIUM SYSTEM

Allowable Leakage	Approximate Equivalent	Leak-Detection Method	Approximate Cost of Equipment, $	Remarks
1.0×10^{-2} std. cc/sec	0.6 cc/minute (6 to 10 ave. size bubbles (in two minutes)	Water submersion, wet test meter, or flowrator.	50	Detectable and measurable with simple equipment.
1.0×10^{-3} std. cc/sec	1 cc in 17 minutes (No bubbles in two minutes)	Water submersion, or wet test meter	50 to 200	Easy to get false indications. Trapped gas between flanges, and outside of seal, may indicate leakage as pressures applied. May show no bubbles when leaking because helium can be absorbed into the water.
10×10^{-6} std. cc/sec	1.0 cc in 12 days	Mass spectrometer or equivalent	5000	Leak check requires approximately 1/2 hr to set up and 1/4 hr to test
1.0×10^{-7} std. cc/sec	1.0 cc in 4 months	Same as above	Same as above	Same as above

Because mass spectrometer devices are capable of determining leakage rates down to 1.0×10^{-9} std cc/sec (1.0 cc in 32 years), actual "zero leakage" as a requirement for a joint could become expensive as well as generally impractical. At a sealing requirement of 1.0×10^{-6} std cc/sec, leakage (or permeability) through castings and forgings becomes a considerable problem and is even more pronounced at 1.0×10^{-9} cc/sec.

Leakage can be caused by permeation or flow through the gasket materials, porosity in the joint materials, or by flow through the sealing interface. This permeation through elastomeric or plastic materials as well as through porous casting, forging, and raw stock metals can become a serious problem in attempting to achieve very low helium and hydrogen leakage rates. Permeation rates of gases or fluids through some of the plastics will

be discussed subsequently. Methods are available for sealing porosity in castings for most fluid systems; however, with some of the new rocket fluids and temperatures, these sealants are either dissolved or tend to run or flake-out at either very high or very low temperatures.

The most predominant leakage path in joints is across the sealing interface. Therefore, the primary problem is one of ensuring that intimate contact is made between the gasket and mating surfaces to achieve the initial seal. In addition, it is necessary to assure that this contact is maintained and not disturbed during the life of the joint as well as under all of the operating, storage, and handling conditions that can be encountered.

The various types and modes of leakage that can occur in the joint depend upon the type of fluid being sealed and the minimum cross-sectional area of the leak path. Experiments have been conducted and analyses were made of both the types and modes of leakage(1)(2). If sufficient plastic deformation at the sealing interface is attained to satisfy the requirements for a gaseous system seal, no leakage problems should be encountered using the joint in a liquid system. There are two basic gaseous leak modes; molecular flow and viscous flow, with a regime called transition flow existing between the two. It is most important that these be thoroughly understood when conducting investigations and evaluations of sealed joints. A detailed study of the referenced literature is essential. Briefly, molecular flow exists when the minimum cross-sectional area of the leakage path is small in comparison to the molecular mean free path of leakage gas flow. As the minimum cross-sectional area of the leakage path becomes large in comparison with the molecular mean free path of the gas, viscous flow commences. The regime existing between pure molecular and pure viscous flow is transition flow. In the molecular flow regime, the leakage rate is proportional to pressure differential across the leak path. In the viscous (laminar) flow regime, the leakage rate for gases is proportional to the difference between the square of the internal pressure and the square of the external pressure, and the leakage rate for liquids is proportional to the difference between internal and external pressure.(3)

Leakage is also directly proportional to the total length of the sealing interface (π x mean sealing interface diameter) and inversely proportional to the width of the sealing interface (the distance that fluid must travel across the interface to escape) in both leak modes. This means that for the same conditions (i.e., gasket stress, mating surface finish, and system pressures), the leakage from large diameter joints will be greater than

(1) Rathbun, Jr., F. O. (Editor), Separable Connector Design Handbook, Advanced Technological Laboratories, General Electric, December 1964

(2) Marr, J. W., Leakage Measurement and Evaluation, Proceedings of the Conference on Design of Leak-Tight Separable Fluid Connectors, NASA, MSFC Huntsville, Alabama, March 24 and 25, 1964

(3) Bauer, Paul; Glickman, Myron; Iwatsaki, Frank, Analytical Techniques for the Design of Seals for Use in Rocket Propulsion Systems, I.I.T. Research Institute, AFRPL-TR-65-61, May 1965

from small diameter joints. For this reason, it is more desirable, from the design aspect, to specify leakage limits ("zero leakage" values) for the joints in a system based upon the length or diameter of the sealing interface (i.e., cc/sec/in. circumference or cc/sec/in. diameter).

The data presented in this report were collected from development and evaluation tests of seals and joints performed by the Aerojet-General Corporation for a number of different propulsion system applications. Permissible leakage rates ("zero leakage" values) were different for each of the systems; therefore, where "zero leakage" values are specified, they are those required for the particular system application.

All leakage rate values are given in volume per unit of time per unit length of sealing interface (cc/sec/in. circumference) where the leakage rate from the joint is converted to standard conditions (14.7 psi and 68°F), then divided by π times the mean circumference of the sealing interface. The justification for using leakage-per-unit-length for the proprietary gaskets is based upon the following analogy.

If the gasket preloading (lb/in.) is essentially the same for a particular cross-sectional size and configuration regardless of diameters (see section III,C,1) and mating surface finish, and the other machining requirements are the same for all sizes, then such things as the sealing stresses and sealing edge width should be essentially the same. Leakage capabilities for the particular cross-sectional size should then be directly proportional to the length of the sealing interface (normal to leakage flow). Comparisons are then possible between different cross-sectional sizes and configurations from the same manufacturer. Also, comparisons can be made between gaskets of approximately the same cross-sectional size (i.e., approximately the same space in the joint and the same preloading) from different manufacturers.

b. The Initial Seal[4]

Plastic deformation of the metal gasket sealing surfaces (the seal mating surface interface) implies that the gasket material must be stressed above the compression yield point (in the plastic range). The compression stress required to promote plastic deformation depends upon the material of the gasket. The amount of plastic deformation required at the seal interface depends upon the fluid to be sealed, the allowable leakage rate, the surface roughness, and the lay (direction of machining) of the mating surface. To some extent, it also depends upon the type of machining used (i.e., cutting tool, grinding, and burnishing). The lay of the mating surface finish is the most important factor determining the amount of plastic deformation (compression stress) required. If the machining marks run across the gasket sealing surfaces, the required compression stresses will be considerably higher than if the machining marks are parallel to the sealing surfaces. Also, the roughness of the mating flange surfaces plays an important part in the required compression stress. In general, a rougher flange surface will require higher compression stresses than will the smoother surface.

(4) Smoley, E. M., et al, Nonmetallic Gaskets, Product Engineering Seals Book, Armstrong Cork Co., June 11, 1964

The type of fluid to be sealed and the maximum allowable leakage rate determines how well the asperities in the flange surface finish must be filled and therefore, the amount of plastic deformation (compressive stress) required. The asperities in the groove surface finish for a liquid system allowing a maximum of a few drops of leakage per hour would not need to be filled as well as for a gaseous helium system requiring leakage rates of less than 1.0×10^{-6} cc/sec.

The deformation in the plastic (i.e., Kel-F and Teflon) gasket materials is neither plastic nor elastic. When sufficient compressive stresses are applied to the plastics, they tend to slowly flow in all directions and unless completely constrained, will continue to flow until the compressive stresses are relieved. The plastic materials are much less dependent upon the mating surface finish to provide a seal than are the metal gaskets. However, unless properly enclosed or constrained, the continuing flow under compressive stress (cold flow) will cause the loss of the initial seal. Cold flow characteristics are a function of compressive stress, time, and temperature as shown on Figure No. 2. Also, depending upon the allowable leakage rates, the permeability of various fluids and gases through a plastic gasket material may be too great to satisfy the leakage requirements as shown on Figures No. 3, No. 4, and No. 5.

The elastomeric materials deform to the mating surfaces by large elastic deformations. The elastomers are also much less dependent upon the mating surface finish to provide a good seal. However, these materials may not be capable of providing a satisfactory initial seal because of their permeability when used with the specific liquid or gas being contained.

Experiments[5] indicate that the compressive (sealing) stresses required for metallic gasket materials to obtain the initial seal vary from approximately 1.5 to 3.0 times the gasket material yield strength and are dependent upon groove surface finish and lay, the fluid to be sealed, the allowable leakage rates, and the pressure differential across the seal. The compressive (sealing) stresses required to obtain the initial seal with plastic (i.e., Kel-F, Teflon, and Nylon) gasket materials vary from approximately 0.5 to 1.5 times the tensile strength (usually equal to the compressive yield strength) of the gasket material. The compressive (sealing) stresses required to obtain the initial seal with the elastomeric gasket materials is the lowest of all the sealing materials. For relatively thick sections, the load-compression characteristics are approximately linear up to approximately 30% of the original thickness (30% squeeze). For the thinner sections and for the compression values above 30%, the load increases rapidly with increased squeeze.

c. Maintaining the Initial Seal[6]

After all the conditions have been satisfied to provide the necessary deformation to fill the asperities in the mating surface and attain the initial seal, the problem becomes one of maintaining this condition

(5) Rathbun, Jr., F. O., op. cit.
(6) Smoley, E. M., op. cit.

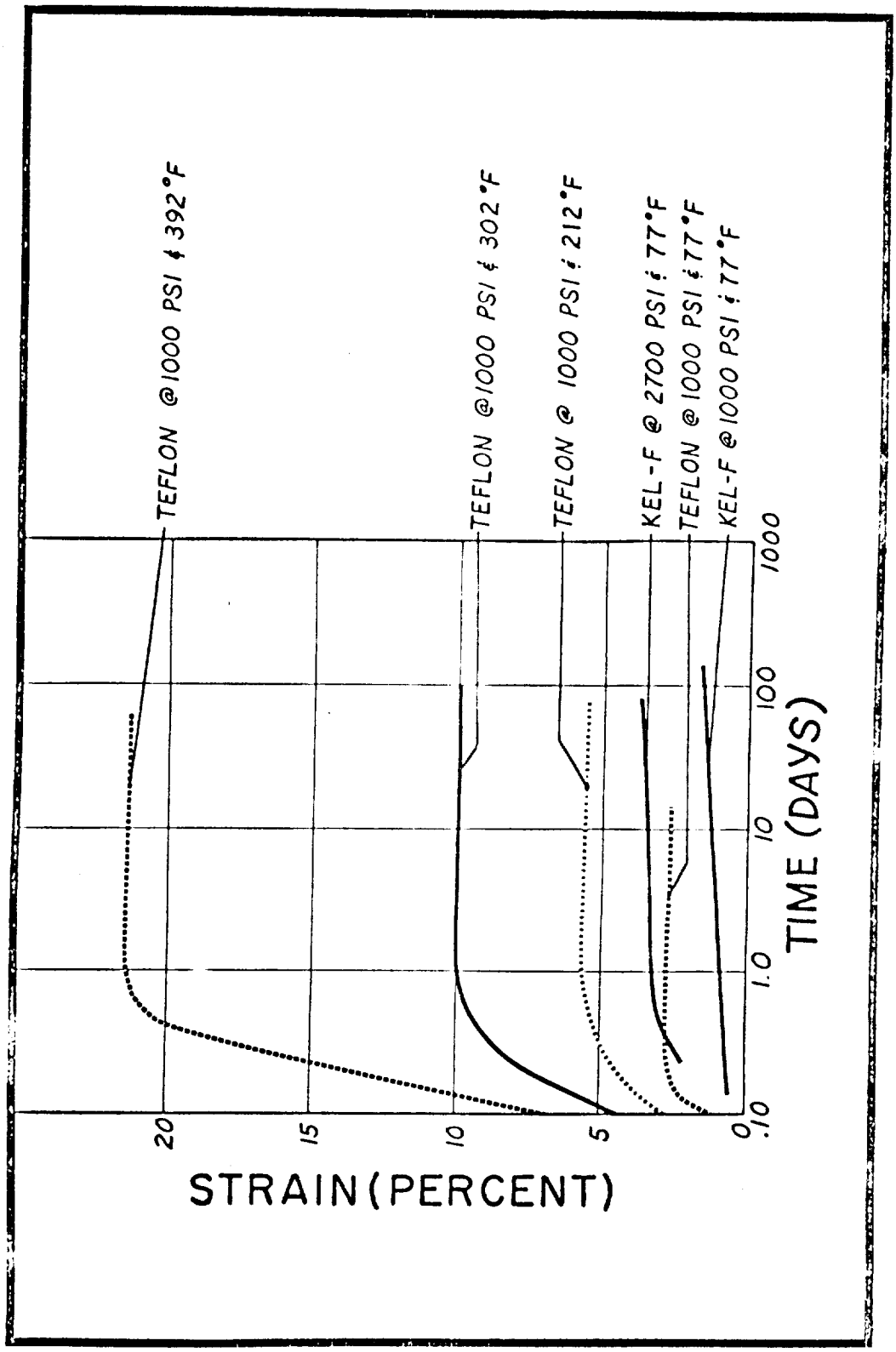


Figure 2

Cold Flow Characteristics of Some Plastic Materials

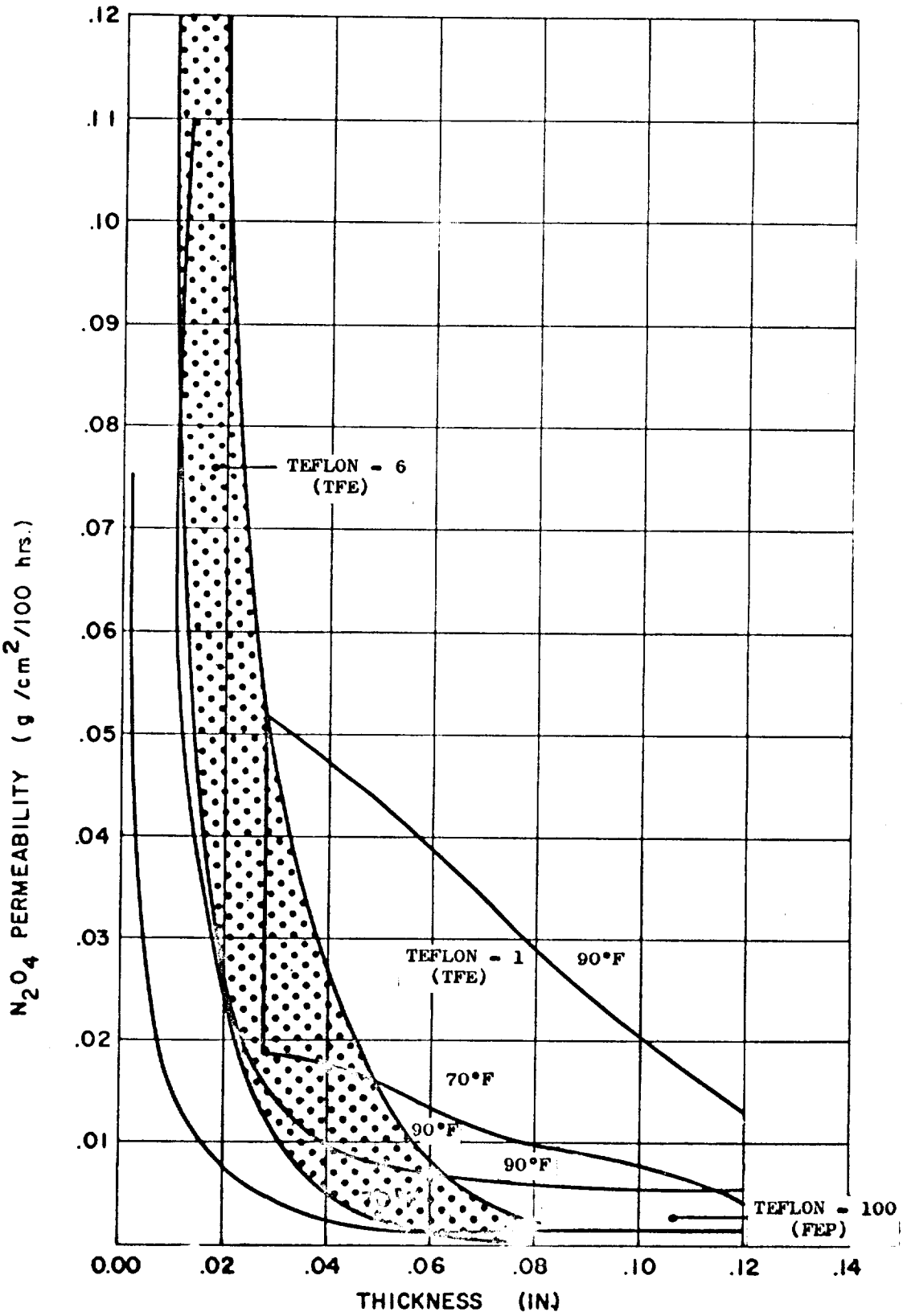


Figure 3
Permeability of Teflon to Nitrogen Tetroxide

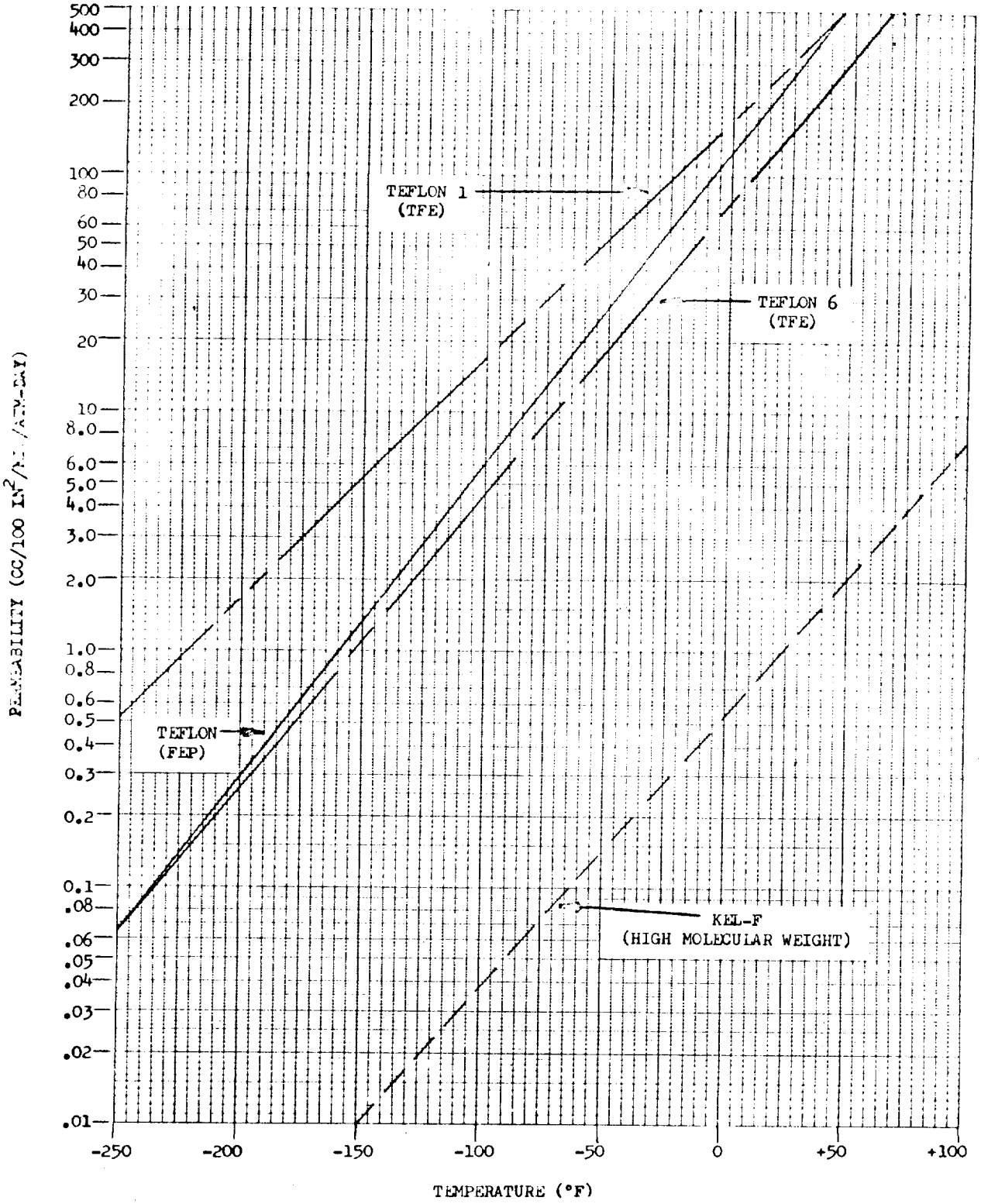


Figure 4

Oxygen Permeability, Perhalogenated Plastics

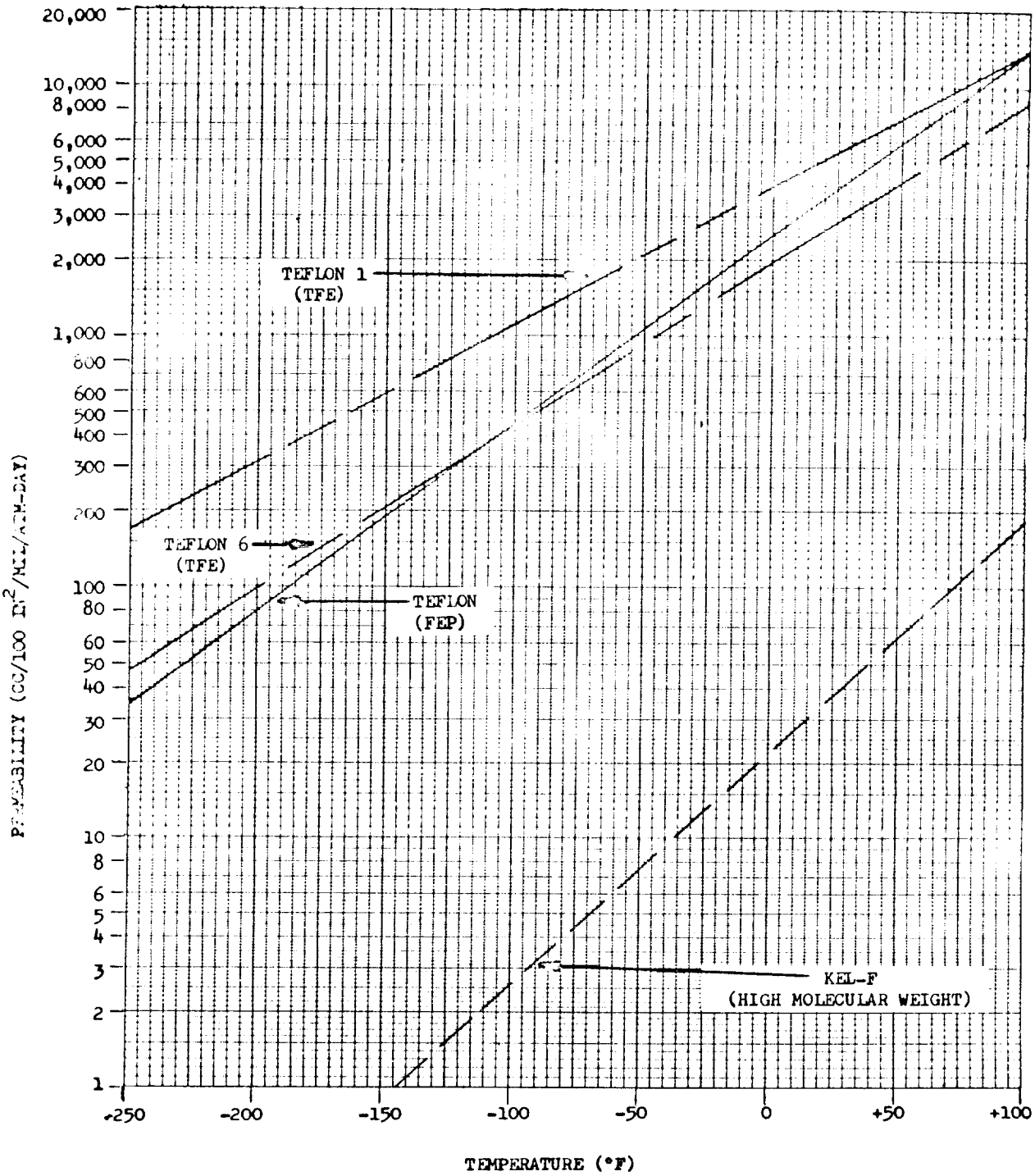


Figure 5

Hydrogen Permeability, Perhalogenated Plastics

at all operating and storage environments during the life of the joint. If no changes occur at the sealing interface under all of the environmental and operating conditions, the seal will be maintained. However, deflections and distortions do occur, particularly in the lightweight joints, as shown on Figure No. 6. Axial movement (straight separation or bending rotation) of the mating surfaces will cause a reduction in the compressive (sealing) stresses all around the sealing interface. Changes in flatness (bowing between flange supports) of the mating surfaces will cause a reduction in sealing stresses at local areas along the sealing interface. Stress relaxation can occur because all highly stressed parts tend to relax and equalize the stresses. This will cause a loss in fastener tension as well as an excessive reduction in sealing stresses and is a particular problem in elevated temperature service. Soft metal gasket materials will creep and relieve the initial sealing stresses as shown on Figure No. 7. Plastic materials, unless completely constrained, will cold flow and relieve the initial sealing stresses as previously shown on Figure No. 2. Elastomeric materials can take a permanent set (commonly called, compression set) as shown on Figure No. 8.

After sufficient sealing stresses are attained to provide the initial seal, a reduction in this stress may be permissible without detriment to the sealing performance. The permissible reduction in sealing stresses is dependent upon many factors, including the gasket material, the initial stresses or deformation, and the configuration of the joint. However, the minimum permissible sealing stresses should generally never become less than approximately 70% of the initial sealing stress.

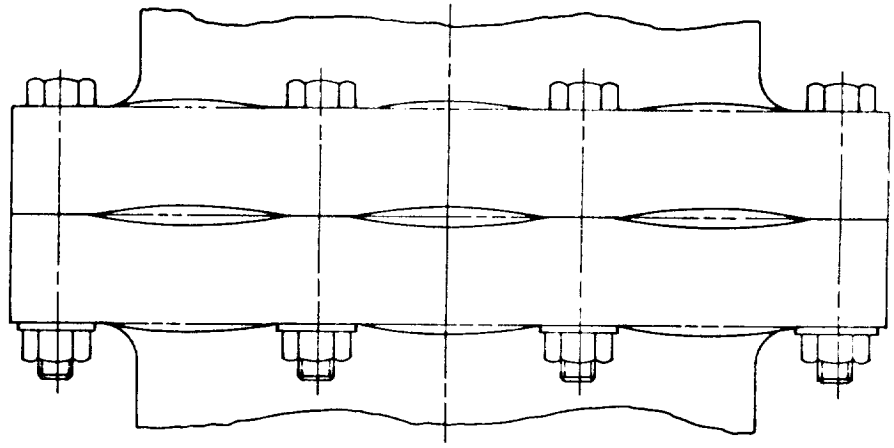
Another condition that must be considered in maintaining the initial seal is the shifting of the mating sealing surfaces. A lateral shift of the sealing surfaces can cause the sealing interface to slide. This will shear the intimate contact between the gasket material and mating surface, thereby destroying the effectiveness of the seal. The condition can be caused by unequal temperatures in the mating surfaces or by differences in the thermal expansion-contraction coefficients or moduli of elasticity of the mating surface materials and the gasket.

Various other phenomena can be encountered during the operation, storage, cleaning, and handling of the joint that could disturb the intimate sealing interface contact. The following are brief summaries of the conditions that could occur:

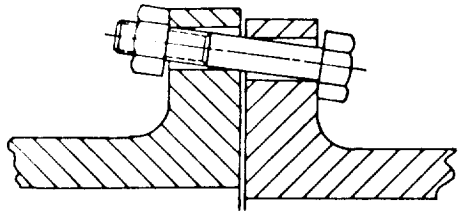
(1) Elastomeric and Plastic Materials

The minimum seating stresses can be maintained but pressure can cause the seal to fail (blowout) in hoop tension.

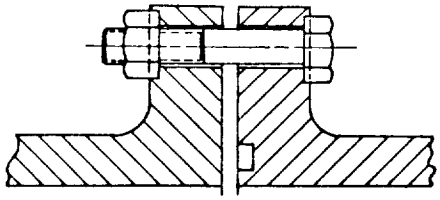
The minimum seating stresses can also be maintained during pressure cycling with a separation of the flange faces occurring. Extrusion of the seal between the surfaces during pressurization and cutting (nibbling) of the material during depressurization can occur with eventual loss of the seal.



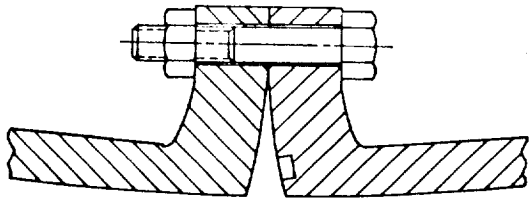
BOWING
(CIRCULAR DEFLECTION OR DESTORTION)



DIFFERENTIAL CONTRACTION
& EXPANSION DUE TO PRESSURE
OR TEMPERATURE
(RADIAL DEFLECTIONS OR
DISTORTIONS)



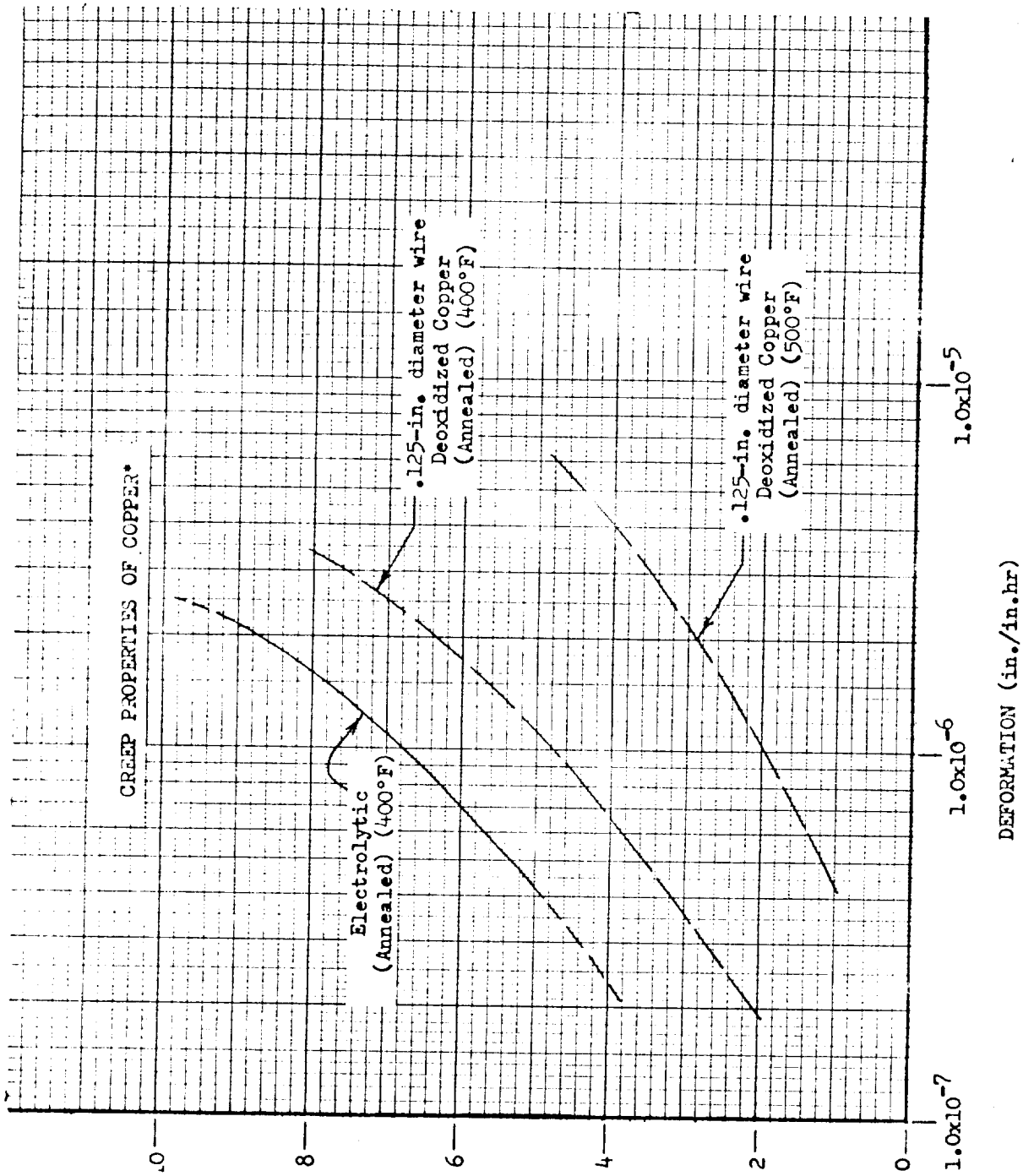
BOLT STRAIN & BOLT HOLE
DISTORTION
(FLANGE SEPARATION)



FLANGE ROTATION
(ANGULAR & AXIAL DEFLECTIONS OR DISTORTIONS)

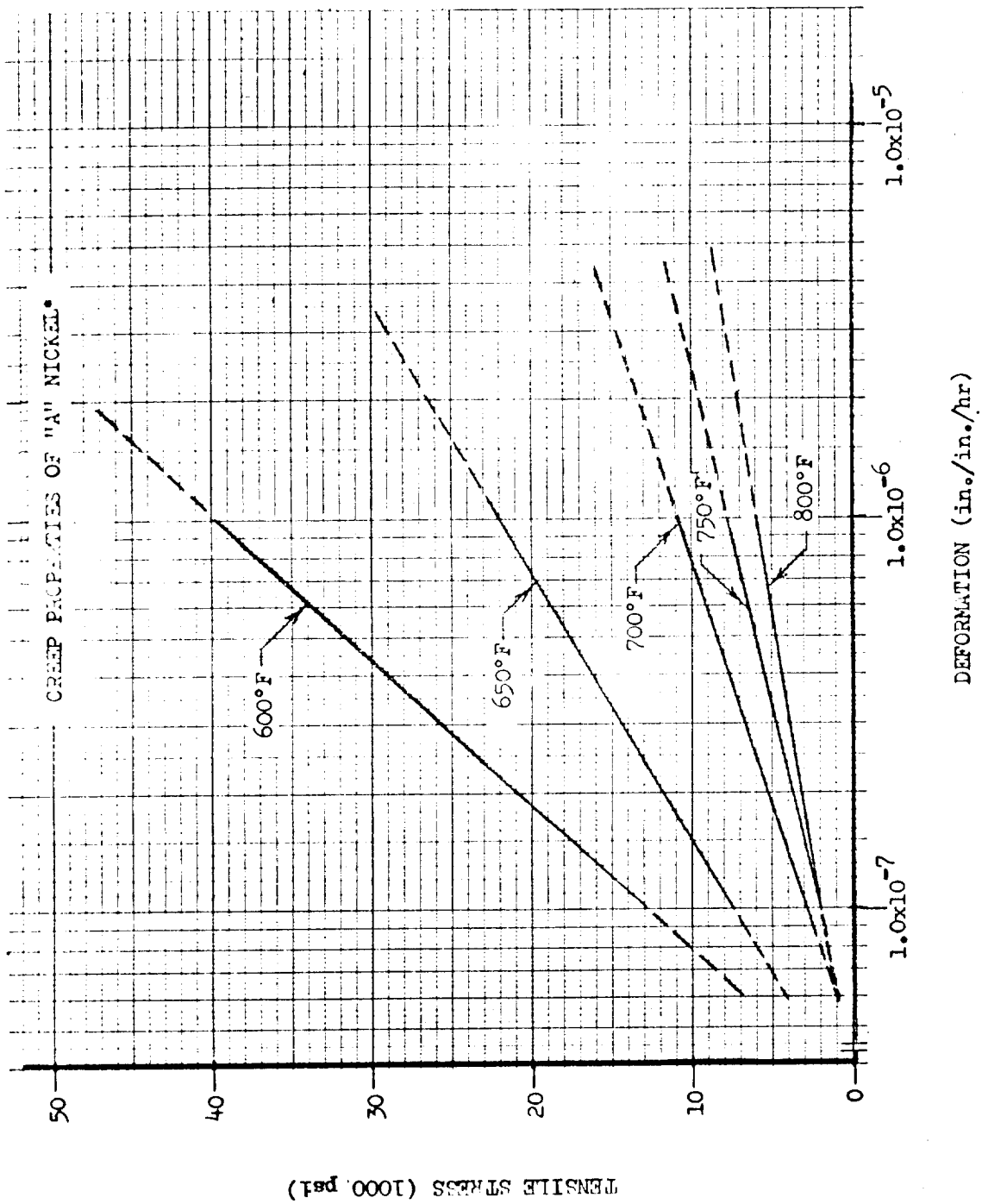
Figure 6

Types of Deflections and Distortions in Joints



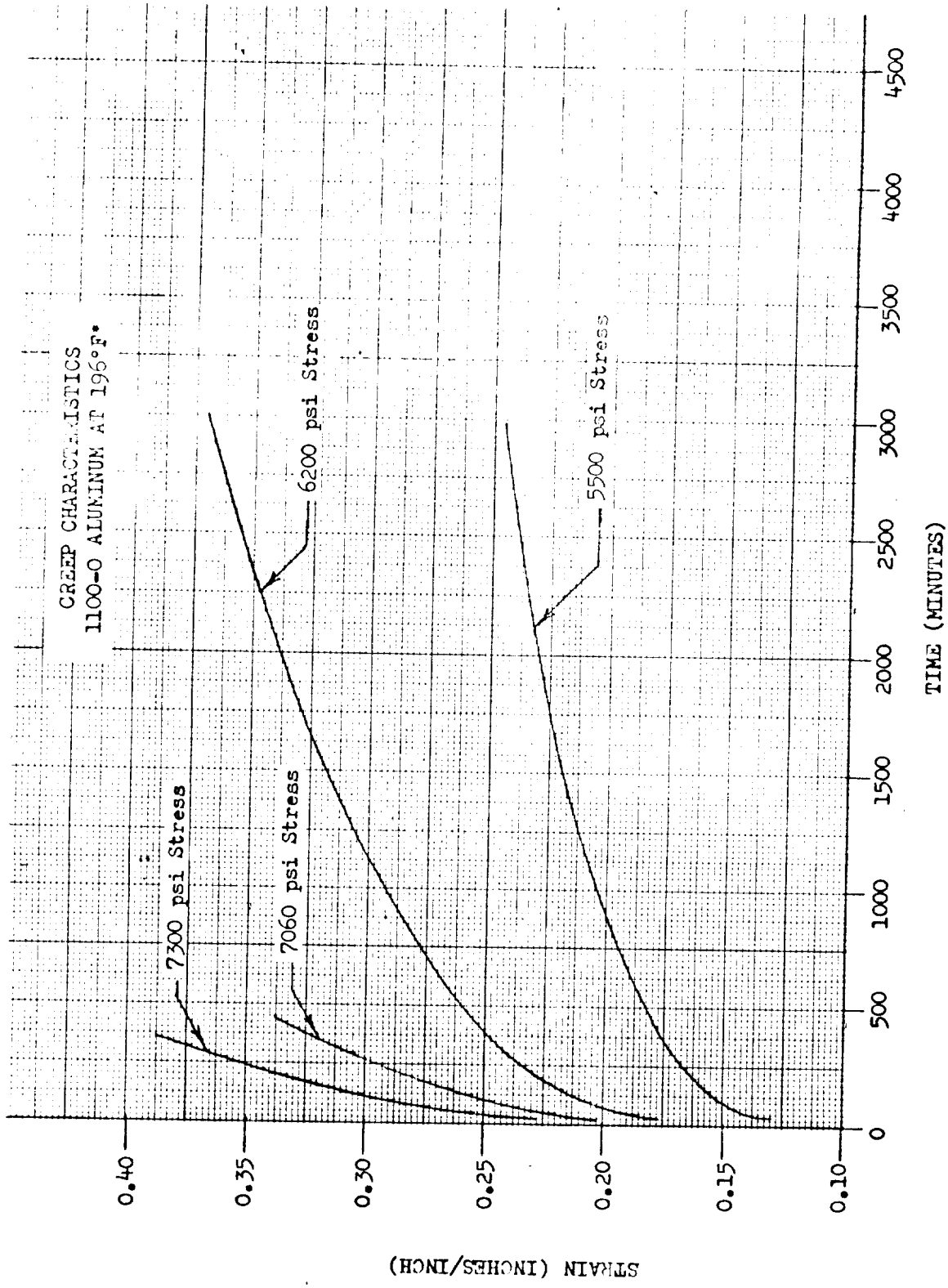
* Metals Handbook, 1948 Edition, pages 906 and 908

Figure 7
(Sheet 1 of 3)
Creep Characteristics of Copper, "A" Nickel and Aluminum



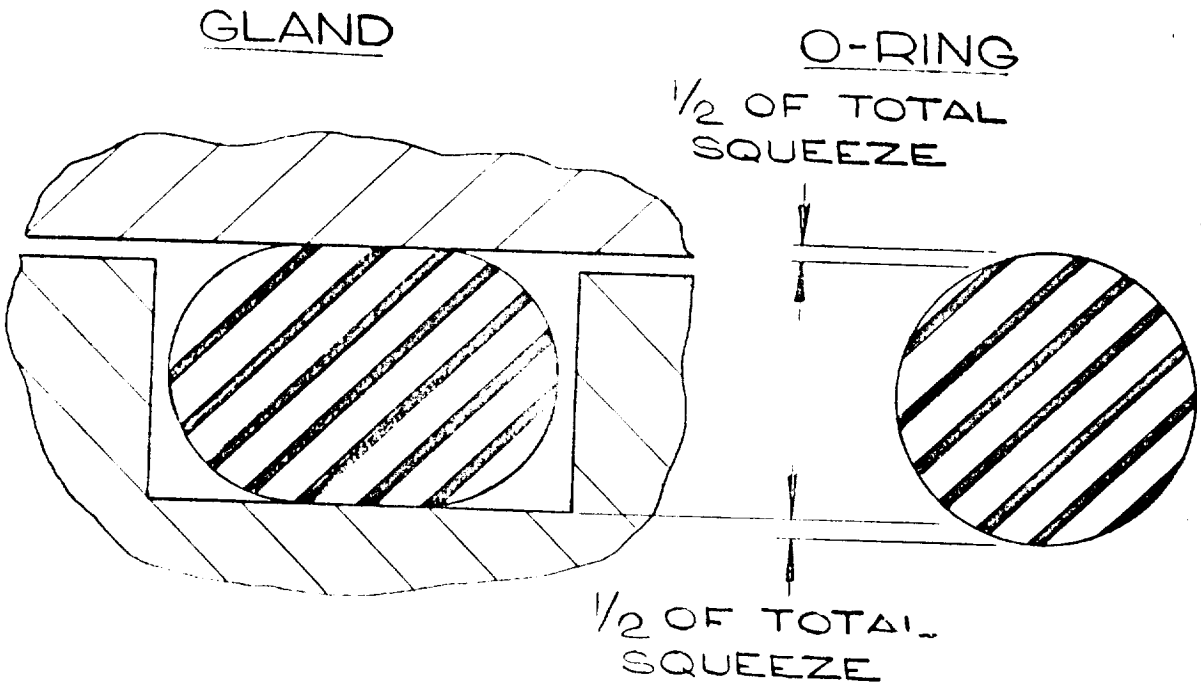
* Engineering Properties of Nickel, International Nickel Company, Technical Bulletin T-15

Figure 7
(Sheet 2 of 3)
Creep Characteristics of Copper, "A" Nickel and Aluminum

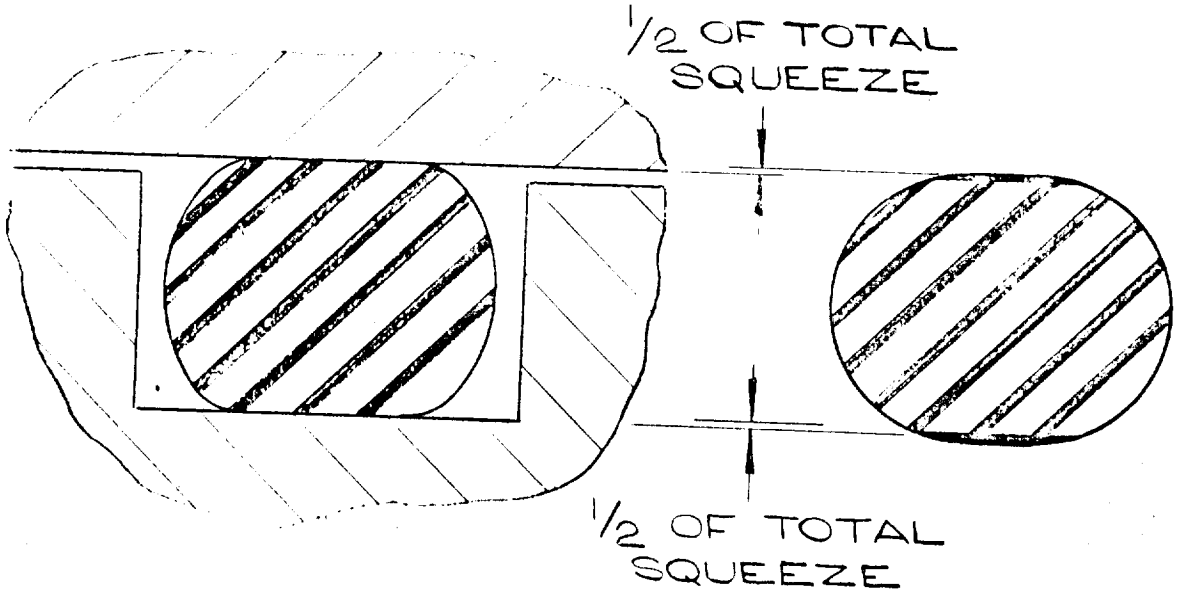


* Jaffe, N. and Dorn, J. E., Effects of Stress on Creep Rate of High Purity Aluminum in the Cross-Slip Region, WADD Technical Report 61-200, September 1961

Figure 7
(Sheet 3 of 3)
Creep Characteristics of Copper, "A" Nickel and Aluminum



AS INSTALLED



AFTER STORAGE

Figure 8

Compression Set of Elastomeric O-Rings

At lower temperature operation, spalling (a flaking at the gasket corners) can occur during vibration conditions. This condition progresses rapidly as the interface area decreases until failure finally occurs.

The fluid being sealed and/or used to clean the system can cause the gasket material to swell, contract, dissolve, erode, or change hardness, strength, or other physical properties which will destroy or affect the intimate contact at the sealing interface. A typical example of a reduction in material strengths is shown on Figure No. 9 for plastics immersed in nitrogen tetroxide. The compatibility of gasket materials with most of the liquid rocket propellants can be found in the published literature.[7]

Some elastomers will decompose when subjected to hard vacuum environments for extended periods. Each elastomer must be thoroughly evaluated when considered for this type of service.

(2) Metallic Materials

Serious galvanic (electrolytic) corrosion or deterioration of the sealing interface as well as other parts of the joint can occur where dissimilar metals are in contact or share an electrolyte.[8] Any fluid that will conduct an electrical current can serve as an electrolyte (moisture in the air, rocket propellants, and certain gas concentrations in air). The electrical potential between the dissimilar materials transfers an ion from the cathode to the anode, thereby destroying the anodic materials. In instances such as these, it is important to eliminate couples between metals of widely divergent electromotive potential. Metals selected for the gaskets, flanges, and fasteners should be relatively close to each other in the electromotive series, or it may be necessary to select a gasket material or coating having good dielectric properties to reduce the galvanic action between the flange materials. The definition and use of dissimilar metals are presented on Table II.

Rust in iron or steel is another form of corrosion that can cause serious sealing problems. This rust, which is caused by the oxidation of the parent metal, can be eliminated through the proper surface treatment of the part. Active chemical solutions, such as many of the liquid rocket propellants, can attack certain metals; therefore, the flange, the gasket, and the exposed fastener material must be selected to resist chemical corrosion. The chemical compatibility of joint materials with the common rocket propellants is also to be found in the published literature.[9]

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(7) Rathbun, Jr., F. O., op. cit.
(8) Dunkle, Heber, H., Metallic Gaskets, Product Engineering Seals Book, Johns-Manville, June 11, 1964
(9) Rathbun, Jr., F. O., op. cit.

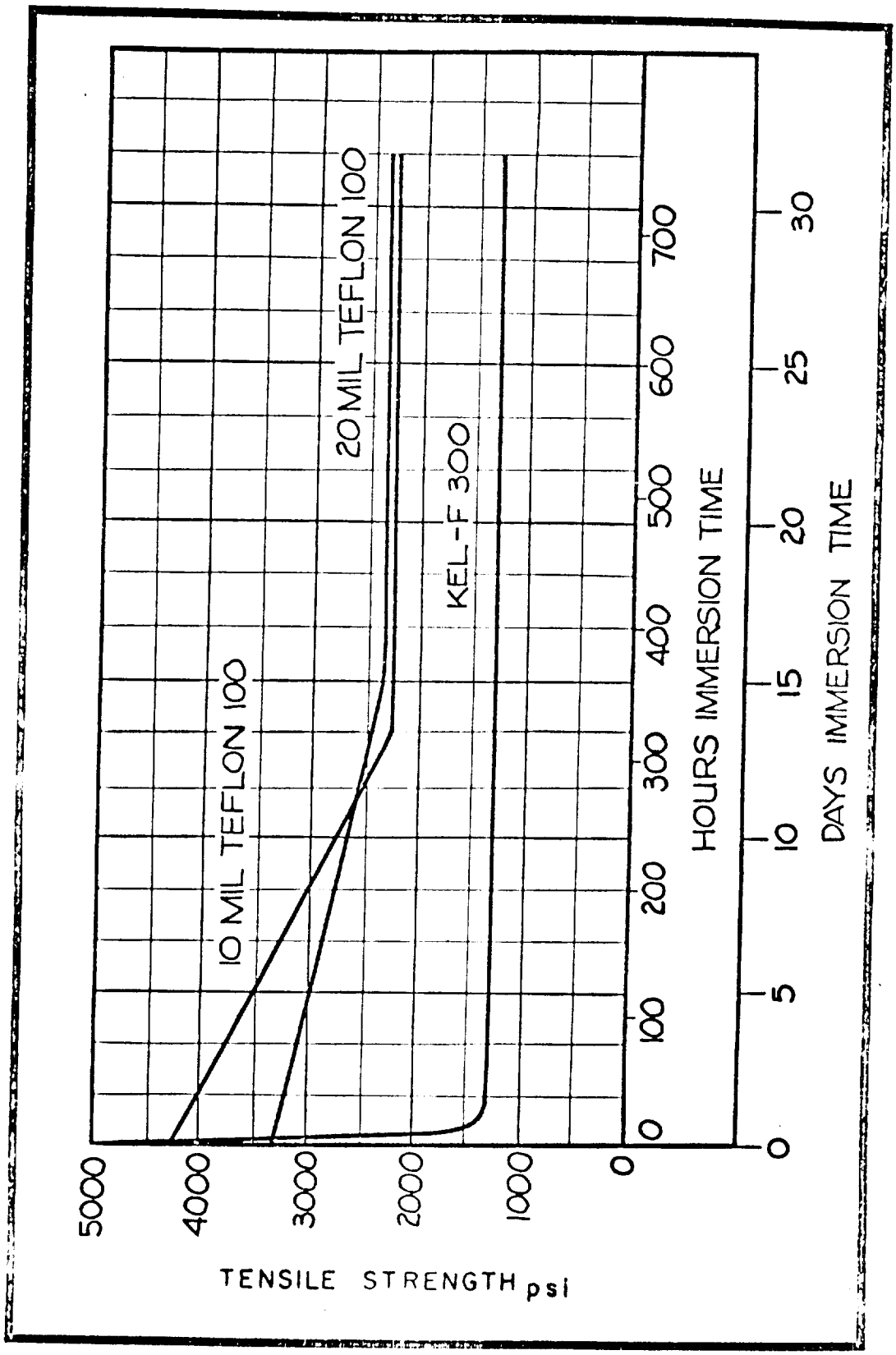

Figure 9
Effect of Nitrogen Tetroxide on the Tensile Strength of Some Halogenated Plastics

TABLE II

DEFINITION AND USE OF DISSIMILAR METALS

LEGEND OF SYMBOLS USED IN THE TABLE:

- X = DISSIMILAR METAL COUPLE, REQUIRES FINISH OR ADEQUATE INSULATION.
- C = COMPATIBLE METAL COUPLE, FINISH TO ENVIRONMENTAL REQUIREMENTS ONLY.
- ■ = FINISH REQUIRED DUE TO INHERENT CORROSIVENESS.
- P = COMPATIBLE COUPLE IF NOT EXPOSED TO MARINE ATMOSPHERE OR IF USED MORE THAN 2 MILES FROM SALT WATER. NOT COMPATIBLE IN HIGH HUMIDITY.

THE METALLIC COUPLE CONSISTS OF THE METALS WHICH ARE ULTIMATELY ON THE SURFACE OF THE FAYING METALS.

	METALS	1 GOLD PLATINUM	2 RHODIUM, GRAPHITE, PALLADIUM	3 SILVER	4 TITANIUM	5 NICKEL, MONEL, COBALT	6 NICKEL-COPPER ALLOYS QQ-N-281, QQ-N-286, MIL-N-20184	7 STEEL AISI 300 SERIES (1), NICKEL CHROME STEEL	8 COPPER, BRONZE, BRASS, COPPER ALLOYS PER QQ-C-551, QQ-B-671, MIL-C-20159, MIL-B-6946, AMS 4640, SILVER SOLDER PER QQ-S-561	9 COMMERCIAL YELLOW BRASS & BRONZE, BRASS COMP.	10 HIGH ALLOY, LEADED, NAVAL BRASS, LEADED BRONZE	11 CORROSION-RESISTANT STEELS 18 ±2% CR, AISI 400 SERIES (2)	12 CHROMIUM PLATE, TUNGSTEN, MOLYBDENUM	13 CORROSION-RESISTANT STEELS 12 ±2% CR, AISI 400 SERIES (3)	14 TIN, TIN PLATE, INDIUM	15 LEAD	16 ALUMINUM 2024, 2014, 7075	17 ALL STEELS EXCEPT CORROSION-RESISTANT TYPES (SEE 7, 11, 13)	18 ALUMINUM ALLOY 1100, 3003, 5052, 6063, 6061, 356	19 CADMIUM & ZINC PLATE, GALVANIZE, BERYLLIUM, CLAD ALUMINUM	20 MAGNESIUM
1	GOLD, PLATINUM	C	C	C	C	X	X	X	X	X	X	X	C	X	X	X			X	X	
2	RHODIUM, GRAPHITE, PALLADIUM	C	C	C	C	C	C	C	C	X	X	X	C	X	X	X			X	X	
3	SILVER	C	C	C	C	C	C	C	C	X	X	X	C	X	X	X			X	X	
4	TITANIUM	C	C	C	C	C	C	C	C	C	C	C	C	C	C	C			C	C	
5	NICKEL, MONEL, COBALT	X	C	C	C	C	C	C	C	C	C	C	C	P	P	X			P	X	
6	NICKEL COPPER ALLOYS QQ-N-281, QQ-N-286, MIL-N-20184	X	C	C	C	C	C	C	C	C	C	C	C	X	X	X			X	X	
7	STEEL AISI 300 SERIES (1), NICKEL CHROME STEEL	X	C	C	C	C	C	C	C	C	C	C	C	C	P	X			P	P	
8	COPPER, BRONZE, BRASS, COPPER ALLOYS PER QQ-C-551, QQ-B-671, MIL-C-20159, MIL-B-6946, AMS 4640, SILVER SOLDER PER QQ-S-561	X	C	C	C	C	C	C	C	C	C	C	C	C	X	X			X	X	
9	COMMERCIAL YELLOW BRASS & BRONZE, BRASS COMP. [illegible]	X	X	C	C	C	C	C	C	C	C	C	C	C	C	X			X	X	
10	HIGH ALLOY, LEADED, NAVAL BRASS, LEADED BRONZE	X	X	X	C	C	C	C	C	C	C	C	C	C	C	X			X	X	
11	CORROSION RESISTANT STEELS 18 ±2% CR, AISI 400 SERIES (2)	X	X	X	C	C	C	C	C	C	C	C	C	C	C	C			P	X	
12	CHROMIUM PLATE, TUNGSTEN, MOLYBDENUM	C	C	C	C	C	C	C	C	C	C	C	C	C	C	C			C	P	
13	CORROSION-RESISTANT STEELS 12 ±2% CR, AISI 400 SERIES (3)	X	X	X	C	P	X	C	C	C	C	C	C	C	C	C			X	X	
14	TIN, TIN PLATE, INDIUM	X	X	X	C	P	X	P	C	C	C	C	C	C	C	C			C	P	
15	LEAD	X	X	X	C	X	X	X	X	X	X	C	C	C	C	C			C	C	
16	ALUMINUM 2024, 2014, 7075																				
17	ALL STEELS EXCEPT CORROSION-RESISTANT TYPES (SEE 7, 11, 13)																				
18	ALUMINUM ALLOY 1100, 3003, 5052, 6063, 6061, 356	X	X	X	C	P	X	P	X	X	X	P	C	X	C	C			C	C	
19	CADMIUM & ZINC PLATE, GALVANIZE, BERYLLIUM, CLAD ALUMINUM	X	X	X	C	X	X	P	X	X	X	X	P	X	P	C			C	C	
20	MAGNESIUM																				

NOTES:

1. Alloys included in this group are: 301, 302, 321, and 347
2. Alloys included in this group are: 431, 440 (all classes)
3. Alloys included in this group are: 410, 416, 420.
4. The criteria set forth in this table conforms to AFBSD 62-88 finish criteria.
5. For further background information on the behavior of coupled dissimilar metals, refer to corrosion handbook edited by H. H. Uhlig.
6. This table does not apply to materials which have been prevented from making metal to metal contact with one another by the protection of paint or other coatings.
7. For intermediate materials selection, when required, refer to Specifications MIL-F-7179, MIL-P-21415, and AFBSD Exhibit 62-88.

The system cleaning fluids, either alone or in mixture with the system fluids, can also chemically attack the joint materials; therefore, it is important to ascertain the effects of other fluids, or their combinations, that will be induced into the system. Usually, this requires some development evaluation as a minimum.

d. Interdependence of Creating and Maintaining the Initial Seal

Creating the initial seal and maintaining the initial seal have been treated as two separate categories. However, they are not completely independent of one another.(10) Adjusting a factor to improve the initial seal can affect how well the initial seal is maintained. Both categories must be considered together during the initial design of the joint. For example, the selection of a softer material for the gasket rather than increasing preload can solve the immediate problem of creating the initial seal; however, the softer material may not retain sufficient properties to maintain the seal at all of the operating and environmental conditions intended for the joint.

2. Components of the Static Seal Joint

There are four basic components of the static sealed joint; the two flange members (which contain the prepared groove or surfaces for the gasket), the gasket, and the fastening members. All of these components must be considered together during the initial design of the joint. Each element of the joint design depends upon the other elements to obtain as well as maintain a satisfactory seal. The gasket selection must be based upon the fluid to be sealed, pressures, temperatures, allowable leakage rates, and the environments that will be encountered during the life of the joint. Then, the flanges and fasteners must be designed and constructed in a manner which will provide optimum utilization of the gasket properties. Each gasket design requires definite minimum machining requirements (i.e., surface finish, flatness, parallelism, and groove depths and diameters) as well as minimum gasket loading requirements to provide the intimate contact necessary to obtain the initial seal. Also, each gasket design imposes definite limitations upon the amount of joint deflection and distortion that can be accommodated while still maintaining the initial seal. The flanges and fasteners must be designed to meet these requirements under all operating conditions and environments of the joint.

3. Variables in the Static Seal Joint

The number of variables and combinations of variables in the static sealed joint is greater than 1.0×10^{22}. This extensiveness precludes a discussion of all the possibilities; however, the more common aspects that require designer consideration along with their posssible effects upon the over-all joint are presented in Table III.

(10) ibid.

Table III
(Sheet 1 of 4)

VARIABLES AND THEIR EFFECT UPON THE STATIC SEAL JOINT

A. GASKET

Variable	Effect
Configuration	Groove configuration and machining; installation, handling, and storage procedures and requirements; joint weight
Material	Fluid compatability; permeability; leakage rate obtainable; required sealing stress; corrosion resistance; mating surface preparation; permissible operating temperature; joint weight
Material Hardness	Required sealing stresses; mating surface preparation; leakage rate; joint weight
Material Modulus	Required squeeze to gain initial sealing stresses (deflections and distortions in joint); leakage rate; required bolt spacing
Material Strength	Required sealing stresses; leakage rate; operating temperature; operating pressures; joint weight
Material Extrusion Characteristics	Permissible deflection in joint; permissible pressure; permissible temperature; mating surface preparation and configuration; bolt spacing; joint weight
Material Contraction or Expansion Characteristics	Sealing stresses; mating groove or surface preparation; permissible deflection in joint; initial gasket groove fits or clearances
Material Elongation Characteristics	Permissible temperature; mating groove or surface preparation and configuration; initial gasket-groove fits or clearances
Material Creep, Cold Flow, Compression Set Characteristics	Groove or mating surface configuration and machining; storage life; leakage rate; required sealing stresses; permissible deflections and distortions in joint; permissible operating temperature
Surface Finish and Lay	Required sealing stresses; permissible reduction in sealing stresses (deflection and distortion in joint); leakage rate
Surface Flatness	Load to gain minimum required sealing stresses; permissible reduction in sealing stresses (deflection and distortion in joint)

Table III
(Sheet 2 of 4)

VARIABLES AND THEIR EFFECT UPON THE STATIC SEAL JOINT

A. GASKET (CONT'D)

Variable	Effect
Parallelism	Assembled parallelism between flanges; load to gain minimum sealing stresses; permissible reduction in sealing stresses (deflection and distortion in joint)
Coating or Plating	Required sealing stresses; corrosion resistance; permissible operating temperature; fluid compatibility; permeability; mating surface preparation
Thickness	Permissible deflection and distortion; required sealing stresses; permissible reduction in sealing stresses; creep rate; compression set; cold flow (pre-load reduction in fasteners); joint weight
Sealing Surface Width	Load to obtain required sealing stresses; leakage rate; mating surface preparation; permissible scores and scratches on mating surface; joint weight
Particle Contaminants on Surfaces	Sealing stresses; leakage rates; mating surface finish; groove reusability

B. FLANGES

Material	Fluid compatibility; permeability (castings, forgings, etc.); corrosion resistance; maximum permissible gasket hardness; permissible operating temperature and pressures
Material Modulus and Strength	Joint deflection rate; total deflection; joint distortions; bolt or fastener spacing
Material Contraction or Expansion Characteristics	Sealing stresses; permissible deflections in joint; surface flatness, roundness, parallelism, etc.; required fits and clearances
Material Elongation Characteristics	Permissible operating temperature; required fits and clearances
Material Creep Characteristics	Sealing stresses; distortions in joint; joint life; leakage rates
Heat Treatment and Process	Groove hardness; flange reusability; surface flatness, roundness, parallelism; sealing stresses; permissible deflections; permissible distortion; leakage rate

Table III
(Sheet 3 of 4)
VARIABLES AND THEIR EFFECT UPON THE STATIC SEAL JOINT

B. FLANGES (CONT'D)

Variable	Effect
Groove Surface Finish and Lay	Required sealing stresses; permissible deflection; leakage rate
Surface Flatness	Sealing stresses; permissible deflection; leakage rate
Groove Depth Tolerances	Sealing stress (squeeze); permissible deflection; leakage rate; operating temperature; operating pressure
Groove Diameter Tolerances	Sealing stresses; permissible deflection; leakage rate; operating temperature and pressures
Concentricities	Sealing stresses, permissible deflection; leakage rate; permissible temperature variations
Parallelism	Sealing stress; leakage rate; permissible deflection; permissible temperature and pressures; vibration sensitivity
Fastener Circle to Sealing Circle Dimension	Deflection and distortion in joint (section rotation); sealing stresses; vibration sensitivity; leakage rate; joint weight
Bolt Spacing	Deflection and distortion in joint (bowing); sealing stresses; vibration sensitivity; leakage rate

C. FASTENERS

Material	Load-Torque-Stress relationships (friction); permissible maximum torque (loading); electrolytic and other corrosion
Material Strengths and Modulus	Permissible maximum torque (loading and stress); required number of bolts; bolt size; load-elongation changes; joint weight
Material Contraction - Expansion Characteristics	Sealing stresses; fastener stresses; flange stresses
Material Elongation Characteristics	Bolt lengths; permissible change in bolt length; permissible temperature; sealing stresses
Bolt Lengths	Permissible total elongation; sealing stresses with temperature variations; axial load

Table III
(Sheet 4 of 4)

VARIABLES AND THEIR EFFECT UPON THE STATIC SEAL JOINT

C. FASTENERS (CONT'D)

Variable	Effect
Threads (Type, Pitch, Class)	Load-Torque-Stress relationships (friction); permissible maximum torque; resultant axial load; sealing stresses
Surface Finishes	Load-Torque-Stress relationships (friction); resultant axial load (sealing stresses)
Lubricants	Load-Torque-Stress relationships (friction); resultant axial load (sealing stresses)

D. ENVIRONMENTS

System Fluids	Corrosion; material strengths; required minimum sealing stresses; leakage rate; surface finishes
System Pressure	Required minimum sealing stresses; fastener loading; flange strength; required gasket strength; permissible deflections (extrusion); joint weight
Fluid Temperature	Sealing stresses (cold flow, creep, compression set, differential contractions); material strength and hardness; permissible deflections
External Loads (Axial, Tension, Bending Moments, Shear)	Required flange strength; required bolt strength; deflections and distortions; sealing stresses; joint weight
External Temperature	Temperature gradients across joint; sealing stresses (differential contraction)
External Pressure	Gasket material properties; sealing stresses
Vibration	Sealing stresses; gasket material properties (fluctuation in sealing stresses)

The design problem appears to be an extremely difficult task because almost all of the variables shown on Table III are interrelated in various degrees. This is true in many instances, particularly in the light-weight designs required for aerospace systems. Basically, the design problem entails selecting the gasket and material based upon the fluid to be sealed (compatibility), the allowable leakage rate (porosity and permeability), environmental and operating temperatures (material characteristics and changes including creep, compression set, cold flow, modulus, strength, hardness), exposure time, and cycle life (involves such considerations as fluid to be sealed, system cleaning solvents, and electromotive potentials).

The design problem further entails that the flanges and fasteners be designed to provide the necessary gasket loading while considering all of the conditions that will tend to unload or disturb the initial deformation (i.e., pressure, external axial tension, bending moments and shear loads, temperature changes and cycling effects, vibration effects, mechanical shock, and storage). This includes determining both the amount as well as the type of deflections and distortions in both the flanges and the fasteners under all of the conditions. The estimated deflections and distortions must be within the capabilities of the gasket selected.

4. Types of Static Seals

There are two basic types of static seals; face seals (gaskets) and diametral seals (gaskets). Examples of these two types as well as a combination of the face and diametral seals are shown on Figure No. 10. The face gasket seals between two flange faces while the diametral gasket seals between two flange diameters. The sealing forces are brought to bear upon the face gasket between the flange faces as the fastening members are tightened. The sealing forces are brought to bear upon the diametral seal by a predetermined diametral (or radial) squeeze. This squeeze occurs with the elastomeric O-rings and plastic diametral gaskets as the two flanges are placed together. In the case of the metal diametral gaskets (i.e., Conoseal®), the squeeze occurs as the fastening members are tightened.

The face gaskets can be further subdivided into crush gaskets and the so-called "pressure-energized" gaskets. All of the face gaskets depend upon flange loading (compression) to obtain gasket compression stresses (deformation) that are sufficient for initial sealing. The crush gaskets depend upon the flange pressures (loading) to maintain sufficient gasket stresses during the operating life of the joint. The pressure-energized gaskets are installed in pre-cut grooves which determine the amount of initial compression. Within limitations, the fluid pressures help to maintain sufficient gasket stresses with the pressure-energized gaskets during the operation of the joint.

Most of the static diametral seals are designed to provide very high sealing stresses (radial) with relatively low axial flange loadings by using the geometry of the gasket-groove configuration.

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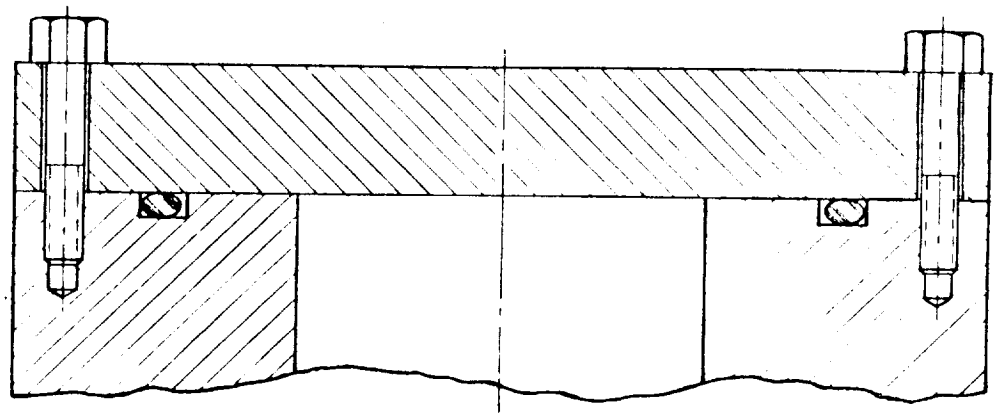
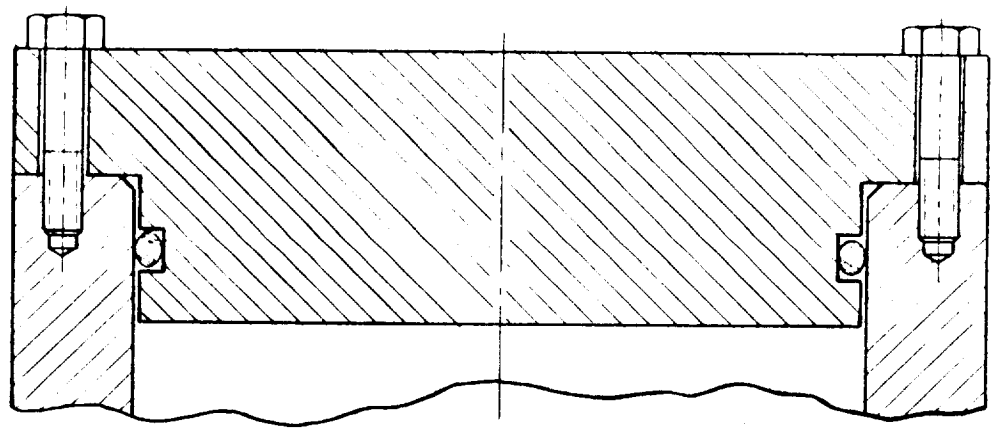
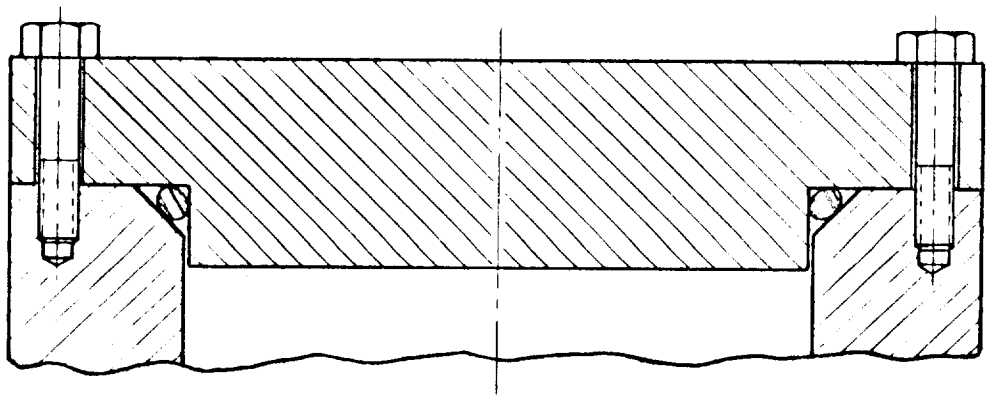
(O-RINGS)



STATIC FACE SEAL



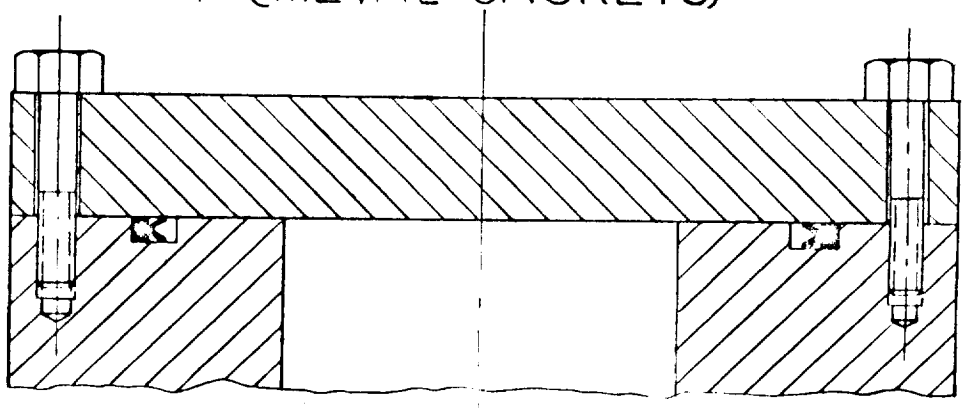
STATIC DIAMETRAL SEAL



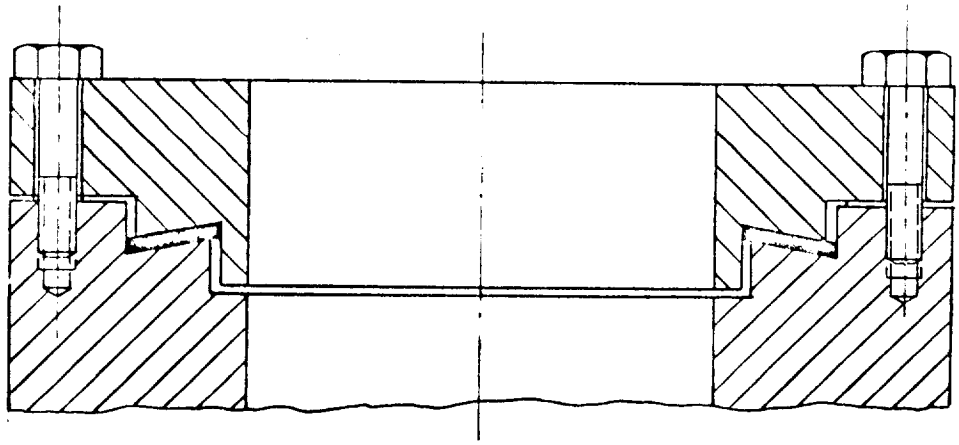
COMBINATION OF FACE AND DIAMETRAL SEAL

Figure 10
(Sheet 1 of 2)
Static Seal Joint Configurations
(O-Rings and Metal Gaskets)

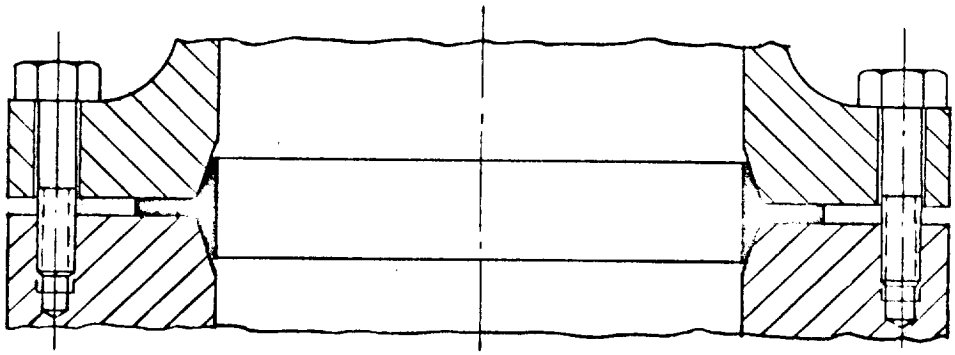
(METAL GASKETS)



STATIC FACE SEAL



STATIC DIAMETRAL SEAL



COMBINATION OF FACE AND DIAMETRAL SEAL

Figure 10
(Sheet 2 of 2)
Static Seal Joint Configurations
(O-Rings and Metal Gaskets)

The variation (or combination) face-diametral seal is designed to completely contain the seal. It is an attempt to gain the advantages of both the face and diametral seals.

5. Advantages and Disadvantages of the Various Types

There is no single seal type or configuration that is best-suited for all possible operating conditions and environments to which sealed joints can be subjected. The designer must select one of several possible gaskets based upon the fluid to be sealed, operating and environmental conditions, as well as many other factors, including cost considerations. A detailed comparison for specific applications must be made. Additional information concerning each type of gasket is presented in subsequent sections. This information should be helpful in selecting a gasket for a specific application. The comparisons made at this point are general in nature and they are based upon the basic types of seals (face gaskets or diametral gaskets) as well as the materials involved.

a. Face Gaskets

Flat gaskets and crush washers are relatively inexpensive and easy to manufacture. If sufficient seal loading can be provided, adequate plastic or elastic deformation can be obtained to achieve excellent sealing capabilities with relatively rough mating surfaces. The elastomeric and plastic materials provide good resiliency and unloading (joint deflection and distortion) capabilities, particularly when loaded well above minimum initial sealing stresses(7). The applications of elastomers and plastics are limited with regard to temperatures, compatibility, compression set during extended storage, cold flow deformation (when proper restraints are not provided), and liquid or gas permeation problems. The soft metals (such as 1100-0 aluminum and annealed copper) are somewhat sensitive to unloading of the initial sealing stresses and require structurally-rigid sealing surfaces. The soft metals can provide a relatively large operating temperature range, depending upon the creep characteristics of the material. Flange configurations and surfaces can be devised to reduce the high flange loading requirements (i.e., concentric serrations and knife edges).

Most of the pressure-energized gaskets are proprietary items. O-rings are exceptions to this, but some materials used to manufacture the elastomeric and filled plastic O-rings are proprietary. In addition, some of these designs are patented. The pressure-energized gaskets require much lighter flange loadings to obtain the initial seal, but they require far more refined mating surfaces as well as more complicated machining than the flat gaskets and crush washers. However, if all the design conditions and machining requirements can be satisfied, the pressure-energized gasket can provide much better unloading capabilities than any of the other gasket designs.

(7) Rathbun, Jr., F. O., op. cit.

b. Diametral Gaskets

The metal diametral gaskets gain very high sealing stresses with relatively low axial loading of the flanges. The geometry between the groove and gasket is such that a mechanical advantage or wedging action provides much higher radial loads than are applied axially. This is particularly advantageous if allowable leakage rates are very low. Its disadvantage is that the critical machining is on the diameters in each flange. For small joints, the machining requirements are not too great a problem; however, as the joint size increases, the machining of the sealing diameters can become a formidable problem.

6. Engineering Evaluation

a. Capabilities

Before beginning the design of a joint for a particular application, the various types of gaskets should be evaluated with respect to their application. It is necessary to know the available materials and coatings, approximate loading requirements, machining requirements, service temperatures, resistance to fluids being sealed and other environments, capabilities of the gasket to withstand deflections and distortions in the joint, as well as many other variables. To examine all of the gasket characteristics and capabilities is a difficult task. However, some of the basic considerations were collected and are presented on Figure No. 11. The gaskets are grouped according to their more common usage; pressure energized face seals (internal and external pressure), flat gaskets and crush washers, diametral seals, and special groove types. Other groupings were attempted but the one presented has proven to be the most useful for the designer. The information presented on Figure No. 11 was gained from available literature, catalog information, and discussions with the manufacturer (in the case of the proprietary seals). Also, much of this information is based upon actual laboratory testing. Because the cross-section of the gasket affects the requirements and, to a larger extent, its capabilities, the information presented is based upon nominal 1/8-in. cross-section gaskets unless otherwise noted in the remarks column. In general, the smaller sections require lower loading and provide less deflection and distortion capabilities whereas the larger cross-sections require higher loading but provide greater deflection and distortion capabilities. This evaluation was used as a basis for selecting gaskets for actual laboratory evaluations over a broad range of services (i.e., petroleums, storable propellants, and cryogenics).

The minimum gasket seating loads required to obtain the initial seal for each type of gasket is shown in the design requirements section of Figure No. 11. In some cases, the gasket loading column refers to a separate figure. The loading requirements for the elastomeric O-rings are shown on Figure No. 12. Loading requirements for the flat gaskets and crush washers are shown on Table IV and Figures No. 13 through No. 16. The loading requirements for the solid section metal rings are shown on Figure No. 17. The allowable axial, circular, and differential radial deflections that the gasket can withstand while still maintaining the initial seal is also presented

FOLDOUT FRAME 1.

FOLDOUT FRAME 2

MANUFACTURER	CONFIGURATION	JACKET OR COATING / BASE MATERIAL	APPROX COST	ASC STD DATE	DESIGN REQUIREMENTS: GASKET LOADING	ALLOWABLE FLANGE DEFLECTION (INCHES): AXIAL DEFORMATION	CIRC. DEFORMATION	RADIAL DEFORMATION	NEW ALLOWABLE FLANGE DISTORTION	SURFACE FINISH	SURFACE STRUCTURE	DEPTH TOL	DIA TOL	OPERATING TEMP RANGE	RADIATION RESISTANT	REMARKS	TEST PRESSURE	TEST MEDIA	TEST TEMP	LEAK RATE	REMARKS
JOHNS-MANVILLE CO.	LAPPED	NONE / CRES OR AL ALLOY		NO	SEE FIGURE 15	.0000	.0000	?	.0000	80 √ C	.001	N/A	N/A	-423°F TO 1000°F	YES	FAIR DEFORMATION. EXTREMELY HIGH ALLOWABLE FLANGE DEFLECTION. PROBLEMS: HIGH GASKET LOADING REQUIRED, HANDLING, INSTALLATION VERY CRITICAL, VIBRATION SENSITIVE. INSTALLATION SUPPORTED BY MATING SURFACES AT FLANGES.					NO ASC TEST EXPERIENCE.
	COBALT WASHER / FLAT OR GRIP	NONE / SYNTHETIC UNPAINED OR METALLIC	15.00 / 2.50	NO	SEE TABLE 1	.0000 TO .0000 DEPENDING ON MATERIAL	.0000 AND .0000 DEPENDING ON MATERIAL	.0000 AND .0000 DEPENDING ON MATERIAL	.0000	52-63 √ C	.001	N/A	N/A	-423°F TO 1000°F	YES	HARD TO ACCOMMODATE ALLOWABLE FLANGE DEFLECTION WITH MATERIALS. STRESS. HIGH GASKET LOADING, HANDLING, INSTALLATION VERY CRITICAL, VIBRATION AND STRESS SENSITIVE. GENERALLY REQUIRES SUPPORT BY OTHER MEANS (BOLTS, THICK FLANGES, ETC.).	0-1500 PSIG	O_2, LN_2	ROOM TEMP -300°F	MORE THAN 1.0×10^{-7} STD CC/SEC	TITAN I SPW. GAP, HAS PROBLEMS.
FLEXITALLIC JOHNS-MANVILLE GASKETS	SPIRAL WOUND / GASKET	FILLER: SYNTHETIC, UNPAINED METALLIC / VARIOUS CRES	25.00 / 1.50	NO	SEE FIGURE 15	.0005 MAX	.0005 MAX	?	.0005 MAX	64-125 √ C	.0002 PER CIRC. INCH	±.0005 MAX	±.010	-423°F TO 1000°F	YES	FAIR DEFORMATION. ALLOWABLE FLANGE DEFLECTION DEPENDS ON MATERIALS, VIBRATION, ETC. (INCLUDES HIGH GASKET LOADING REQUIRED, VIBRATION SENSITIVE, HAS A RELUCTANCE TO GASKET MOVEMENT, INSTALLATION VERY CRITICAL, SPECIAL GROOVE REQUIRED, DATA BASED ON 1.5" THICK FLANGE WITH C/S.	0-1500 PSIG	O_2	ROOM TEMP.	MORE THAN 10×10^{-7} STD CC/SEC IN INSTALLATION	TITAN I & II SPW. GAP, HAVE PROBLEMS.
JOHNS-MANVILLE CO.	RING	NONE / CRES OR AL ALLOY, GOLD, LUPYAR		NO	SEE FIGURE 17	.0000 DEPENDING ON GROOVE DESIGN	.0000 DEPENDING ON GROOVE DESIGN	.0000 DEPENDING ON GROOVE DESIGN	.0000	?	?	?	?	-423°F TO 1000°F	YES	FAIR TO GOOD DEFORMATION. EXTREMELY HIGH TO GOOD ALLOWABLE FLANGE DEFLECTION. PROBLEMS: HIGH GASKET LOADING, INSTALLATION VERY CRITICAL. SPECIAL GROOVES REQUIRED WHICH DEPEND FOR GASKET SHEAR DEFORMATION BUT COMPLICATE.					NO ASC TEST EXPERIENCE.
JOHNS-MANVILLE CO.	OVAL	NONE / CRES OR AL ALLOY		NO	SEE FIGURE 17	.0000	.0000	?	?	?	?	?	?	-423°F TO 1000°F	YES	FAIR TO GOOD DEFORMATION. EXTREMELY HIGH TO GOOD ALLOWABLE FLANGE DEFLECTION. PROBLEMS: HIGH GASKET LOADING, INSTALLATION VERY CRITICAL. GROOVES DESIGNED FOR GASKET SHEAR DEFORMATION BUT.					NO ASC TEST EXPERIENCE.
JOHNS-MANVILLE CO.	OCTAGONAL	NONE / CRES OR AL ALLOY		NO	SEE FIGURE 17	.0000	.0000	?	?	?	?	?	?	-423°F TO 1000°F	YES	SAME AS OVAL.					NO ASC TEST EXPERIENCE.
JOHNS-MANVILLE CO.	DELTA	NONE / CRES OR AL ALLOY		NO	SEE FIGURE 17	.0000	.0000	?	?	?	?	?	?	-423°F TO 1000°F	YES	GOOD DEFORMATION. DUE TO FAIR ALLOWABLE FLANGE DEFLECTION DEPENDING ON GROOVE DESIGN. GROOVE SURFACE INSPECTED FROM DAMAGE. PROBLEMS: INSTALLATION AND HANDLING CRITICAL, SPECIAL GROOVE DESIGN. NO GASKET LOADING INFORMATION AVAILABLE.					NO ASC TEST EXPERIENCE.
GARAN CORP.	GARAN	NONE / CRES OR AL ALLOY	100.00 / 10.00	NO	300 LBS/CIRC INCH	.005 MAX	.002 MAX	.0001 DEPENDS ON MATERIAL MAGS-SEC. FROM A DIA.	.0000	32 √	.010	±.010	±.001	-423°F TO 1000°F	YES	FAIR DEFORMATION. FAIR ALLOWABLE FLANGE DEFLECTION. PROBLEMS: SPECIAL GROOVE REQUIRED, GROOVE SEALING DIAMETER TOLERANCE, DIFFERENTIAL RADIAL DEFLECTION.	0-750 PSIG	O_2, GH_2, LF_2	-320°F TO -320°F	LEAK THAN 1.0×10^{-3}	LAB TESTS ONLY
NATIONAL UTILITIES CO.	MECHSEAL	NONE / CRES OR AL ALLOY	350.00 / 27.00	NO		.005 MAX	.002 MAX	.001 DEPENDS ON MATERIAL MAGS-SEC. FROM A DIA.	.0002 MAX	64 √	.001	±.005	±.003	-423°F TO 1000°F	YES	FAIR DEFORMATION. FAIR ALLOWABLE FLANGE DEFLECTION. GROOVE SURFACE PROTECTED FROM DAMAGE. PROBLEMS: GROOVE SEALING DIAMETER TOLERANCE, SPECIAL GROOVES REQUIRED. DIFFERENTIAL RADIAL DEFLECTION.					LAB TESTS ONLY
INTERNATIONAL CORP.	INTERSEAL / SEAL	NONE / CRES	10.00	NO	2000 LBS/CIRC INCH	.005 MAX	?	?	?	√	.001	±.002	±.001	-423°F TO 1000°F	YES	FAIR DEFORMATION. FAIR ALLOWABLE FLANGE DEFLECTION. GROOVE SURFACE PROTECTED FROM DAMAGE. PROBLEMS: HIGH GASKET LOADING, GROOVE SEALING DIAMETER TOLERANCE.	0-1000 PSIG	O_2, N_2	ROOM TEMP.	LEAK THAN 1.0×10^{-6} STD CC/SEC TO 1.0×10^{-7} STD CC/SEC	LAB TESTS ONLY
AEROJET-GENERAL CORP.	AEROJET	NONE / VARIOUS CRES OR AL ALLOYS	UNKNOWN	NO	?	?	?	?	?	?	?	?	?	-423°F TO 1000°F	YES	PRELIMINARY DESIGN STAGE. NO D & D TESTING.					NO ASC TEST EXPERIENCE.
AEROQUIP CORP. MARMAN DIVISION	CONICAL	NONE / CRES SUPPLIED OR AL ALLOY	28.00 / 2.50	YES AS 4096	400-600 LBS/CIRC INCH	.020 MAX (SINGLE SEAL)	.005 MAX (SINGLE SEAL)	.008 MAX (SINGLE SEAL)	.008 MAX (SINGLE SEAL)	√	.010	±.005	±.002	-423°F TO 1750°F	YES	GOOD DEFORMATION. GOOD ALLOWABLE FLANGE DEFLECTION. GROOVE SURFACE PROTECTED FROM DAMAGE. PROBLEMS: GROOVE SEALING DIAMETER TOLERANCE, SPECIAL GROOVES REQUIRED, DIFFERENTIAL RADIAL DEFLECTION. ALLOWANCE BETWEEN MATING WITH GASKETS CRITICAL. DATA BASED ON 1.5" C/S.	0-1500 PSIG	N_2, LN_2, H_2, LH_2, O_2	-1000°F ROOM TEMP. -425°F	NO 0.1 x 10^{-7} STD CC/SEC O_2	LAB TESTS AND REMARKS 0.1 AND DEPTH ENGINE EXPERIENCE
GRAY TOOL CO.	GRAYLOC SEAL	NONE / CARBON STEEL	1000 / 10.00	NO	?	?	?	?	?	BOLT BORE 100	4 MIN. LAND FRACTURE	?	?		YES	FAIR DEFORMATION. ALLOWED FOUR ALLOWABLE FLANGE DEFLECTION. PROTECTED GROOVE SURFACE. PROBLEMS: GROOVE SURFACE FINISH AND PARALLELISM, SAFALOY; SPECIAL GROOVE REQUIRED.					NO ASC TEST EXPERIENCE.
AEROJET-GENERAL AND SEALOL	AERO-CEL	VARIOUS PLATING GOLD, NI-TEP, ETC. / VARIOUS CRES OR AL ALLOYS	UNKNOWN	NO	?	?	?	?	?	√	?	?	?	-423°F TO 1000°F	YES	FAIR DEFORMATION. GOOD ALLOWABLE FLANGE DEFLECTION. PROTECTED GROOVE SURFACE. PROBLEMS: SPECIAL GROOVES REQUIRED.	0-1500 PSIG	O_2, H_2, N_2	AMB.	MORE THAN 1 CC/SEC	TOLERANCE STUDY HARDWARE

*NEUTRON FLUX = 1.0×10^{13} NEUT/CM²-SEC., GAMMA FLUX = 1.0×10^{14} REM/CM²-SEC.

Figure 11 A
(Sheet 1 of 2)
Static Seal Review

36-A

Fig. 11-B

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36-B

FOLDOUT FRAME 1

FOLDOUT FRAME 2

MANUFACTURER	CONFIGURATION	JACKET OR COATING	BASE MATERIAL	APPROX COST (TOP)	APPROX COST (BOTTOM)	AEC STD SIZE	SEATING LOADING	ALLOWABLE FLANGE DEFLECTION: AXIAL SEPARATION	ALLOWABLE FLANGE DEFLECTION: [illegible]	ALLOWABLE FLANGE DEFLECTION: [illegible]	MAX ALLOWABLE FLANGE [illegible]	SURFACE FINISH	SURFACE FLATNESS TIR	DEPTH TOL.	DIA TOL.	OPERATING TEMP RANGE	RADIATION RESISTANT	REMARKS	TEST PRESSURE	TEST MEDIA	TEST TEMP	LEAK RATE	REMARKS
VARIOUS	O-RING	NONE	SYNTHETIC ORGANIC & PLASTIC	70.00	3.50	YES	DEPENDS ON SQUEEZE (SEE FIG. 12)	DEPENDS ON OPER. TEMP. & PRESS. .010 MAX.	DEPENDS ON OPER. TEMP. & PRESS. .010 MAX.	DEPENDS ON OPER. TEMP. & PRESS.	DEPENDS ON OPER. TEMP. & PRESS. .010 MAX.	32-64 √c	.005 MAX.	DEPENDS ON C/S ±.001 TO ±.0015	DEPENDS ON TEMP. UP TO ±.010	LIMITED RANGE FROM -100°F TO +500°F	NO	GOOD DEFORMATION. HIGH LOADING REQUIRED FOR LOW TEMPERATURE SERVICE. PROBLEMS: H_2 & H_e PERMEABILITY, LOW TEMPERATURE BRITTLENESS, MATERIAL-FLUID COMPATIBILITY, COMPRESSION SET, COLD FLOW, SPECIAL GLANDS FOR LOW AND HIGH TEMPERATURE SERVICE.	0-5000 PSIG	N_2, N_2O_4, LH_2, LO_2, STD FLUID, LUBE OIL	-320°F TO +120°F	VARIABLE	TITAN I AND II GROUND EXP.
NACO ENGINEERING DIV. U.S. GASKET CO.	NACO SEAL	TEFLON (FEP OR TFE) KEL-F	CRES SPRING	250.00	15.00	YES AS 4061 AS 4075	50-100 LBS/CIRC INCH	.015 MAX. @ 1200 PSIG @ AMB.	.008 MAX. CIRC. IN. TOTAL .005 MAX.	DEPENDS ON OPER. TEMP. .006 MAX.	DEPENDS UP OPER. TEMP. & PRESS. .010 MAX.	32 √c	.005 MAX.	±.001	1.0 TO 10.0 DIA'S ±.001 TO ±.004	-423°F TO +300°F	NO	GOOD DEFORMATION. GOOD ALLOWABLE FLANGE DEFLECTION. PROBLEMS: LOW TEMPERATURE BRITTLENESS, H_2 & H_e PERMEABILITY APPROXIMATELY 1.0 × 10⁻⁵ STD CC/SEC/INCH CIRCUMFERENCE. UNI-DIRECTIONAL. HIGH PRESSURE EXTRUSION. SENSITIVE TO RADIAL SCRATCHES, ACIDS, ETC. ON MATING SURFACE.	0-1500 PSIG	N_2, N_2O_4, LH_2, H_e, N_2O	120°F ROOM TEMP.	LESS THAN 1.0 × 10⁻⁵ STD CC/SEC H_e	TITAN II GRD. EXP.
AEROQUIP CORP. AIRCRAFT DIVISION	OMNISEAL	TEFLON (FEP OR TFE) KEL-F OR COPPER & ETHAR. ELGILOY	CRES SPRING	750.00	12.00	YES AS 4062 AS 4075	50-300 LBS/CIRC INCH	.015 MAX. @ 1200 PSIG @ AMB.	.003 MAX./CIRC. INCH TOTAL .005 MAX.	DEPENDS ON OPER. TEMP. .006 MAX.	DEPENDS UP TEMP. & PRESS. .010 MAX.	32 √c	.005 MAX.	±.001	1.0 TO 10.0 DIA. ±.001 TO ±.004	-423°F TO +300°F	NO	ESSENTIALLY SAME AS NACO SEAL.	0-1500 PSIG	N_2, N_2O	ROOM TEMP	LESS THAN 1.0 × 10⁻⁵ STD CC/SEC	TITAN II GRD. EXP.
												SEE A-D5232 & A-D5233											
BAKKEL ENGINEERING CO.	CREAVEY SEAL	TEFLON TUBE	CRES SPRING			NO	?	?	?	?	DEPENDS ON OPER. TEMP. & PRESSURE	32 √c	.005 MAX.	±.005	+.050 -.000	-423°F TO +400°F	NO	GOOD DEFORMATION. FAIR ALLOWABLE FLANGE DEFLECTION. MULTI-DIRECTIONAL FOR LOW PRESSURES. PROBLEMS: LH_2 BRITTLENESS, H_2 & H_e PERMEABILITY/TUBING AND JOINT. PLASTIC	—	—	—	—	NO AEC TEST EXPERIENCE.
T&C SEAL CORP.	T&C-RING	TEFLON	RIDGED WAVE OR CABLE CRES SPRING			?	?	?	?	?	?	?	?	?	?	?	NO	GOOD DEFORMATION. GOOD ALLOWABLE FLANGE DEFLECTION. UNI-DIRECTIONAL. PROBLEMS: LOW TEMPERATURE BRITTLENESS, H_2 & H_e PERMEABILITY.	—	—	—	—	NO AEC TEST EXPERIENCE.
PRESSURE SCIENCE INC.	"C" RING	VARIOUS PLATING GOLD, SILVER, TEFLON, ETC.	CRES, COPPER, OR AL ALLOYS	175.00	10.00	YES	400-700 LBS/CIRC INCH	.001 MAX.			.003 MAX.	32 √c	.002 PER CIRC. INCH MAX.	±.002	+.005 -.000	-423°F TO +1200°F	YES	FAIR DEFORMATION. POOR ALLOWABLE FLANGE DEFLECTION. PROBLEMS: SENSITIVE TO RADIAL SCRATCHES, HANDLING, PLATING ADHERENCE. PARALLELISM BETWEEN MATING SURFACES AT ASSEMBLY.	3-7000 PSIG	N_2, H_e	ROOM TEMP	0.35 - 13.5 STD CC/SEC	LAB TESTS ONLY
PRESSURE SCIENCE INC.	"E" RING	VARIOUS PLATING GOLD, SILVER, TEFLON, ETC.	VARIOUS [illegible]		25.00	YES AS 4063 AS 4075	300-500 LBS/CIRC INCH	.010 MAX.	.005 MAX.	.001 MAX.	.005 MAX.	32 √c	.002 MAX.	±.005	±.001 TO ±.005	-423°F TO +1200°F	YES	FAIR DEFORMATION. FAIR ALLOWABLE FLANGE DEFLECTION. PROBLEMS: SENSITIVE TO RADIAL SCRATCHES, HANDLING. PARALLELISM BETWEEN MATING SURFACES AT ASSEMBLY.	3-1500 PSIG	H_e	ROOM TEMP.	LESS THAN 1.0 × 10⁻⁵ STD CC/SEC	LAB TESTS ONLY
												SEE ASD 5232 & ASD 5233											
UNITED AIRCRAFT PRODUCTS ADVANCED PRODUCTS D.A.B. MFG. CO.	METAL O-RING	VARIOUS PLATING GOLD, SILVER, TEFLON, ETC.	VARIOUS CRES OR AL ALLOY	150.00	7.00	YES AS 7011 AS 7017	700-2800 LBS/CIRC INCH	.001 MAX.			.001 MAX.	32 √c	.0002 PER CIRC. INCH MAX.	±.002	+.004 -.000	-423°F TO +1200°F	YES	FAIR DEFORMATION. POOR ALLOWABLE FLANGE DEFLECTION. PROBLEMS: SENSITIVE TO RADIAL SCRATCHES, HANDLING, PLATING ADHERENCE. PARALLELISM BETWEEN MATING SURFACES AT ASSEMBLY.	0-1000 PSIG	N_2, H_e	500°F ROOM TEMP. -250°F	.35 - 10.6 STD CC/SEC	LAB TESTS ONLY
VARIDCOP MFG. CO.	"V" SEAL	VARIOUS PLATING GOLD, SILVER, TEFLON, ETC.	VARIOUS CRES [illegible]	1500.00	150.00	YES AS 4064 AS 4076	30-60 LBS/CIRC INCH	.004 MAX.	.0005 MAX CIRC. IN. .002 MAX TOTAL	.010 MAX	.001 MAX.	16-32 √c	.0002 PER CIRC INCH MAX.	±.001	±.001 TO ±.005	-423°F TO +1500°F	YES	POOR DEFORMATION (SEALING EDGE WIDTH). FAIR ALLOWABLE FLANGE DEFLECTION. PROBLEMS: SENSITIVE TO RADIAL SCRATCHES, EXTREME HANDLING, PLATING ADHERENCE. PARALLELISM BETWEEN MATING SURFACES AT ASSEMBLY.	0-1500 PSIG	N_2, N_2O, N_2O_4	ROOM TEMP	1.5 - 6.5 STD CC/SEC	LAB TESTS AND HARDWARE
												SEE ASD 5232 & ASD 5233											
HARREL ENG.		VARIOUS PLATING GOLD, SILVER, TEFLON, ETC.	VARIOUS CRES OR AL ALLOYS	1000	30.00	YES	50-200 LBS/CIRC INCH	.002 MAX.			.0005 MAX.	8-32 √c	.0005 MAX.	±.001	±.010	-423°F TO +1200°F	YES	POOR DEFORMATION. FAIR ALLOWABLE FLANGE DEFLECTION. PROBLEMS: SENSITIVE TO RADIAL SCRATCHES, HANDLING. PARALLELISM BETWEEN MATING SURFACES AT ASSEMBLY.	0-2000 PSIG	N_2O, GH_2, H_e	ROOM TEMP +1100°F	.16 STD CC/SEC TOO GREAT TO MEASURE	LAB TESTS AND TITAN II GRD. EXP.
ROCKETDYNE	RD-FLEX	VARIOUS PLATING GOLD, SILVER, TEFLON, ETC.	CRES OR AL ALLOY	250.00		NO	50-100 LBS/CIRC INCH	.006 MAX.			.002 MAX.	16-32 √c	.001 MAX.	+.002 -.000	+.010 -.000	-423°F TO +1200°F	YES	FAIR DEFORMATION. FAIR ALLOWABLE FLANGE DEFLECTION. PROBLEMS: SENSITIVE TO RADIAL SCRATCHES, HANDLING, PLATING ADHERENCE.					NO AEC TEST EXP.
DEL MFG. DIV.	PRESFLEX TYPE A	VARIOUS PLATING GOLD, SILVER, TEFLON, ETC.	CRES OR AL ALLOY	250.00	12.00	YES	50-200 LBS/CIRC INCH	.002 MAX.			.002 MAX.	16 √c	.0002 MAX TIR CIRC. INCH	+.005 -.000	+.030 -.000 ?	-423°F TO +1200°F	YES	FAIR DEFORMATION, FAIR ALLOWABLE FLANGE DEFLECTION. PROBLEMS: SENSITIVE TO RADIAL SCRATCHES, HANDLING, PLATING ADHERENCE.					NO AEC TEST EXP.
PARKER SEAL CO. TETRAFLUOR	V-SEAL TYPICAL	VARIOUS PLATING GOLD, SILVER, TEFLON, ETC.	VARIOUS CRES, [illegible] AL ALLOYS	1000	10.00	YES AS 4066 AS 4078	1100 LBS/CIRC INCH	.010 MAX.	.003 MAX.	.004 MAX.	.002 MAX.	16-32 √c	.002 MAX.	±.001	±.001 TO ±.008	-423°F TO +1500°F	YES	FAIR DEFORMATION. FAIR ALLOWABLE FLANGE DEFLECTION. PROBLEMS: SENSITIVE TO RADIAL SCRATCHES, HANDLING. PARALLELISM BETWEEN MATING SURFACES AT ASSEMBLY.	0-200 PSIG	H_e	ROOM TEMP. 500°F TO -320°F	LESS THAN 2.9 × 10⁻⁹ STD CC/SEC	SMALL SEALS TESTED TO DATE .750 - 1.250 DIA.
												SEE ASD 5232 & ASD 5233											
ASTRO-DYNAMICS SEAL DIVISION AEROSPACE COMPONENTS CORP.		VARIOUS PLATING GOLD, SILVER ETC.	INCONEL-X		40.00	YES AS 4065 AS 4077	50-500 LBS/CIRC INCH	.005 MAX.				32 √c	.002 MAX.	±.002 ?	±.001 TO ±.006	-423°F TO +1200°F	YES	FAIR DEFORMATION. FAIR ALLOWABLE FLANGE DEFLECTION. PROBLEMS: SENSITIVE TO RADIAL SCRATCHES, HANDLING, PLATING ADHERENCE. PARALLELISM BETWEEN MATING SURFACES AT ASSEMBLY.	0-1000	H_e	ROOM TEMP.	LESS THAN 10 × 10⁻⁷ STD CC/SEC [illegible] STD CC/SEC	LAB TESTS AND HARDWARE
												SEE ASD 5232 & ASD 5233											
KOPPERS CO.	PRESSURE LOCK	NONE	CRES	[illegible]		YES	200-1000 LBS/CIRC INCH	?	?	?	?	8-32 √c	?	?	?	?	YES		0-1650 PSIG	GN_2	ROOM TEMP.	LESS THAN 1.0 STD CC/SEC	LAB TESTS ONLY
E. D. WIGGINS OIL TOOL CO.	WIG-X	VARIOUS PLATING GOLD, SILVER ETC.	VARIOUS CRES	530.00	30.00	YES	300-1000 LBS/CIRC INCH	.0005 MAX.			.0005 MAX	8-16 √c	.0005 MAX	±.005 ?	±.0025	-423°F TO +1200°F	YES	POOR DEFORMATION. POOR ALLOWABLE FLANGE DEFLECTION. PROBLEMS: EXTREMELY SENSITIVE TO RADIAL SCRATCHES, HANDLING, HIGH LOADING. PARALLELISM BETWEEN MATING SURFACES AT ASSEMBLY.	0-500 PSIG	N_2, H_e	ROOM TEMP.	.08 STD CC/SEC N_2 .11 STD CC/SEC H_e	LAB TESTS ONLY
METAL SEAL CO.	METALFIX SPRING SEAL	VARIOUS PLATING GOLD, SILVER, TEFLON, ETC.	VARIOUS CRES OR AL ALLOYS	1000	20.00	NO	50-100 LBS/CIRC INCH	.002 MAX.			.001 MAX.	8-16 √c	.0002 MAX PER CIRC INCH	±.0015	±.005	-423°F TO +1200°F	YES	POOR DEFORMATION. FAIR ALLOWABLE FLANGE DEFLECTION. PROBLEMS: SENSITIVE TO RADIAL SCRATCHES, HANDLING, REQUIRES SPECIAL GLAND. PARALLELISM BETWEEN MATING SURFACES AT ASSEMBLY.	0-1500 PSIG	N_2	ROOM TEMP	MORE THAN 1.0 × 10⁻² STD CC/SEC	LAB TESTS ONLY
[illegible] AIRTONIC		NONE	300 CRES 17-4 PH, A-286									16 √c	.0002 MAX PER INCH [illegible]	±.0005	±.001	[illegible] TO +700°F	YES	FLAT [illegible] N_2 [illegible]					LAB TESTS ONLY
AEP INDUSTRIES AEROJET-GENERAL		LEAD	ALUM	500.00		YES	600-900 LBS/CIRC INCH	?	?	?	?	32 √c	.0005 MAX	±.002 ?	±.001 ?	-423°F TO +700°F	YES	GOOD DEFORMATION. DOES NOT MELT OR LOSE STIFFNESS TO [illegible] DEFORMATION. PROBLEMS: HANDLING.	0-800 PSIG	H_e, N_2, LH_2	-423°F TO ROOM TEMP	LESS THAN 1.0 × 10⁻⁶ STD CC/SEC	TESTS IN [illegible] EXPERIENCE. 36" TO 52" DIA'S ONLY. LEAK RATES LESS THAN 1.0 × 10⁻⁶ STD CC/SEC TOTAL.

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Fig. 11-71

Figure 11-72
(Sheet 2 of 2)

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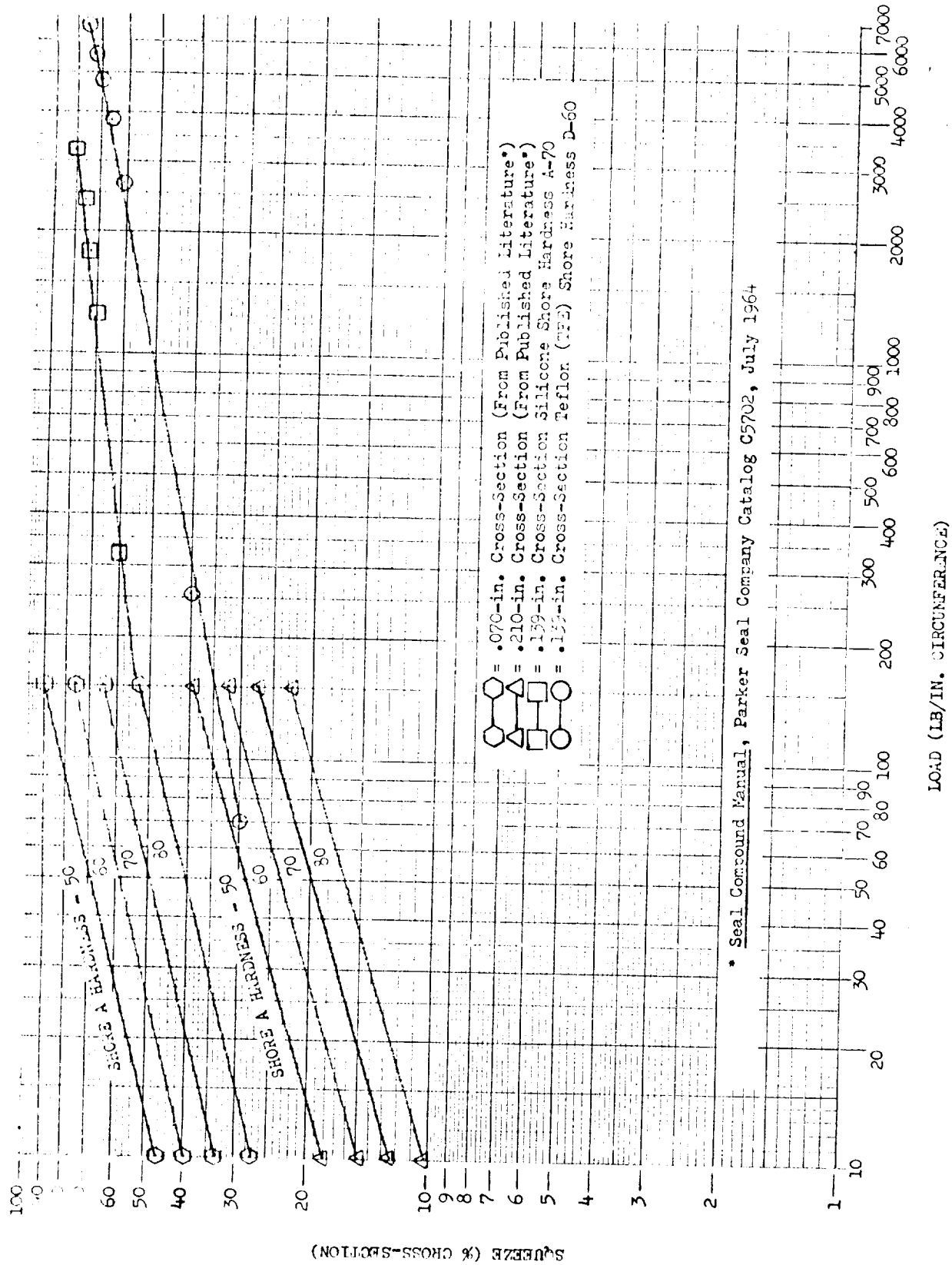


Figure 12

Load vs. Squeeze (O-Ring Gaskets)

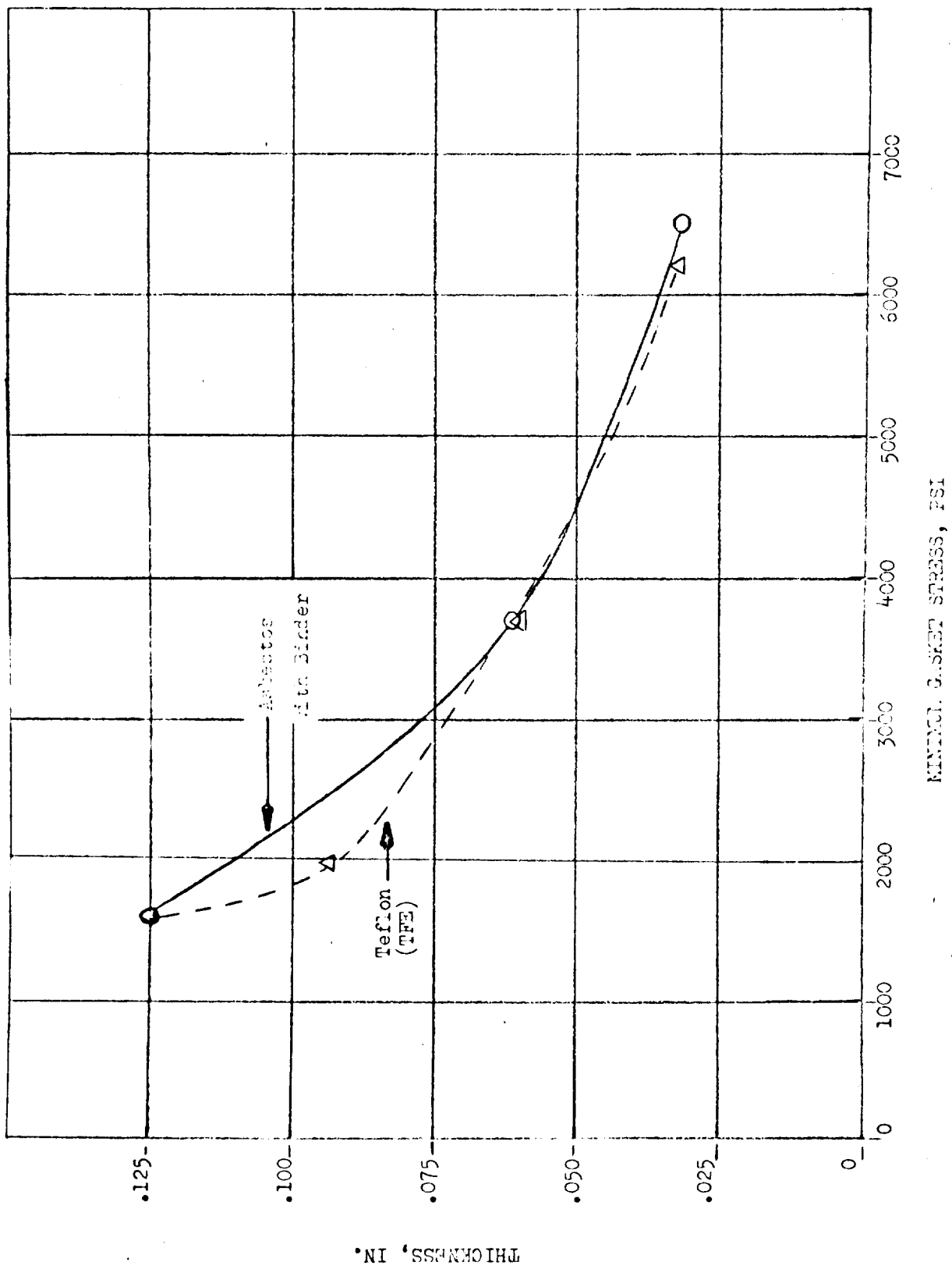


Figure 13
Minimum Gasket Stress vs. Thickness for Flat Gaskets

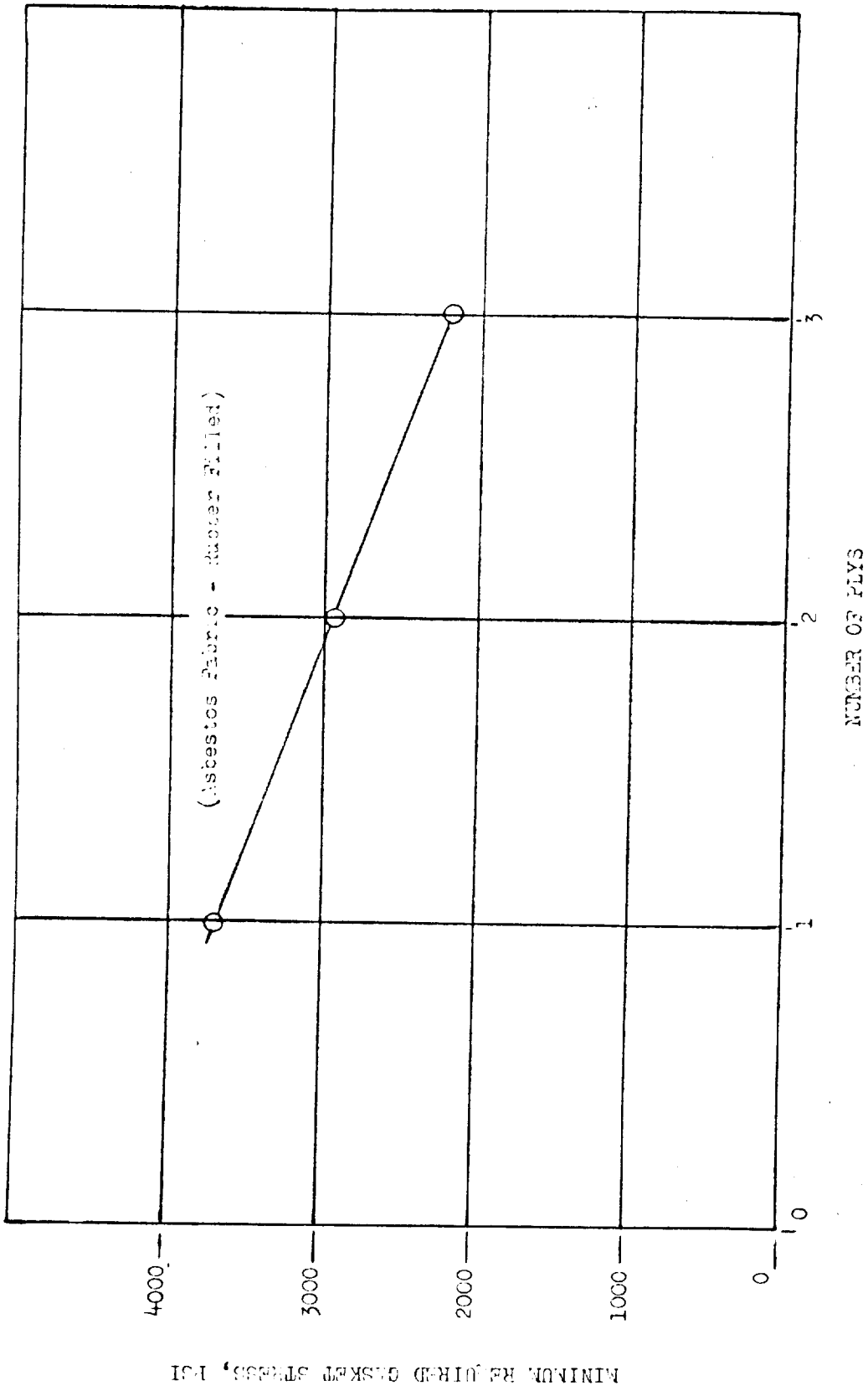


Figure 14
Minimum Required Gasket Stress vs.
Number of Plys for Flat Gaskets

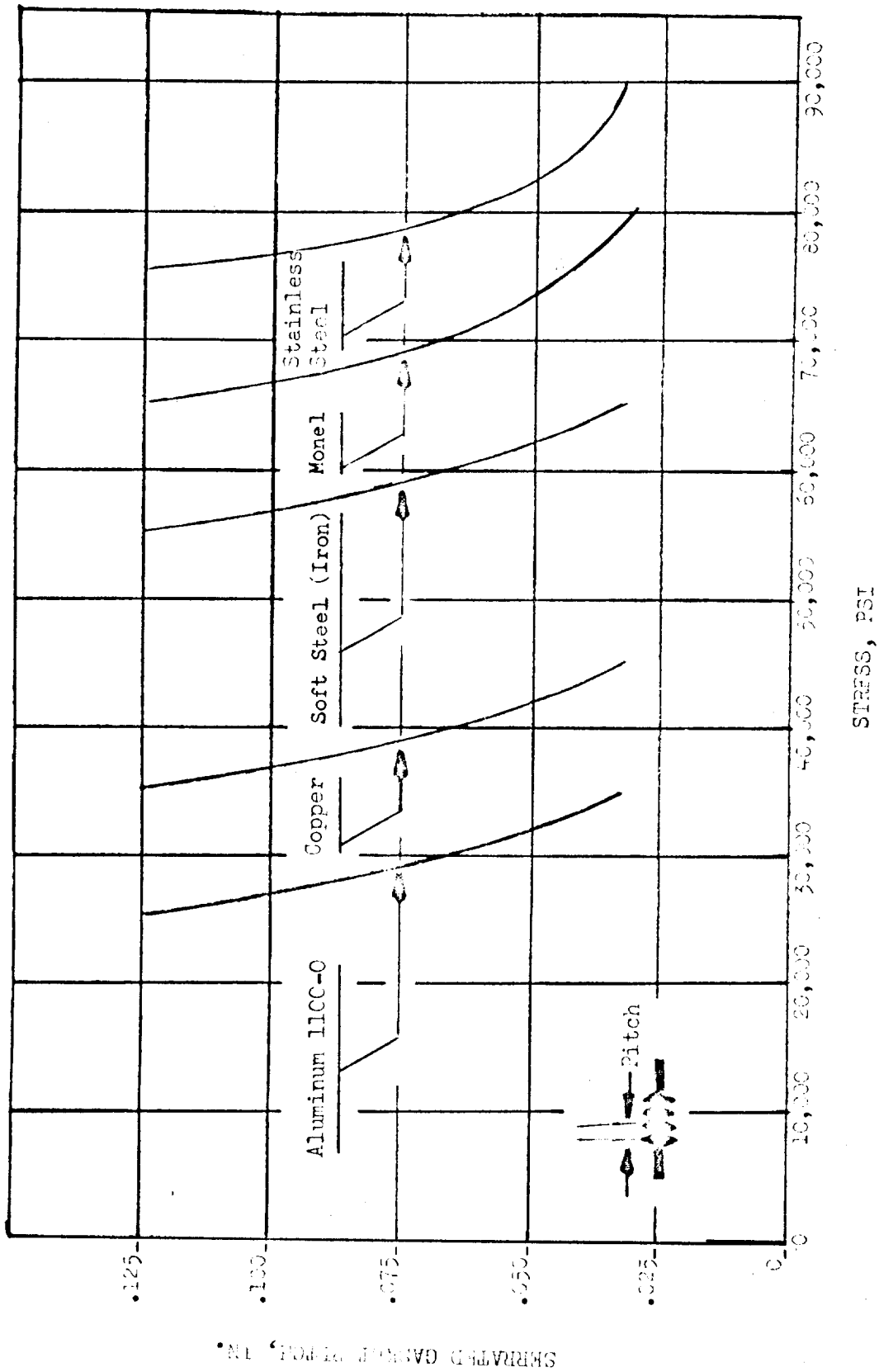


Figure 15
Minimum Required Gasket Seating Stress vs.
Pitch for Serrated Metal Gaskets

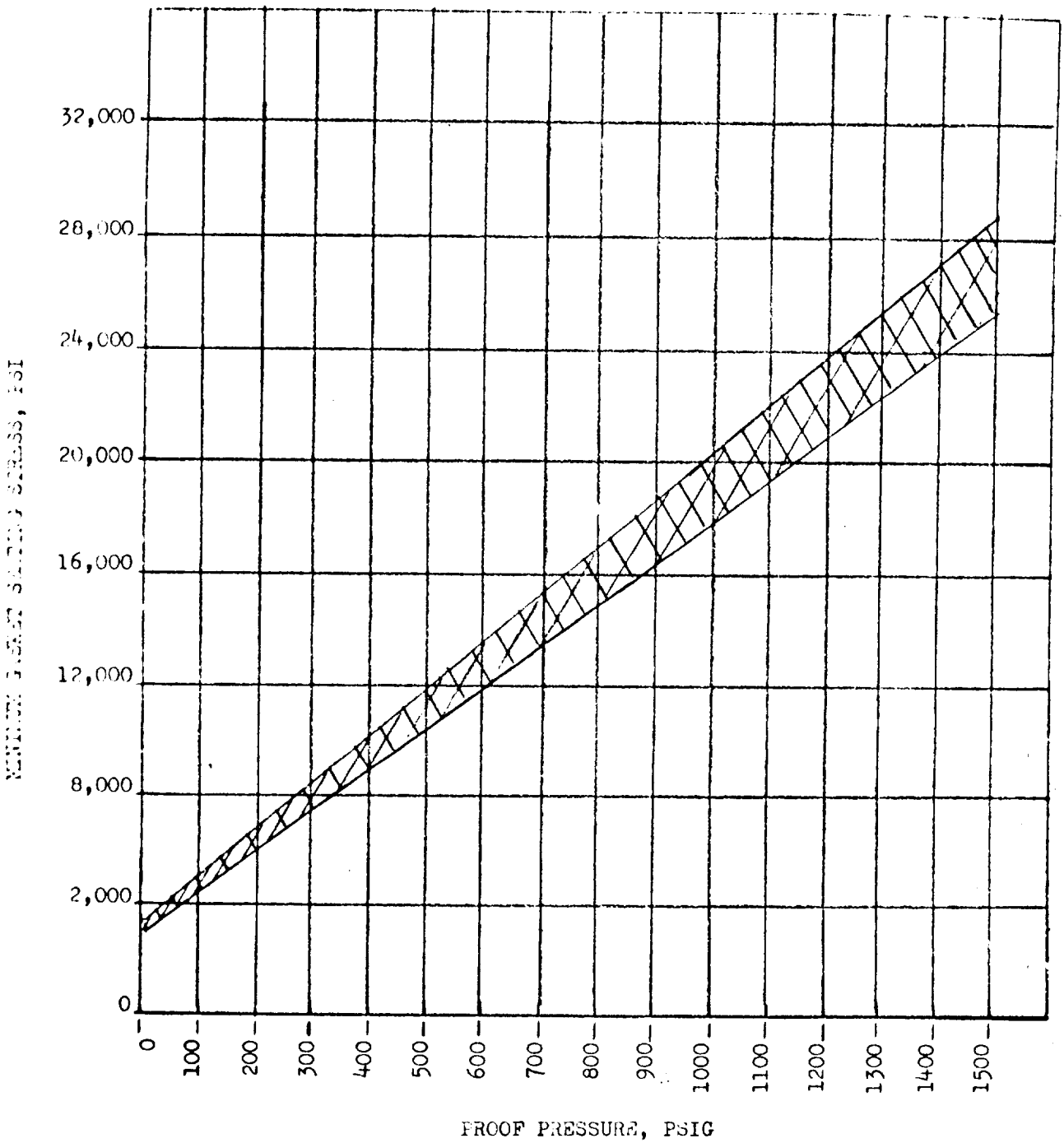
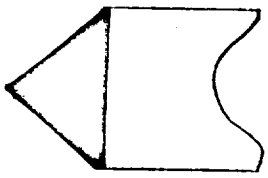


Figure 16
Typical Range of Minimum Seating Stress vs.
Proof Pressure for Spiral-Wound Gaskets

CONFIGURATION	MATERIAL	MINIMUM REQUIRED GASKET SEATING STRESS, PSI	GROOVE DESIGN
Round:	Aluminum	1300 lb/circular inch	"V" or Flat Bottomed "V"
	Copper	1300 lb/circular inch	
	Gold	1600 lb/circular inch	
	Soft Steel (Iron)	4500 lb/circular inch	
	Stainless Steel	6000 lb/circular inch	
Oval:	Aluminum	1300 lb/circular inch	"V" or Flat Bottomed "V"
	Copper	1300 lb/circular inch	
	Soft Steel	4500 lb/circular inch	
	Stainless Steel	6000 lb/circular inch	
	Monel	5000 lb/circular inch	
	Gold	1600 lb/circular inch	
Octagonal:	Aluminum	1300 lb/circular inch	"V" or Flat Bottomed "V"
	Copper	1300 lb/circular inch	
	Soft Steel	4500 lb/circular inch	
	Stainless Steel	6000 lb/circular inch	
	Monel	5000 lb/circular inch	
	Gold	1600 lb/circular inch	
Delta: 	Aluminum	Depends upon the groove and gasket angles	Special Triangular
	Copper		
	Gold		
	Soft Steel		
	Stainless Steel		

NOTE: This information from:

Dunkle, H. H., Metallic Gaskets, Johns-Manville Product Engineering - Seals Book, June 11, 1964

Figure 17
Minimum Gasket Seating Stress for Various Materials of Construction for the Solid-Section Metal Rings

TABLE IV

MINIMUM REQUIRED GASKET SEATING STRESS FOR VARIOUS MATERIALS OF CONSTRUCTION FOR FLAT GASKETS AND CRUSH WASHERS

Material	Minimum Required Gasket Stress, psi
Asbestos with Binder	Figure No. 13
Asbestos Fabric, Rubber Filled	Figure No. 14
O-Ring Materials	Figure No. 12
Teflon	Figure No. 13
Aluminum	16,000
Gold (Pure)	19,000
Copper	36,000
Soft Steel (Iron)	55,000
Monel	65,000
Stainless Steel	75,000
Corrugated Gaskets	Figure No. 15
Spiral Wound Gaskets	Figure No. 16

in the design requirements column of Figure No. 11. Because deflections and distortions in the flanges are usually combinations of the allowables shown, if any combination exceeds one of these allowables, the flanges and fastener configuration must be changed to hold the deflections within the indicated limits. The allowable axial deflection is a function of the springback capabilities (percentage of return to original height when loading is removed), pressure effects, material strength, resiliency, creep, and cold flow. The elastically-deformed gaskets (elastomeric O-rings and flat gaskets) are usually limited by extrusion or tensile strength while the metal gaskets are limited by springback and pressure effects. The allowable circular (bowing) deflection is a function of circular stiffness of the gasket, pressure effects, and resiliency. The elastomeric gaskets have almost equal capabilities in circular and axial deflections, but the spring-loaded Teflon jacketed and metal gaskets are limited by the circular stiffness of the gasket springs or legs. Allowable differential radial deflections are a function of the forces encountered tending to cause sliding between the sealing surfaces and thus disrupting (shearing) the intimate sealing interface contact when the two mating

surfaces move radially with respect to each other. If the radial stiffness of the gasket is relatively high (as with crush washers, most of the metal pressure-energized gaskets, and some of the diametral gaskets), the allowable differential radial deflections will be small. If the radial stiffness of the gasket is relatively low (as with elastomeric O-rings and flat gaskets), the allowable differential radial deflections will be larger.

The machining requirements affect the loading requirements and allowable deflection capabilities of the gasket. The machining requirements shown on Figure No. 11 are necessary to obtain initial sealing and provide the allowable deflection capabilities shown. With proper surface finish (and lay), surface flatness, groove depth tolerance, and diametral tolerances for the pressure energized gaskets, the necessary initial sealing stresses are obtained without excessively stressing any other part of the gasket. Also, the deflection and distortion capabilities are retained without losing the necessary intimate contact at the interface sealing surfaces. Relaxation of the machining tolerance and finish requirements can result in a complete loss of deflection and sealing capabilities.

The operating temperature range depends upon the base materials and coatings used in the gasket. The values shown on Figure No. 11 are based upon the widest temperature range obtainable from the materials available for the particular gasket. If the operating and environmental temperatures for an application are within the values shown, the materials selected for the gasket must be further investigated to determine the permissible temperature range. The values shown are also based upon continuous duty service at temperature. If the actual service consists of a small number of short (few seconds) durations, the temperature capabilities may be greater than those shown. However, evaluation tests should be conducted at simulated operating conditions before attempting to use the gasket in actual hardware.

Operating pressure capabilities were purposely omitted because these capabilities depend upon many factors (i.e., gasket configuration and materials, degree and type of deflections and distortions in the joints at operating conditions, permissible leak rates, and fluids to be sealed).

Radiation resistance was based upon material properties changes at a neutron flux of 1.0×10^{13} neut/cm^3-sec and a gamma flux of 3.0×10^{14} MEV/cm^3-sec. If the radiation levels are much less than those shown, or if environments will tend to prevent radiation effects upon materials properties, the gasket should be further investigated for the particular application. Also, if the radiation levels are higher than those shown, the materials selected for the gasket should be further investigated. Although information is available for most materials[11] at various levels of radiation,

(11) The Effects of Nuclear Radiation on Elastomeric and Plastic Components and Materials, Radiation Effects Center, Batelle Memorial Institute, REIC Report No. 21, September 1961 (Addendum, August 1964)

the selected materials should usually be evaluated for any particular radiation application.

The remarks column provides a brief outline of relative gasket deformation, joint deflection and distortion capabilities, as well as a "flag" of possible problem areas to be studied for particular applications. The remarks are by no means intended to be all-encompassing because other variables can become more critical for a particular application. Rather, these remarks are intended to alert the designer for further study of such items as materials, coatings, and platings as well as to compare these items with other materials and gasket configurations available for the application.

The test experience column is included to show a brief summary of results from actual laboratory testing with a particular joint design. The fluids, range of operating temperatures and pressures, as well as the resultant leakage rate are indicated.

b. Design Practices

The forces tending to separate or cause deflections in the joint are: internal proof (or burst) pressure and axial tension; bending moments; and shear loads from external forces on the joint. These forces must be known to properly design the joint. To prove that a joint can withstand the operating pressures and temperatures, the joint is subjected to a pressure above the maximum expected operating pressures with factors included to compensate for material properties at operating temperatures. The proof pressures required will depend upon the safety factors desired for a particular application, but usually are 1.3 to 2.0 times the maximum expected operating pressure plus the ratio of the relative material strengths at ambient and operating temperature. The required minimum burst pressure will also depend upon desired factors for the particular application but are approximately 1.33 times the proof pressure (the 1.33 ratio is the approximate ratio of ultimate to yield strength for most materials). The joint must be capable of withstanding the sum of the proof pressure, axial tension, bending moment, and shear loads while maintaining the minimum required sealing stresses in the gasket. If the ratio of ultimate to yield strength of any of the flange (or fastener) materials is less than the ratio of burst to proof pressure, the critical design load will be the total of burst pressure, axial tension, bending moment, and shear loads. Also, minimum permissible sealing stresses must be maintained at these loads to prevent gross leakage in the joint.

In joint designs where the "pressure-energized" face gaskets (see Sheet 1 of Figure No. 11) are used, the flanges are drawn face-to-face (the sealing stresses are predetermined by squeeze on the gasket) so that the load path by-passes the seal. Therefore, the total preload required for the joint is the greater of either the sum of the loads tending to separate the joint or the required gasket load. However, the joint must not deflect under these loads in excess of the gasket capabilities (allowable deflections).

The load path is transmitted in the joint through the flat gasket, crush washers, and solid section metal gaskets because these gaskets depend upon flange loading to obtain and maintain the required sealing stresses. The flanges and fasteners must be designed to provide sufficient load to obtain the initial sealing stresses and to maintain sufficient sealing stresses (usually, approximately 70% of the initial stresses) when all the proof pressure and external loads are applied to the joint. This means that the necessary preloading is the greater of either the required initial sealing stress load or the sum of the proof pressure, external axial tension, bending moment, and shear loads plus the load needed to maintain the minimum required sealing stresses.

The load path for some of the diametral gaskets is also through the gasket. Because the sealing stresses are applied radially and the groove-gasket configuration can allow some rotation (change in angle) of the gasket without losing the minimum required sealing stresses, the flange preloading requirements are the greater of either the initial gasket loading or the sum of the proof pressure, axial tension, bending moment, and shear loads. However, joint deflection must be within the allowable limit that the gasket can withstand. Additionally, investigation may be necessary to assure that, if some joint deflection or separation occurs, the loads applied to the joint do not excessively overload or unseat the gasket at the sealing surfaces.

Angular misalignments (parallelism) between the two mating surfaces can either prevent the attainment of the initial seal or drastically reduce the allowable deflection capabilities of the gasket. The elastomeric flat gaskets can be capable of some misalignment at assembly, providing the joint preload is increased to assure that the minimum required gasket loading is attained in all areas of the gasket for the initial seal and maintained during the life of the joint. The flanges must be brought face-to-face with the pressure energized seals (including the elastomeric O-rings). If not brought face-to-face, either the initial seal will not be attained, the deflection capabilities will be drastically reduced, or in the case of the elastomeric O-rings and Teflon jacketed gaskets, extrusion of the seal can occur. Parallelism between the mating surfaces is extremely critical with the soft metal gaskets and solid section metal rings in attaining the initial seal. Every effort should be made in the design of the joint to eliminate the possibility of angular misalignment. This may require that flexible sections and appropriate line supports be included in the design. The loading requirements for the joint must then be re-evaluated to assure that the necessary preload is available for the joint.

Radial misalignment between the mating surfaces can also be a serious problem during assembly. Within reasonable limits, flat gaskets, crush washers, and the pressure-energized gaskets are not seriously affected by radial misalignments of mating surfaces providing the widths of the prepared sealing surfaces are greater than the total of the gasket width and the expected misalignment. However, if rubbing between the gasket and flange occurs after they make contact, the mating sealing surfaces or the gasket surfaces can be damaged or galled, and a good joint will not be attained. This is a particular problem with the larger, bolted joints where the mating

surfaces can be rotated with respect to each other or moved to line up the bolt holes during assembly. The diametral seals present an even more serious problem. Because the mating surfaces are on diameters in each of the flanges, they must be properly aligned to obtain and maintain the initial sealing stresses. In all of these cases, where bolted joints are used, alignment pins (with mating alignment holes) or alignment shoulders must be provided in the design to assure that proper alignment is attained before mating with the gasket. This necessitates the provision of axial clearances between flanges during assembly and disassembly of components from the system.

7. Joint Design

The approach to designing a joint can be separated into four basic steps; gasket selection, preparation of the surfaces (or grooves), the flange design, and the fastener design. These elements are strongly interrelated, but the approach provides a somewhat logical method for preparing the preliminary design for further design analysis. The gasket material can be determined (based upon fluid effects, temperatures, pressures, and environments), the size determined, the machining requirements ascertained, and the required loading and deflection capabilities estimated. From this, the requirements for machining the mating flange surfaces can be specified (although this may affect gasket loading in some cases), the flanges can be designed (based upon a knowledge of the gasket requirements, the specified fluid, pressures, external loads, and temperatures), and the fasteners can be determined based upon the required preloads as well as the maximum spacing of the individual loading points. This approach is used in the ensuing discussion, which is completed by describing the preliminary stages of a typical design problem.

a. Gasket Selection

The first consideration in selecting the gasket is the fluid to be sealed and the allowable leakage rate. The effects of fluid exposure to the gasket materials and the permeability rates of the fluid through the gasket must be considered. Although it is always desirable to use gaskets that deform elastically, if possible, exposure to fluids or corrosive environments can cause undesirable deterioration and materials property changes (as shown on Figure No. 9) or permeability rates alone may exceed the allowable leakage rates (see Figures No. 3, No. 4, and No. 5). If the fluid (or associated environments) is a good electrolyte, it may be necessary to select the flange materials so as to reduce or eliminate galvanic cells between the flanges and the gasket (see Table II and its supporting discussion). If the fluid effects will permit the use of elastomers and plastics, the gasket material can be selected for good dielectric strengths provided that the two flange materials to be used are widely separated in the electromotive series. The system cleaning solvents or fluids that will be used should also be determined to anticipate their affects upon the gasket materials. Also, the affects of solutions of the solvents and system fluids should be ascertained. In some cases (i.e., N_2O_4 and pure water), the solutions of the solvent and system fluids may be more serious than the fluid alone.

Cold flow and creep characteristics (see Figures No. 2 and No. 7) must be considered to determine the necessary groove configuration (in the case of flat plastic gaskets) and the necessary preload (in the case of the crush washers, and other solid section gaskets).

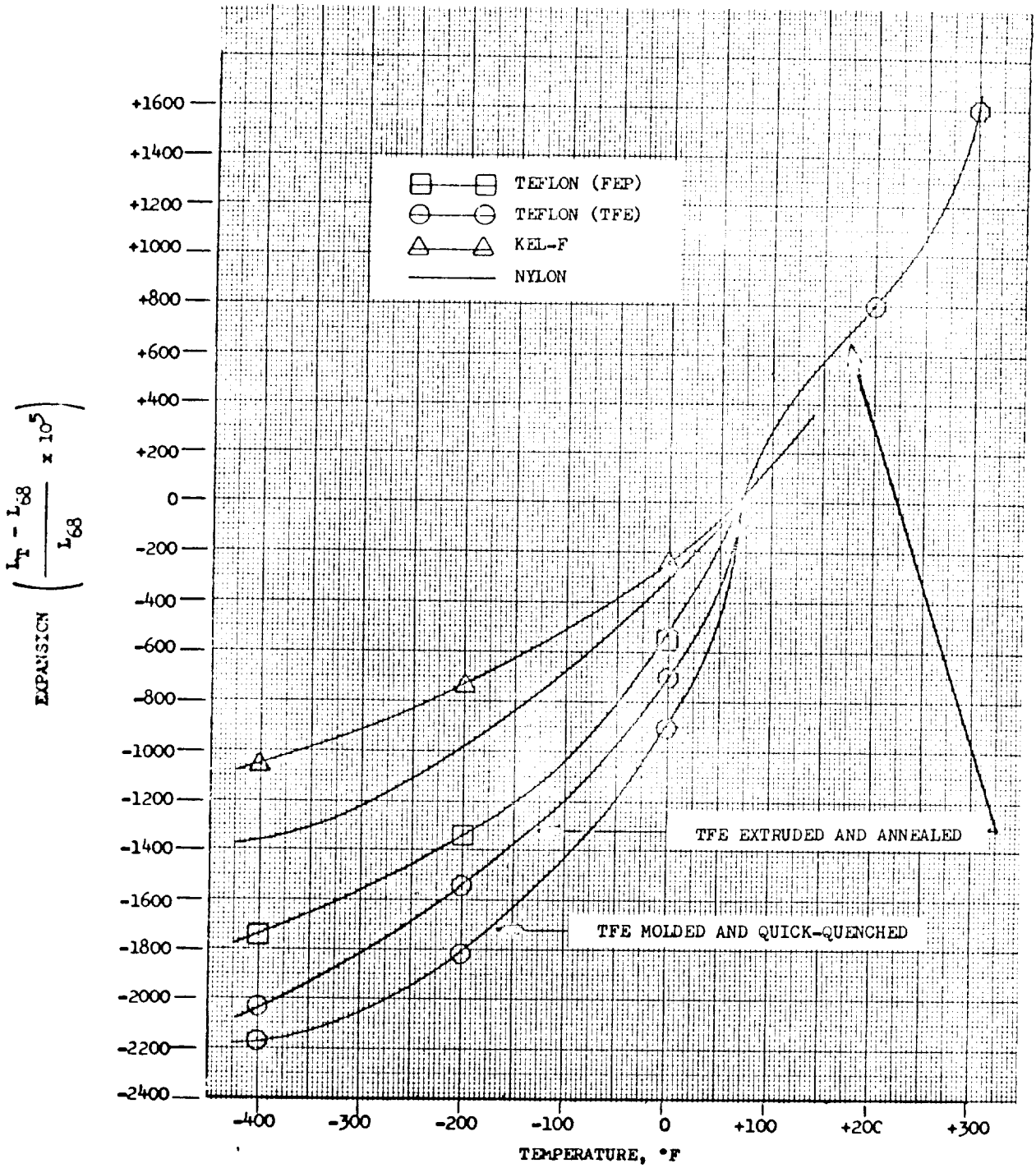
The expected temperature and temperature cycling can have a definite affect upon cold flow and creep characteristics as well as presenting compression set problems (see Figure No. 8) with the elastomers. Compression set in the elastomers is a function of time and temperature and, to some extent, the degree of compression.

The expansion-contraction characteristics of the gasket material must be known and compared with the flange expansion-contraction characteristics to determine the possibility of shear or sliding between the gasket and flange surfaces. Differential contractions between the gasket and flanges (i.e., face seal joints) can develop sufficient loads to overcome the compression load and friction at the sealing interface, which would cause the surfaces to move with respect to each other and shear the intimate sealing contact. Differential contractions (i.e., diametral seal joints) can also cause a gradual decrease in sealing stresses as temperature is reduced until the necessary intimate sealing contact is lost. Typical thermal expansion characteristics for some of the plastic materials is shown on Figure No. 18, and for some of the elastomers on Figure No. 19. Note the rapid change (slope of the curve) in expansion-contraction rates of the TFE Teflons between 50°F and 100°F, and of the silicone elastomers between -50°F and -100°F. In addition, elastomers change from a rubbery state to a "glassy" or brittle state at lower temperatures. The temperature at which this occurs is called the "glass transition temperature" or "Tg." The approximate values of Tg are also presented for the elastomers shown on Figure No. 19. The brittleness of these materials either at or below Tg is definitely not conducive to good sealing. The thermal expansion characteristics of other materials is available in existing literature(12).

If the gasket material remains elastic and soft or retains good strength and elongation properties, the differential expansion or contraction between the gasket and flanges may not be a serious problem. Teflon, as an example, expands and contracts more rapidly than the flange materials, but remains relatively soft while retaining good strength and elongation properties over a broad temperature band. If the gland and/or gasket is designed to reduce or eliminate cold flow problems, Teflon is useful over a broad temperature range. However, it is necessary to investigate other material properties (i.e., tensile strength, compression stength, and elongation) of the gasket. Typical strength and elongation properties for some of the plastics is shown on Figures No. 20 through No. 23.

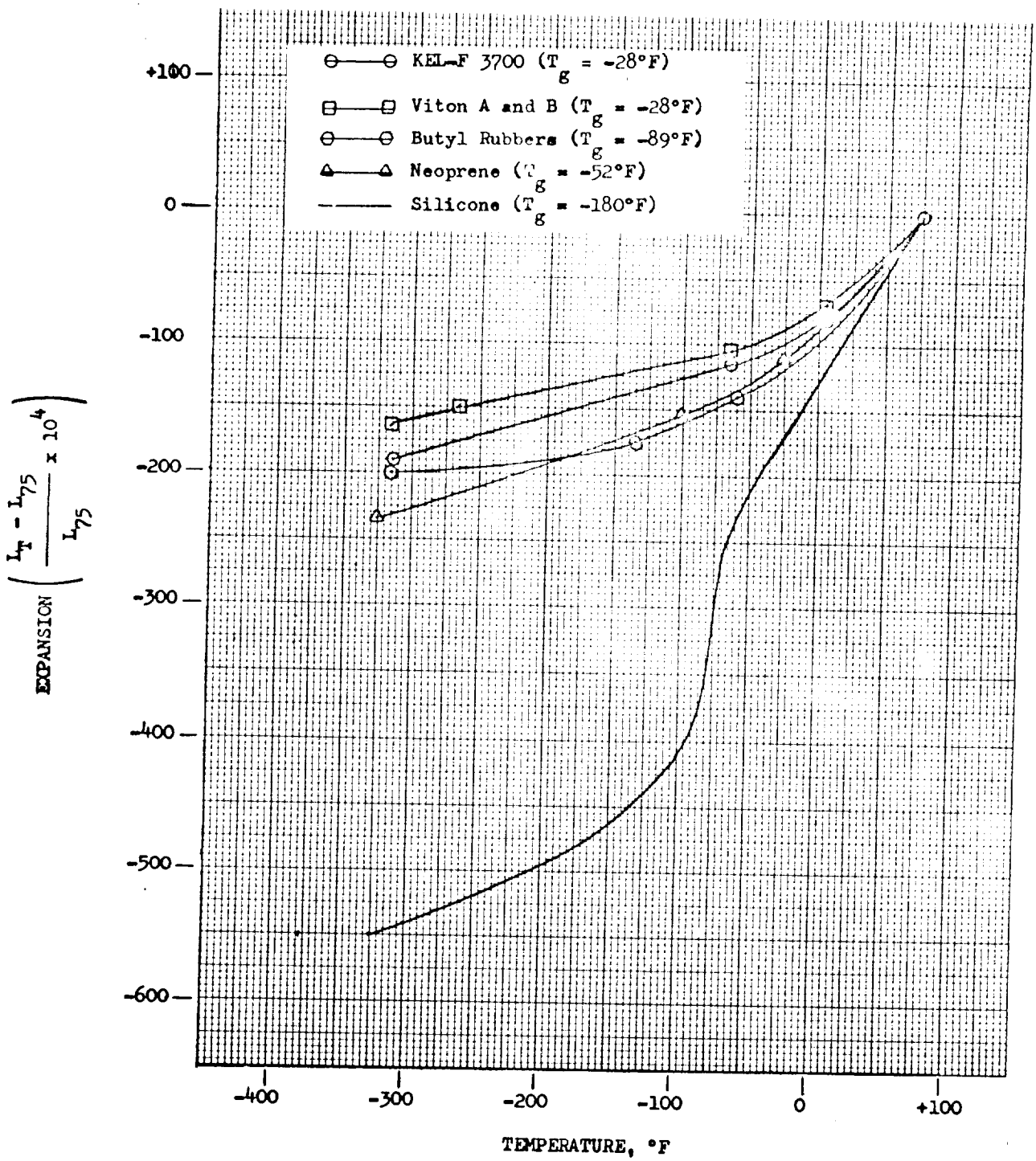
Extrusion (see Figure No. 24) of the plastic or elastomer gaskets can be the limiting factor in allowable joint deflection capabilities. Extrusion is a function of: pressure; the clearance or gap between the flange faces or diameters; the hardness of the gasket material; and in some instances,

(12) Cryogenic Materials Data Handbook, National Bureau of Standards



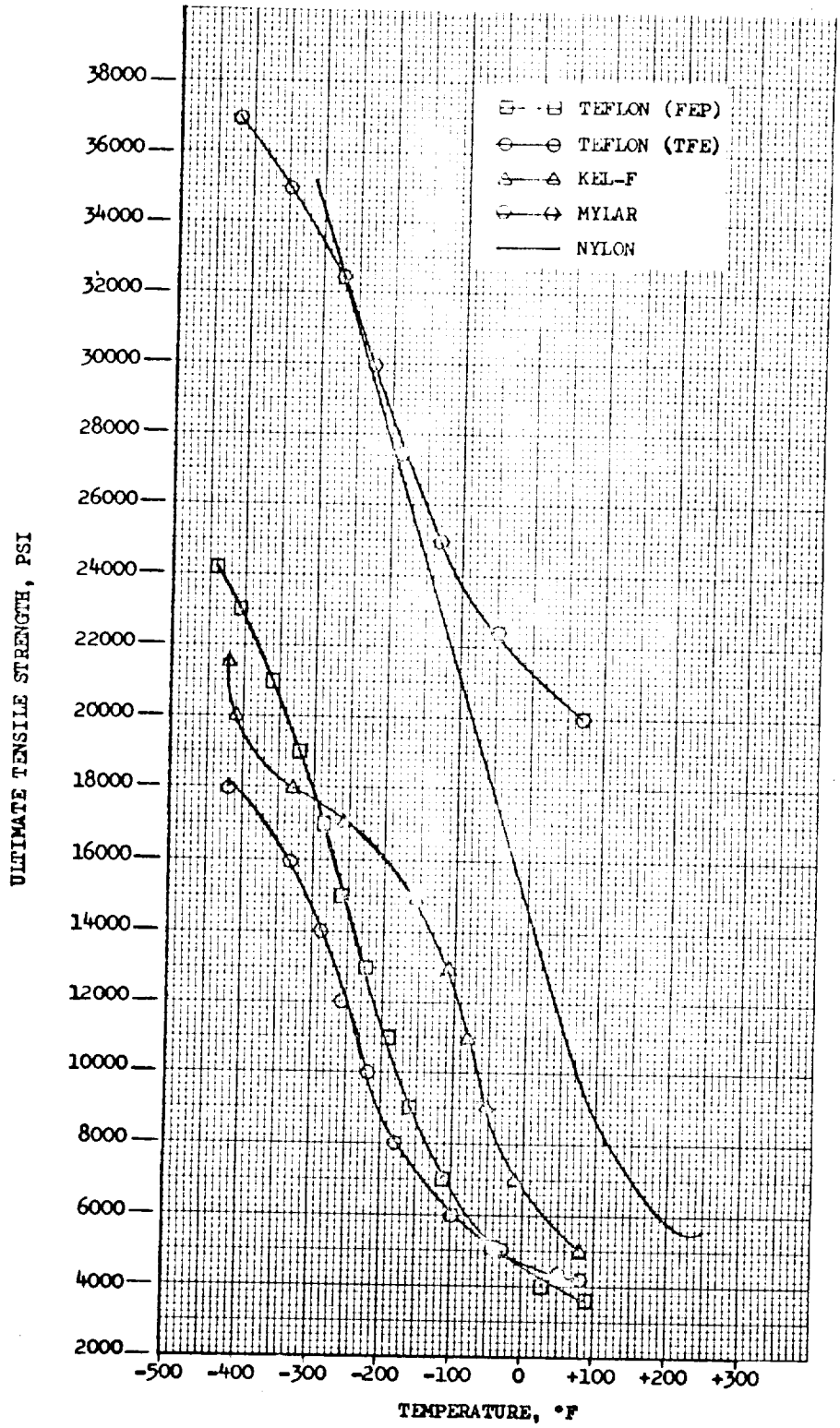
NOTE: This information from -
Cryogenic Materials Data Handbook, National Bureau of Standards

Figure 18
Thermal Expansion Characteristics of Some Plastic Materials



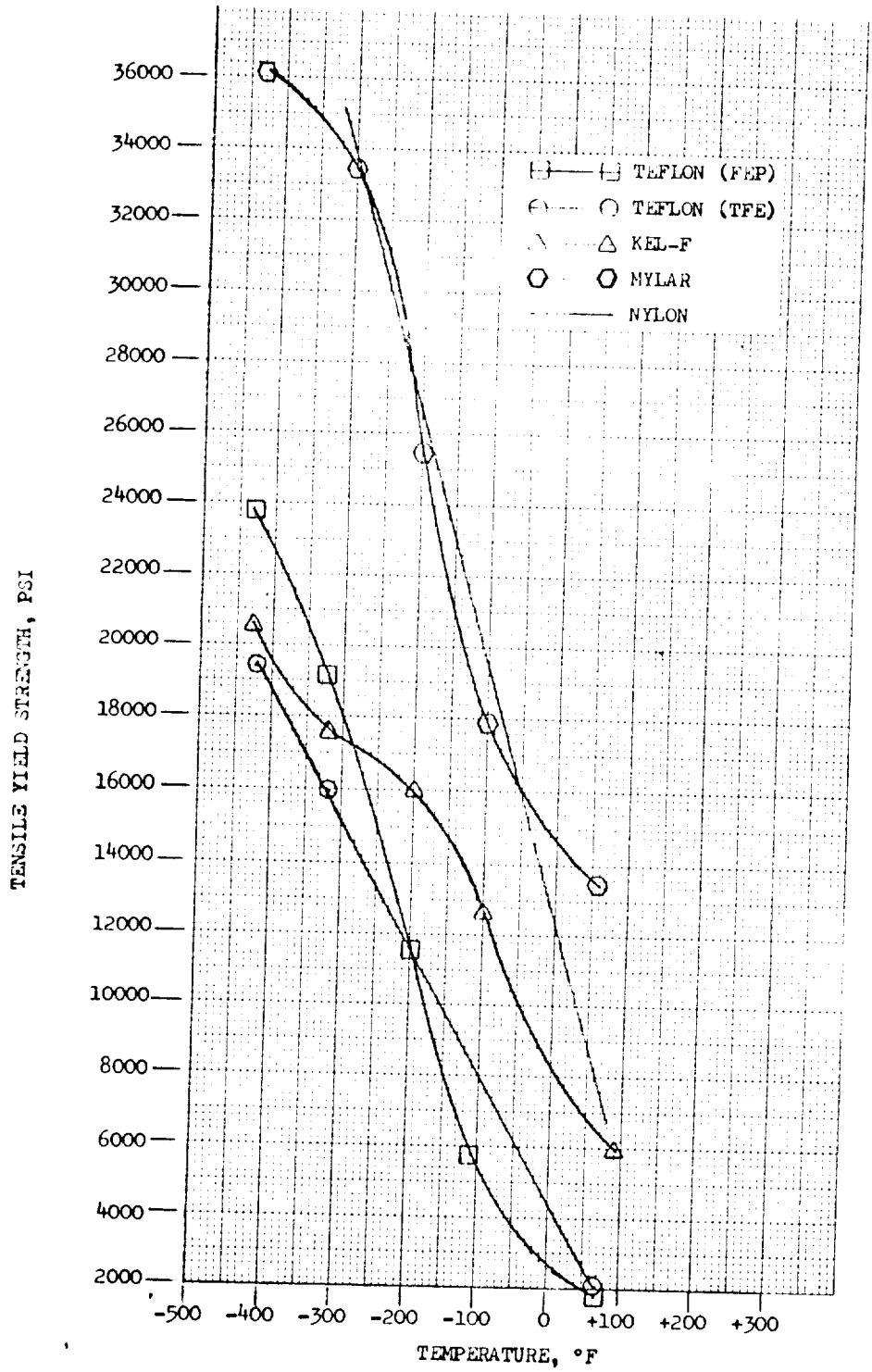
NOTE: This information from-
Elastomeric Seals and Materials at Cryogenic Temperatures, Part II, Report No. ASD-TDR-62-31, May 1963

Figure 19
Thermal Expansion Characteristics of Some Elastomers



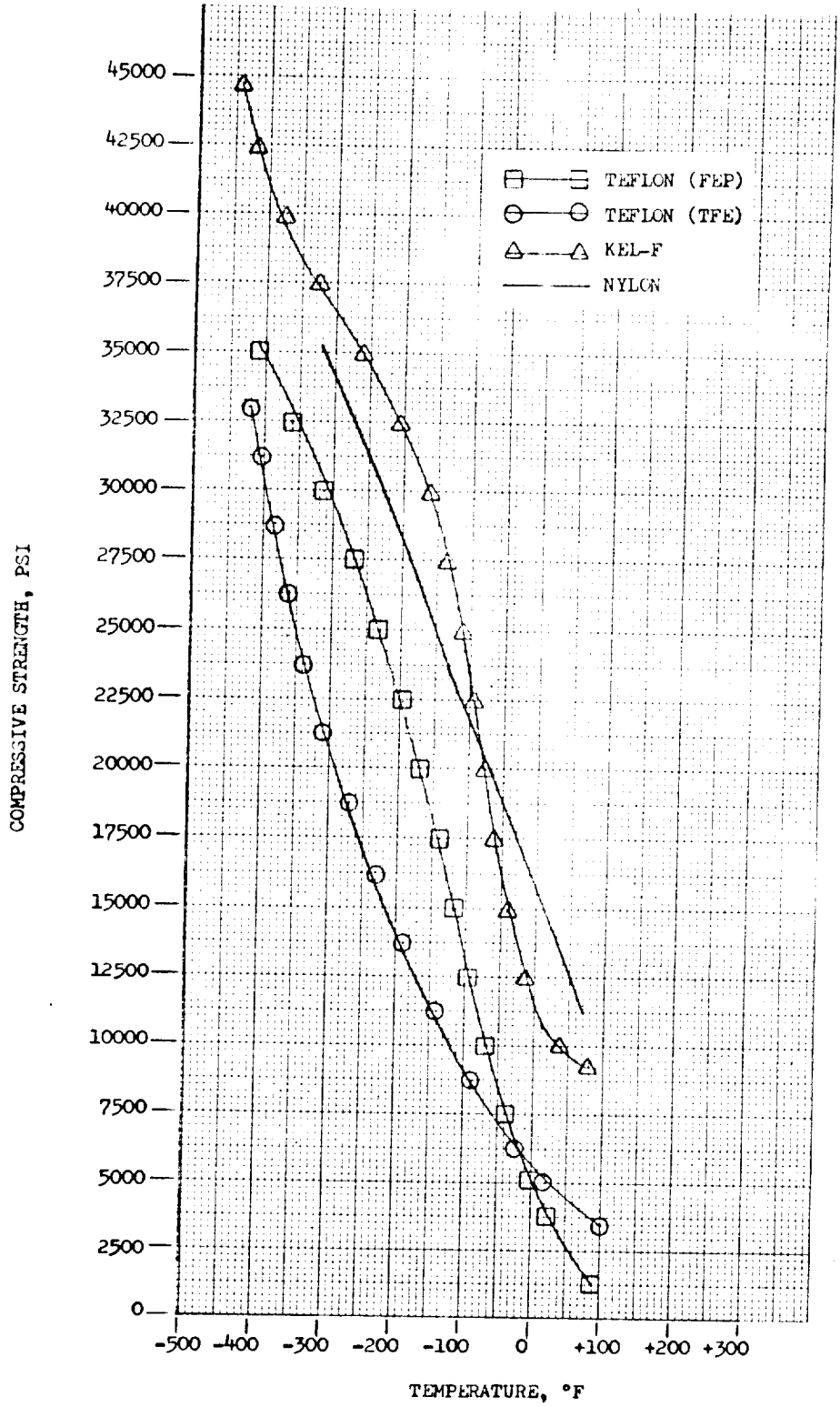
NOTE: This information from - Cryogenic Materials Data Handbook, National Bureau of Standards

Figure 20
Effect of Temperature on Ultimate Tensile Strength of Some Plastic Materials



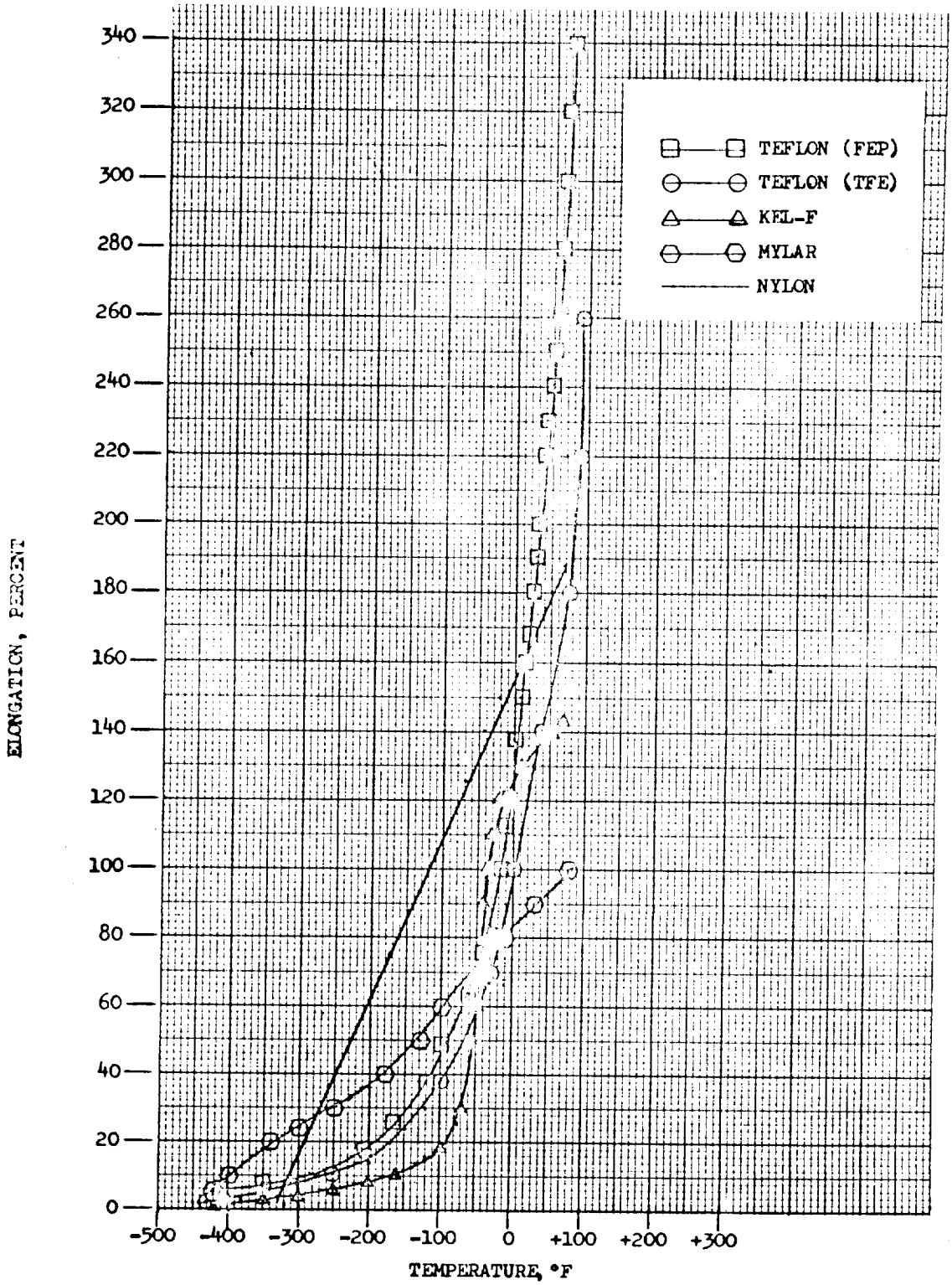
NOTE: This information from - Cryogenic Materials Data Handbook, National Bureau of Standards

Figure 21
Effect of Temperature on Tensile Yield Strength of Some Plastic Materials



NOTE: This information from -
Cryogenic Materials Data Handbook, National Bureau of Standards

Figure 22
Effect of Temperature on Compressive Strength of Some Plastic Materials



NOTE: This information from - Cryogenic Materials Data Handbook, National Bureau of Standards

Figure 23

Effect of Temperature on the Elongation of Some Plastic Materials

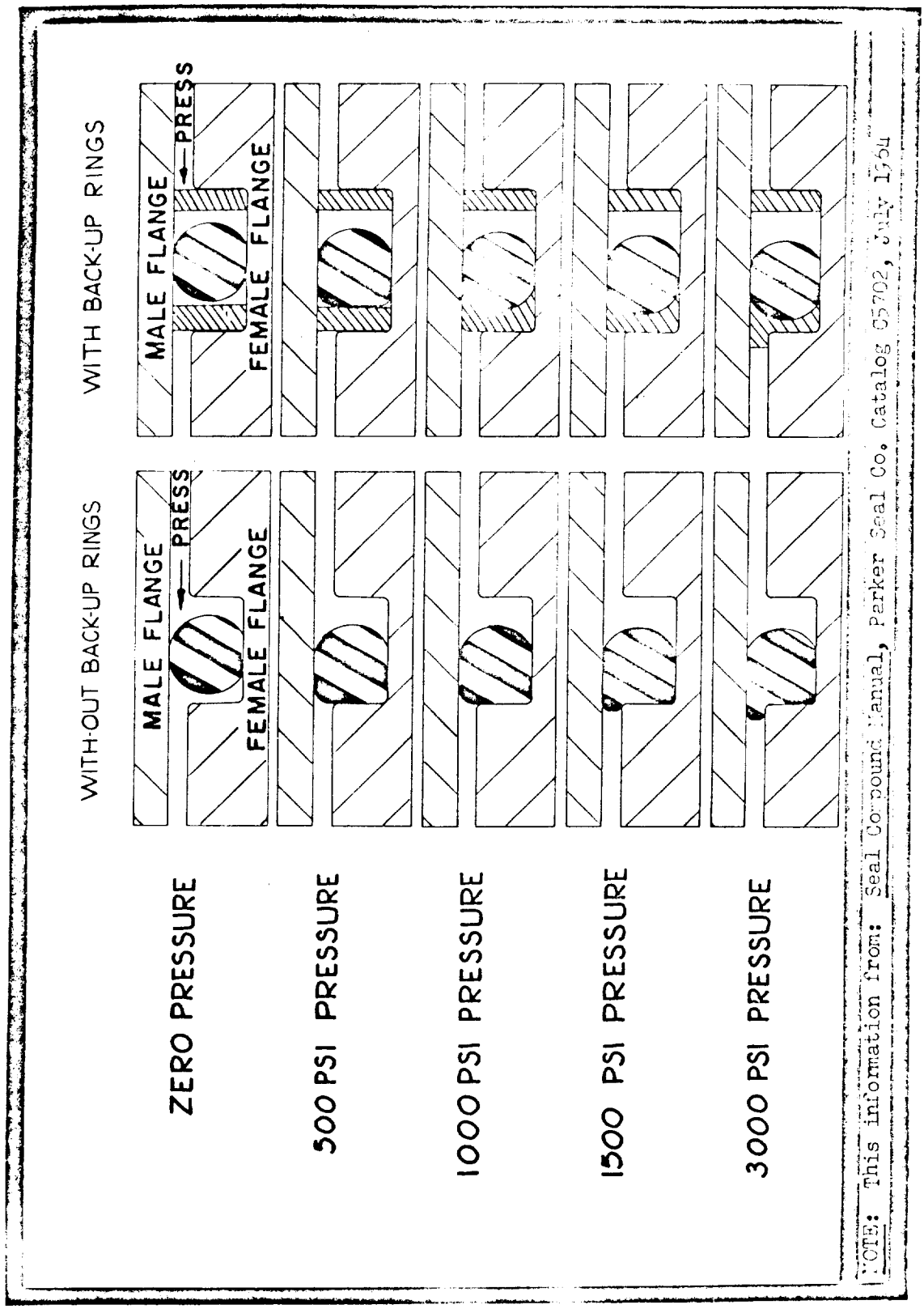


Figure 24
Extrusion of O-Rings

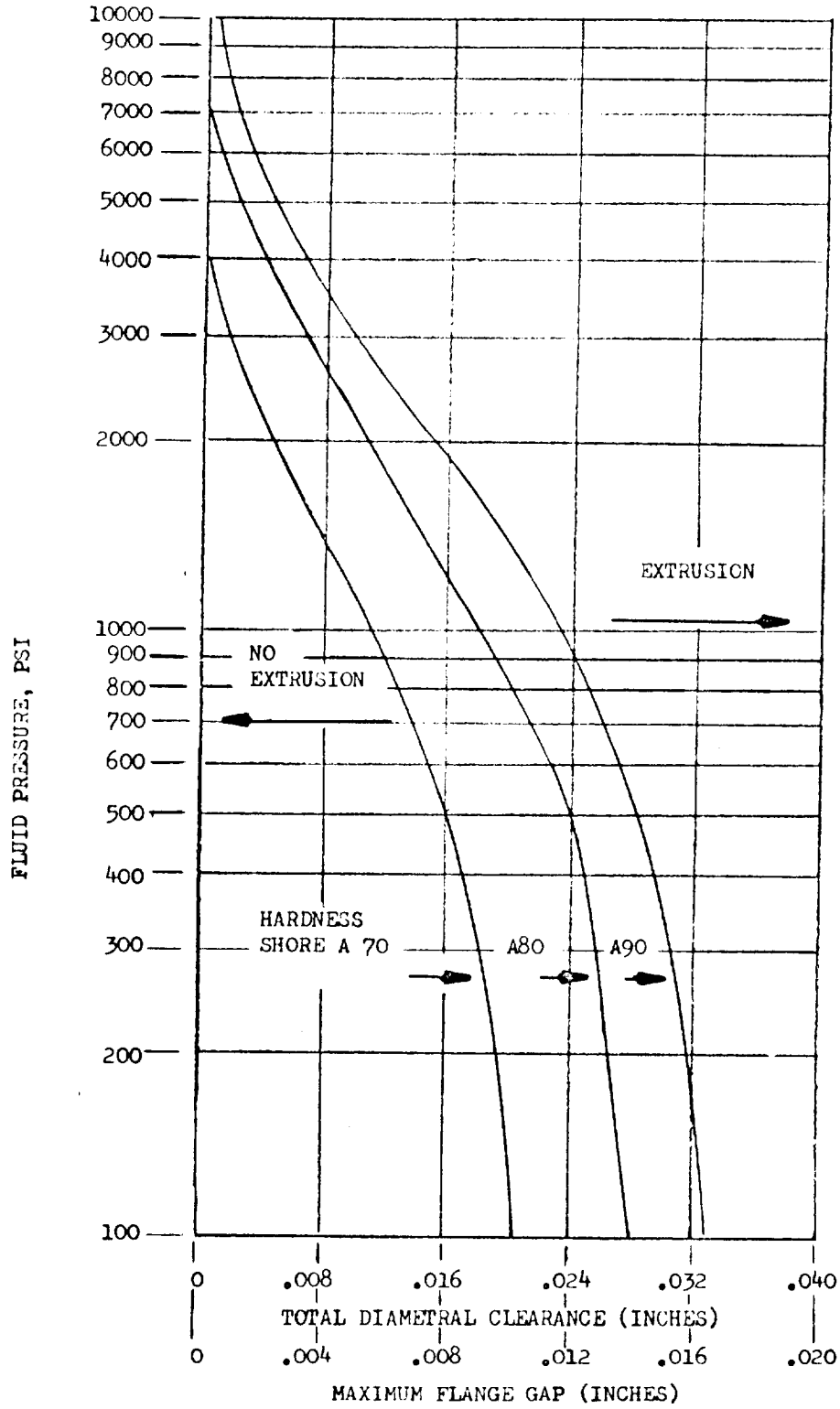
the length of time at pressure. Therefore, the extrusion characteristics of the selected gasket material should be investigated to determine the permissible gap (or clearance) between flange surfaces at proof pressure (usually conducted at room temperature). Also, because temperature affects the material hardness, the permissible gap (or clearance) should be determined at operating temperature and pressure conditions. This could be the limiting condition. Gasket extrusion is one of the most "unsafe" or undesirable joint failures because the gasket may provide a good seal up to the point of failure, then it suddenly "blows-out" and gross leakage is the result. Another form of extrusion failure, called "nibbling," can occur during pressure cycling. The gasket partially extrudes into a gap (or clearance) because of pressure effects. This extruded portion is then cut-off when the pressure is suddenly released. The condition progressively worsens until the gasket fails to seal or "blows-out." Typical extrusion characteristics for elastomeric O-rings are shown on Figure No. 25. Although some plastics can exhibit the same characteristics, there is no known literature published for this subject.

The effects of the expected vibration levels as well as the combinations of vibration and temperatures must also be investigated. The primary problems caused by vibration are fatigue and disintegration. If excessive vibration results in repeated compression and decompression of the gasket, the gasket will eventually fail because of material fatigue. The limit is dependent upon the gland design and the materials used. Disintegration can occur if the gasket material is brittle (i.e., elastomers at or below Tg) and is repeatedly overcompressed so that a portion of the gasket spalls off from time to time. This disintegration is an accelerating process because the contact area decreases as the edges spall away.

Mechanical shock and its resulting impact loads upon the gasket must also be investigated, particularly at extreme low temperatures. Impact loads at low temperature cause a plastic gasket to break or shatter in a manner similar to glass. Impact strengths for some of the plastics are shown on Figure No. 26.

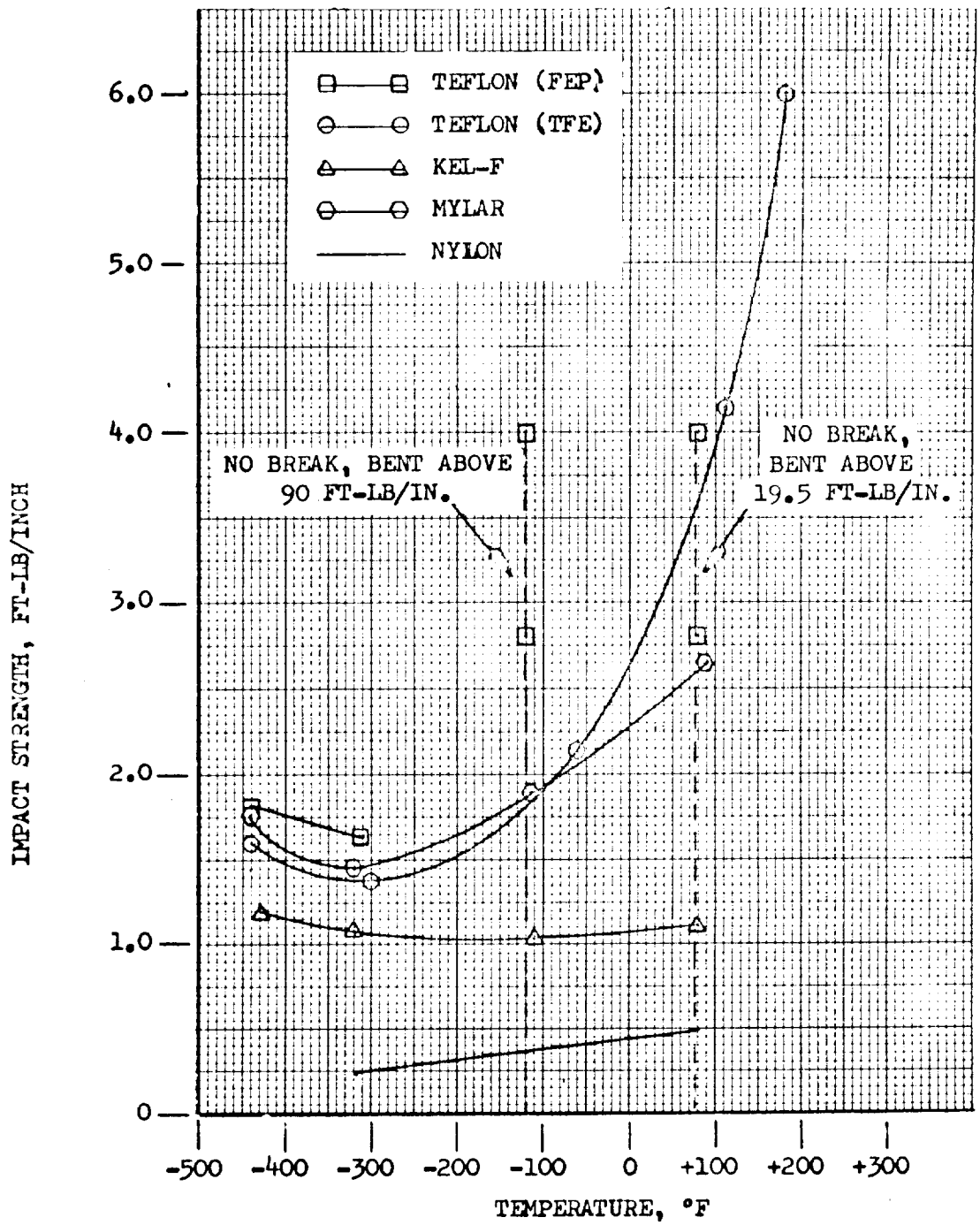
If the joint is to be used in a nuclear radiation environment, the gasket material property changes must be investigated at the expected or known radiation flux levels. The effects of radiation are basically dependent upon the radiation levels and exposure time. Investigations(13) have shown that radiation effects are also dependent upon the surrounding environment (temperature, pressure, fluid) and the fluid to be sealed. The relative radiation stability of some of the elastomers and plastics when exposed in air (normal atmosphere) is presented on Figure No. 27. A material is considered stable up to that point of exposure where no significant change occurs in any of the physical properties of the material (i.e., tensile strength and elongation). Threshold damage occurs at the point of exposure where at least one physical property of the material begins to change (the lowest point on the "mild-to-moderate" damage bar of Figure No. 27). Any deviation from the original value (increase or decrease) is considered to be damage. The lowest point on the "moderate-to-severe" damage bar is that point

(13) REIC Report No. 21, op. cit.



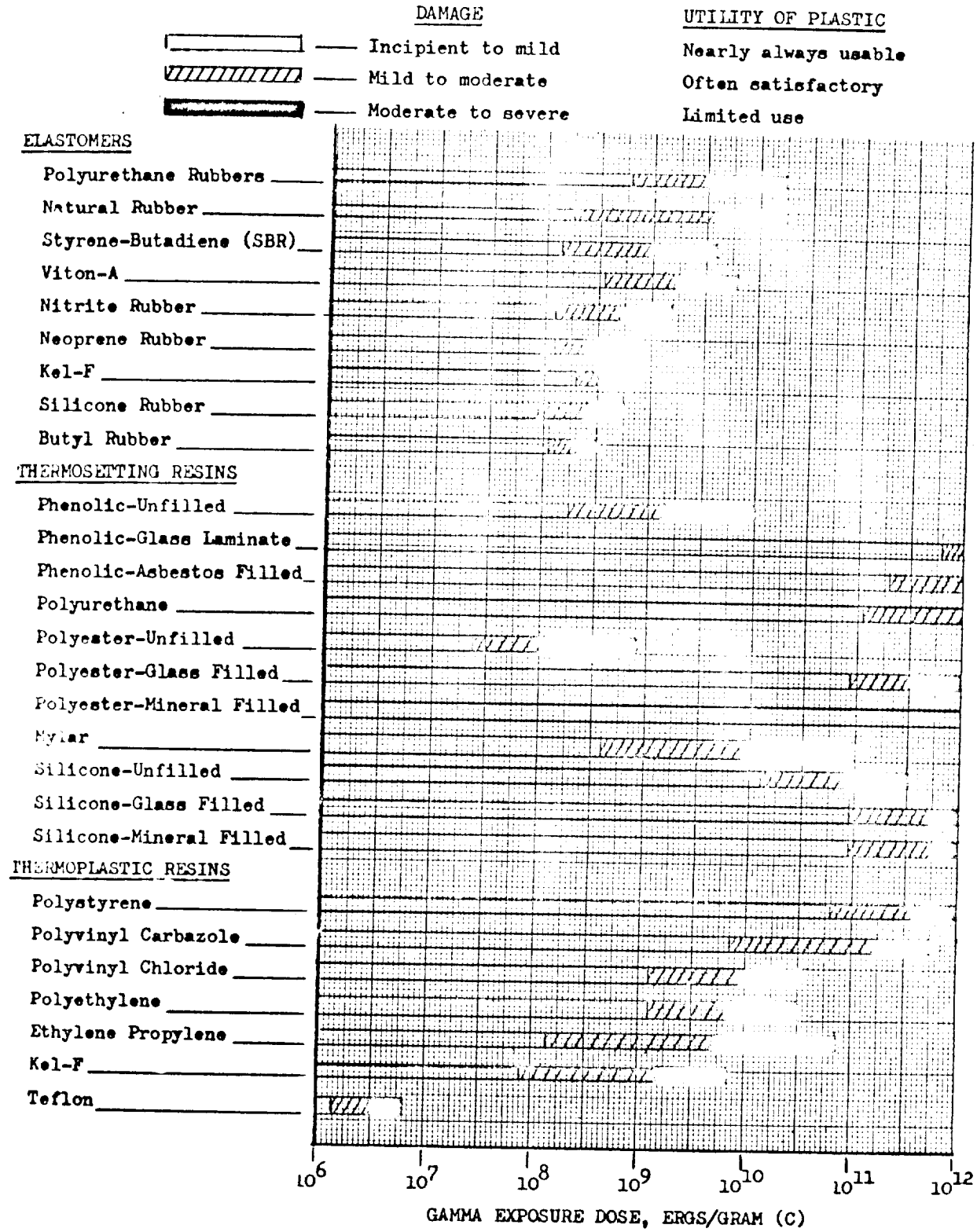
NOTE: This information from -
Seal Compound Manual, Parker Seal Co.
Catalog C5702, July 1964

Figure 25
Extrusion Characteristics of O-Rings



NOTE: This information from - Cryogenic Materials Data Handbook, National Bureau of Standards

Figure 26
Effect of Temperature on Impact Strength of Some Plastic Materials



NOTE: This information extracted from The Effects of Nuclear Radiation on Elastomeric and Plastic Components and Materials, Radiation Effects Center, Batelle Memorial Institute, REIC Report No. 21, September 1961 (Addendum, August 1964)

Figure 27
Relative Nuclear Radiation Stability of Elastomers and Plastics in Air

of exposure where a 25% (plus or minus) change occurs in at least one physical property of the material. While Figure No. 27, provides good comparisons of the effects of nuclear radiation upon the elastomers and plastics, it must be used with caution because it does not represent specific application data for seals and gaskets. Details of the effects of radiation for a particular material are available in the existing literature(14).

Generally, elastomers are less radiation resistant than the plastics(15) with the more resistant ones being the polyurethane and natural rubbers. Most elastomers increase in hardness when irradiated; however, Butyl rubber softens and becomes liquid when exposed to high doses of radiation. Most elastomers acquire a high compression set when irradiated at elevated temperatures while they are compressed. However, if they are irradiated in the free state, thereafter compression set is less than the initial value. The silicones are satisfactory for use above 300°F, but are below average in radiation resistance. Nitrile and neoprene rubbers appear to be best for use below 300°F where radiation stability is somewhat better than that of the silicone rubbers.

Generally, plastics are equal to or superior to the elastomers in radiation resistance, with the rigid types being the most resistant(16). The filled-phenolics and polyurethanes are among the most resistant thermosetting plastics; however, the hardness of these materials limits their use as gaskets or seals. Of the thermoplastics, polystyrene and polyvinyl carbazale are equally as resistant to radiation, but have much lower strength and heat resistance. The fluorine-containing plastics (Teflon, Kel-F, PVC) degrade in a radiation environment and liberate halogen acid which has corrosive effects upon adjacent hardware. Gassing of most plastics is a problem in enclosed or poorly-ventilated systems.

The importance of the effects of irradiation in conjunction with the operating environments and conditions and the system fluids cannot be overemphasized. Seals and gaskets exposed to anticipated combined environments when irradiated have often shown better service than was anticipated from the static data. Conversely, radiation resistance can be degraded when irradiated in combined environments.

The initial gasket loading requirements and deflection capabilities must also be known for the specific gasket cross-section, diameter, and materials selected. If the gasket is the nominal 1/8-in. cross-section, the loading and capabilities shown on Figure No. 11 can be used for preliminary design work, but the actual values must be determined before making the final design. If the gasket selected is of the non-proprietary type, the minimum loading requirements and capabilities for preliminary design can be determined from the values shown on Figure No. 11 providing the machining requirements specified are used. As an example, a soft aluminum gasket

(14), (15), (16) The Effects of Nuclear Radiation on Elastomeric and Plastic Components and Materials, Radiation Effects Center, Batelle Memorial Institute, REIC Report No. 21, September 1961 (Addendum, August 1964)

material has been selected and the known gasket dimensions are 1.50-in. outside diameter and 1.25-in. inside diameter. Then, from Table IV, the minimum required stress is 16,000 psi for the initial seal and 11,000 psi (70%) must be maintained at all conditions. The initial gasket load required would be:

$$W_g = A_g \times f_g$$ (See Appendix A for all symbol definitions)

where:

W_g = Load required to seat the gasket (lb)

A_g = Compression area of the gasket (in.^{2}) = $\frac{\pi}{4}(Dgo^2 - Dgi^2)$

f_g = Required gasket stress (psi)

or

$$W_g = .7854\ (1.5^2 - 1.25^2) \times 16{,}000$$

$$W_g = 8{,}640 \text{ lb}$$

The maximum allowable deflection then depends upon the difference in stress levels (initial less minimum permissible), the material modulus, and the gasket thickness;

$$\rho_a = \frac{f_g - F_g}{E} \times t_g$$

where:

ρ_a = Allowable deflection (in.)

f_g = Initial gasket stress (psi)

F_g = Minimum permissible gasket stress (psi)

E = Modulus of elasticity (psi)

t_g = Gasket thickness (in.)

or

$$\rho_a = \frac{16{,}000 - 11{,}000}{10{,}000{,}000} \times t_g$$

$$\rho_a = \frac{5{,}000}{10{,}000{,}000} \times t_g$$

$$\rho_a = .0005\ t_g$$

If the gasket thickness is .010-in., the permissible deflection at the gasket sealing surface would be .000005-in. If the gasket is .10-in. thick, the permissible deflection would be .00005-in.

These deflection capabilities are only for axial (flange separation and/or flange rotation) deflections and circular (bowing between bolts) deflections or combinations of these. They do not indicate the permissible differential radial deflection capabilities which are a function of the shear strength, modulus, elongation, and thickness of the gasket and the mating flange surface finish.

b. Groove Machining Requirements

After the gasket type and the material are selected, the machining requirements for the mating surfaces must be determined. The gasket size will dictate the space that must be allowed in the flange design. Generally, the larger the gasket cross-section, the greater the capability of the gasket to take deflections in the flanges without losing seal effectiveness. However, the larger the gasket cross-section, the higher the required initial sealing load becomes. The mating surface finish (and lay) and the surface flatness will affect the required initial sealing stresses and can be the governing factor in whether the initial seal will be attained or maintained during the life of the joint. If the gasket selected is one of the proprietary "self-energized" types, the manufacturer should be consulted to determine the necessary groove requirements. These requirements must be thoroughly analyzed regarding surface finish and lay, flatness, parallelism, diameters and tolerances, depths and tolerances, as well as other considerations to assure that all necessary requirements are specified for the groove. Any machining requirements that are omitted will have an adverse affect upon the initial sealing stresses as well as the deflection and distortion capabilities of the gasket.

If surface preparations other than those shown on Figure No. 11 are used for flat gaskets and crush washers (such as knife edges or other shear deformation configurations), the allowable deflection values can be as much as ten times greater than those calculated in the preceding section. However, this should be thoroughly investigated by actual testing before the design is made final.

c. Flange and Fastener Design

The flanges and fastener designs are interrelated. The required stiffness in the flanges to effect the initial seal, and to maintain it, is determined by the type of fasteners to be used. If it is desirable for the joint to be capable of relatively easy disconnection, a quick uncoupling device (i.e., V-couplings) can be used for the fastener. Some of these devices provide reasonably equal distribution of loads all around the flange periphery. This decreases the problem of bowing between flange loading points and to some extent, decreases the material necessary to obtain the required stiffness in the flanges. The loading circle (circle of the applied load) can also be closer to the gasket circle which decreases the flange rotation (see Figure No. 6) problem and further decreases the material required to obtain the necessary stiffness in the flange. However, sufficient stiffness must be designed into the coupling device to prevent axial separation, bowing, and related problems. Some of these devices also provide a considerable

radial force inward towards the joint centerline and can help resolve a differential radial deflection problem. Care must be used in specifying these devices because they can present some difficult design and assembly problems associated with reducing friction between the parts as well as to assure that the loading is equally distributed. Lubricants are usually necessary and assembly techniques must be developed to assure proper loading. A thorough understanding of the mechanics involved with the particular coupler to be used is necessary before the coupling and flange design can be made final. Design analyses and assembly specifications are available from some manufacturers of these devices. However, unless previous experience has been gained with the particular coupling joint to be used and the mechanics, design requirements, and assembly techniques are known, a considerable amount of development work should be conducted before the design is made final.

There are many other types of flange and fastener configurations including "break-over-center" connections, "breech-lock" or "pressure-cooker" connections, and the threaded connections. Though they are used in large diameters for some applications, they become somewhat bulky above 2.0-in. diameters. The threaded connections for small diameters will be discussed subsequently. If these devices are used, the mechanics must be thoroughly understood before the flange and fastener combination is designed to assure that they will provide sufficient loading for the gasket under all conditions during the life of the joint. Also, problems may have to be resolved to prevent rubbing or rotation between the gasket and mating surfaces, galling between fasteners and flanges, as well as excessive misalignment of parts.

The most widely-used flange and fastener design for larger diameters is the bolted connection. The size and number of bolts must be selected to provide the necessary preloading on the gasket. Also, the flanges must be designed to provide the necessary stiffness to properly preload the gasket and maintain minimum loading under all conditions during the life of the joint. Therefore, the flange and fastener design problem becomes one of a series of compromises between the weight of the joint, ease of assembly, and many other variables, while still satisfying the requirements for attaining and maintaining the seal. If a small number of <u>large</u> diameter bolts are selected to provide the necessary preload, the distance from the bolt circle to the gasket circle and the distance between bolts is large; therefore, the flanges must be stiffer (and heavier) to prevent excessive deflections in the joint which would unload the gasket and lose the seal. However, the joint would be relatively easy to assembly. If a large number of <u>small</u> bolts are selected to provide the required preload, the distance from the bolt circle to the gasket circle and the distance between bolts is smaller; therefore, the flange stiffness (and weight) can be reduced and still remain within the allowable deflections for the gasket. In this case, the assembly becomes more difficult because of the greater number of bolts. The bolt lengths and the bolt-length-to-diameter ratios can present problems of excessively unloading the gasket during operation <u>or</u> prevent repeatable preloading in the bolts during assembly. For optimum weight design, there is a "point of no return" in decreasing bolt size and increasing the number of bolts used.

Reduction in required flange weight (stiffness) beyond a certain point can be more than off-set by the increased total weight of the bolts. A comprehensive method for analysis to provide an optimum design is presented in the available literature[17]. Design analyses are presented for initial "hand" computations as well as final analysis using computer methods.

The method utilized at assembly to preload the bolts determines whether the proper loads obtained from the analysis are realized in the finished assembly. This deserves very serious consideration in the design of the joint. More than the material strength and bolt area must be considered to assure the necessary bolt loading. Friction between threads and under the bolt heads or nuts presents a serious problem in the proper loading of a bolt. Bending in the bolt caused by the bolt and nut surfaces being out of parallel increases the problem. The total amount of friction involved is determined by: bolt, nut, and flange materials; surface finish of the threads and between bolts, flange and nut surfaces; and the lubricants used. The specified machining requirements for bolt and nut surface parallelism as well as the stiffness in the flanges determines the degree of bending produced in the bolts. Several methods have been proposed for estimating or determining the actual axial bolt preloads[18][19][20] that will be attained during assembly. These include the use of tightening torque values and the measurement of elongation in the bolt.

The problem is one of transcribing bolt (or nut) torque values or bolt elongation values into axial loads and total stresses. The method for measuring bolt elongation requires preset measurement points in the bolt and reasonably assures that the axial loads are provided; however, this does not assure that the maximum total stresses in the bolt do not exceed the allowable stresses because it does not account for friction in the threads. The method using tightening torque values provides a reasonably accurate estimate of total stresses in the bolt, but cannot assure the axial loads because the friction between the threads and bolt (or nut) contact with the flanges is difficult to estimate. An acceptable and widely-used method of assembly is to specify final tightening torque values for the bolts (or nuts). To use this method effectively, the designer must be able to accurately estimate the axial loading and total combined stresses in the bolts considering the bolt, nut, and flange materials, type of threads, and the lubricants to be used. A tabulation of torque-load-stress relationships was taken from existing literature[21] and is shown on Table V. These values can be used for preliminary design work. The total combined stress values are accurate within a few percentiles because an increase in thread friction reduces the tension stresses with an accompanying increase in torsional stress so that the total combined stress is relatively unaffected. However, the axial load values can be in error by a factor

(17) Rathbun, Jr., F. O., op. cit.
(18) Korn, Arthur H., Torque-Stress-Load Relationships in Bolts and Screws, Product Engineering, November 1943
(19) Steward, W. C., Determining Bolt Tension from Torque Applied to the Nut, Machine Design, November 1955
(20) Nolt, I. G. and Smoley, E. M., Gasket Loads in Flanged Joints, Armstrong Cork Co., Machine Design, September 28, 1961
(21) Korn, Arthur H., op. cit.

TABLE V

TORQUE-STRESS-LOAD RELATIONSHIP IN BOLTS AND SCREWS

Loads and Stresses per unit torque for numbered and fractional size threads -

SIZE (No. or In.)	MAJOR DIA. (In.) (d_b)	NUMBER OF THREADS PER INCH: COURSE	FINE	EXTRA FINE	BASIC AREA OF SECTION AT ROOT OF THREADS (Sq. In.)	MAX. COMBINED TENSILE STRESS (Lb per sq. in.) IN SCREW WITH TORQUE OF 1-IN.-LB ON NUT (s_b)	MAX. AXIAL LOAD (Lb) ON SCREW WITH TORQUE OF 1-IN.-LB ON NUT (w_b)	BOLT FACTOR $B = d_b \frac{S_b}{W_b}$ (psi/lb/inch)
0	0.060		80		0.0015	72,600	74	58.86
1	0.075	64			0.0022	43,900	65	50.65
			72		0.0024	38,100	66	43.40
2	0.086	56			0.0031	26,200	55	40.27
			64		0.0034	23,400	56	35.94
3	0.099	48			0.0041	18,200	50	36.04
			56		0.0045	16,100	51	31.25
4	0.112	40			0.0050	12,600	42	27.87
			48		0.0057	10,700	43	27.90
5	0.125	40			0.0067	7,890	36	27.40
			44		0.0072	7,190	37	24.39
6	0.138	32			0.0075	6,810	34	27.64
			40		0.0087	5,620	35	22.16
8	0.164	32			0.0120	3,560	30	19.46
			36		0.0128	3,270	30	17.88
10	0.190	24			0.0145	2,660	26	19.44
			32		0.0175	2,100	27	14.78
12	0.216	24			0.0220	1,600	23	15.03
			28		0.0226	1,410	23	13.24
¼	0.250	20			0.0269	1,115	20.6	13.52
			28		0.0326	879	21.5	10.22
				32	0.0352	807	21.7	9.30
5/16	0.3125	18			0.0454	512	16.6	9.65
			24		0.0524	432	17.1	7.90
				32	0.0590	377	17.6	6.70
3/8	0.375	16			0.0678	292	14.3	7.66
			24		0.0809	261	14.9	6.57
				32	0.0890	213	15.3	5.22
7/16	0.4375	14			0.0933	179	12.2	6.42
			20		0.1090	150	12.6	5.21
				28	0.1217	132	13.0	4.44
½	0.500	13			0.1257	118	10.9	5.41
			20		0.1486	97.1	11.4	4.26
				28	0.1634	86.3	11.7	3.69
9/16	0.5625	12			0.1620	82.3	9.9	4.67
			18		0.1888	68.8	10.3	3.76
				24	0.2054	63.3	10.5	3.39
5/8	0.625	11			0.2018	59.3	8.72	4.25
			18		0.2400	48.2	9.24	3.26
				24	0.2686	44.3	9.33	2.97
3/4	0.750	10			0.3020	32.1	7.34	3.28
			16		0.3513	27.0	7.64	2.65
				20	0.3725	25.3	7.68	2.47
7/8	0.875	9			0.4193	20.4	6.52	2.74
			14		0.4805	17.6	6.78	2.27
				20	0.5200	16.1	6.92	2.04
1	1.000	8			0.5510	13.6	5.73	2.37
			14		0.6464	11.6	5.99	1.94
				20	0.6921	10.6	6.09	1.74
1-1/8	1.125	7			0.6931	9.61	5.08	2.13
			12		0.8118	8.06	5.32	1.70
				18	0.8772	7.43	5.43	1.54
1¼	1.250	7			0.8898	6.71	4.62	1.82
			12		1.0238	5.73	4.80	1.49
				18	1.0969	5.35	4.90	1.36
1-3/8	1.375	6			1.0541	5.17	4.18	1.70
			12		1.2602	4.24	4.40	1.32
1½	1.500	6			1.2938	3.84	3.86	1.49
			12		1.5212	3.31	4.16	1.19
				18	1.6101	3.17	4.32	1.10

* This information from: Korn, A. H., Torque-Stress-Load Relationships in Bolts and Screws, Product Engineering, November 1943

of as much as 90% because variations in thread friction and friction under the nut or bolt head will cause wide variations in the axial tension developed in the bolt. Therefore, unless there is previous experience with the selected bolt-flange material combination and lubricant to develop appropriate friction factors, the axial load-torque relationships must be determined by actual tests. Figure No. 28 presents a typical example of the variation in axial loads with the same lubricant and with different lubricants. The data was gained from actual tests of an 8-32 CRES screw, CRES washer, mating threads in a CRES flange, and a cast aluminum flange.

d. Typical Design Problem for Aerospace Service

A complete and final design analysis for a joint is too extensive to be included in this report. The purpose of this section is to present the basic approach to preliminary design of the joint as well as the items to be considered and the precautions to be taken in evolving an optimum design with minimum weight while maintaining a satisfactory joint. The analysis used herein is a straightforward one because its main purpose is to illustrate gasket selection methods, give a preliminary idea of flange and bolt sizes, and to provide a starting point for a complete analysis to ascertain a final joint design.

All of the symbols used throughout this report are delineated in Appendix A.

The sample problem is based upon an assumed required dissimilar material line connection for a liquid hydrogen system. The known data and requirements for the joint are as follows:

(1) System Fluid: Liquid Hydrogen

(2) Temperature: +120°F to -423°F

(3) Maximum Permissible Leakage: 1.0×10^{-6} cc/sec/inch of gasket circumference

(4) Line Size: 10-in. O.D.

(5) Pressures: Nominal Operating = 1500 psig
Maximum Operating = 1850 psig
Proof = 2250 psig
Burst = 3375 psig

(6) External Loads: Axial Tension = 12,000 lb
Bending Moment = 18,000 in.-lb

(7) Flange Materials: One Flange = 6061-T6 Aluminum
Opposite Flange = 347 CRES

(8) A secondary (or dual) seal arrangement is also required for all joints in the system for two purposes; to prove that the

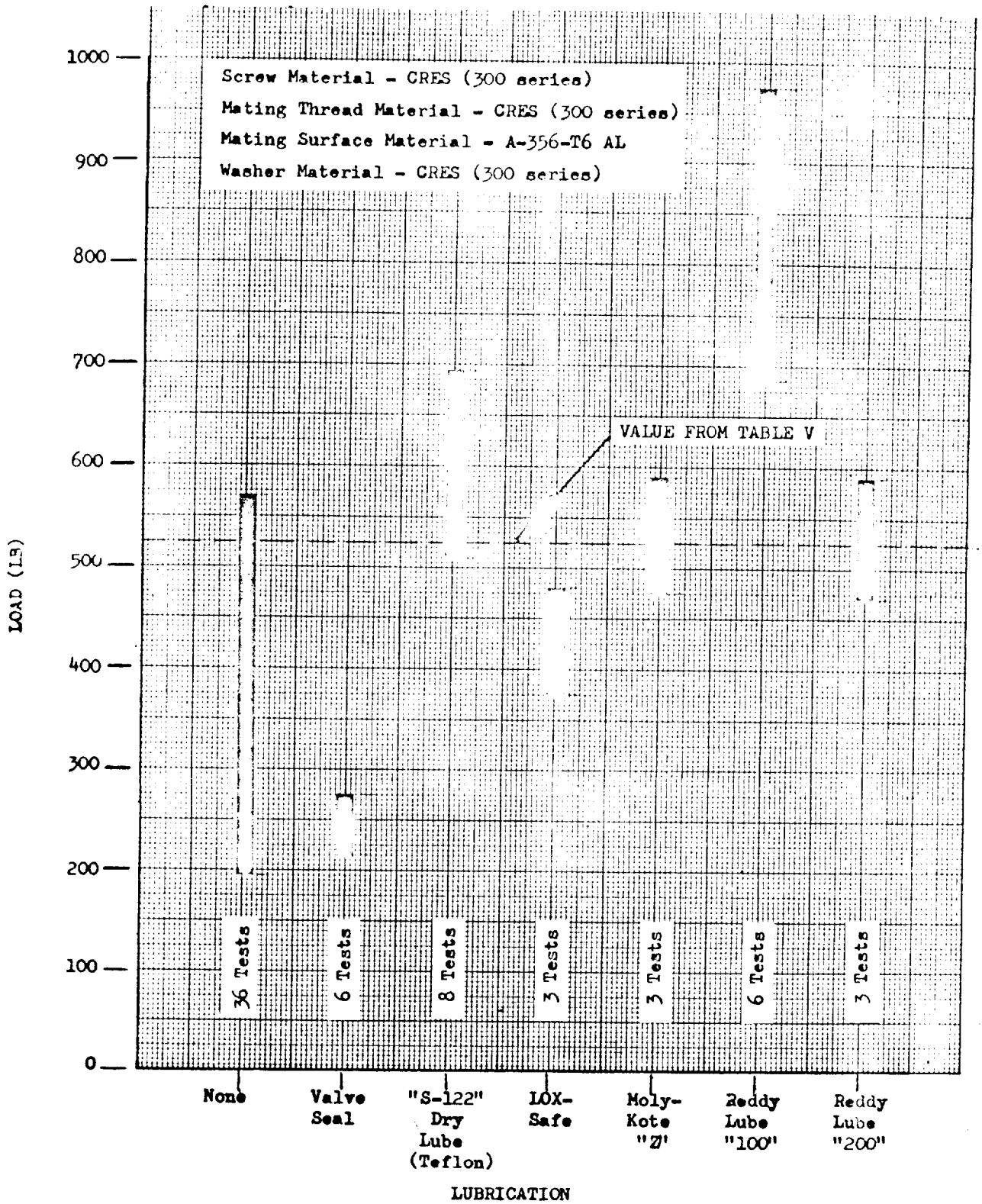


Figure 28
Variation of Load with Various Lubricants on a No. 8-32
Coarse Thread Screw at 10 lb-in. Torque

primary seal meets the leakage requirements and to provide a collector for leakage past the primary seal. The inter-seal cavity is vented overboard to a safe area.

In this case, it will be assumed that the selected flange materials are compatible with the system, no electrolytic corrosion problems exist between the materials, no problems of notch sensitivity, creep, or other factors exist with the flange materials in the system fluids and at the temperatures specified. However, any condition that may affect strength, ductility, or the machined condition of the flanges (i.e., creep, material instability resulting from thermal cycling, and surface corrosion) must be analyzed to assure that these conditions will not affect the leakage capabilities or reliability of the joint.

The gasket type and material(s) are selected based upon their capacity for meeting the leakage requirements, compatibility with the system fluid, temperature effects upon the materials, and compatibility with the mating flange materials (includes galvanic corrosion effects and gasket/flange surface hardness relationships). In addition, where a selection of gaskets is still available considering these requirements, deflection and distortion capabilities of the gasket at the design loads specified are considered. The design for this particular joint requires that the secondary seal must also be a good positive seal at all operating conditions because the only reliable methods of leak detection for the leak rates specified is the mass spectrometer which requires the inter-seal cavity to be evacuated to .10 or .20 microns. Although there are several possible ways to attain this requirement, they are not discussed in this report. Assume that the gaskets selected are of the Conoseal ® design. The dual Conoseal design has been used in similar applications with good success when all the necessary requirements for the gaskets are met in the design of the other components for the joint. The disadvantages are that most of the "dual-seal" joint concepts increase the diameter of the flange, which increases the weight of the joint, and the interactions between the two glands which can cause an overall reduction in the deflection and distortion capabilities of the gasket(s).

The next steps are to determine the mating flange and gasket hardnesses (gasket softer than either of the two flanges) and galvanic corrosion problems between flanges or between the flanges and the gasket. The hardness of the annealed 347 CRES flange is approximately 145 on the Brinell Scale and the hardness of the 6061-T6 flange (if properly processed during machining and heat treatment) is approximately 100 on the Brinell Scale. In this case, 6061-T4 aluminum gasket with a hardness of approximately 70 on the Brinell Scale could be used. Galvanic corrosion between bare aluminum and the stainless steel can be a very serious problem. However, this can be reduced or eliminated by applying a chromate conversion coating to the gasket and the aluminum groove surfaces. Anodized groove surfaces and gaskets are not recommended as this process causes surface porosity and results in leakage, which is greater than the allowable.

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The preliminary design layout now follows. This entails the determination of gasket space requirements, gasket loading requirements, flange loading and resultant stresses, and fastener requirements. The basic flange diameters are determined beginning with the tubing size (in this case, 10.00-in. O.D.), laying in the groove requirements for the gasket, calculating required flange loading, and determining the number of fasteners required as well as their sizes. The preliminary flange design is then built around these requirements. The tube wall thickness will be assumed to be 0.50-in. for the preliminary design layout. Hoop stresses in the tube are approximately 32,000 psi at burst pressure and 22,000 psi at proof pressure, which is reasonable for the 6061-T6 part of the joint. The Conoseal gasket is a diametral seal; therefore, pressure and temperature changes must not cause excessive changes in squeeze (difference between male and female sealing diameters) on the gasket. If changes do occur, it is preferable to increase squeeze (a tighter seal) rather than decrease squeeze. The determination of which flange should be the male or female half of the joint is also important. The difference in expansion (caused by pressure) and contraction (caused by temperature) on the free (unrestrained) tube sections can be used as the basis for this selection in the preliminary layout.

Changes in tube diameter caused by proof pressure:

$$\Delta D_p = \frac{f_t D_t}{E} \quad \text{(see Appendix A for all symbols)}$$

For the 347 CRES Tube; $f_t \cong 22{,}000$ psi and $E = 29 \times 10^6$ psi.

Therefore,

$$\Delta D_{P_{347}} = \frac{(22\text{x}10^3)\ (10)}{29\text{x}10^6}$$

$$\Delta D_{P_{347}} = .76 \times 10^{-2} = \underline{\underline{.0076\text{-in.}}} \text{ expansion}$$

For the 6061-T6 Tube; $f_t \cong 22{,}000$ psi and $E = 10 \times 10^6$ psi.

Therefore,

$$\Delta D_{P_{Al}} = \frac{(22\text{x}10^3)\ (10)}{10\text{x}10^6} = \frac{22\text{x}10^4}{10\text{x}10^6} = 2.2\text{x}10^{-2}$$

$$\Delta D_{P_{Al}} = \underline{\underline{.022\text{-in.}}} \text{ expansion}$$

The aluminum expands with pressure at a greater rate than the 347 CRES. The difference in the change is .022-in. - .0076-in. or .0144-in.

Changes in tube diameters caused by temperature:

$$\Delta D_T = D_{68} \times K_T \text{ (see Appendix A for all symbols)}$$

For the 347 CRES Tube; $K_T = -3.0 \times 10^{-3}$ [22].

Therefore,

$$\Delta D_{T_{347}} = 10 \times (-3.0 \times 10^{-3})$$

$$= \underline{\underline{-.030\text{-in.}}} \text{ contraction}$$

For the 6061-T6 Aluminum Tube; $K_T = -4.2 \times 10^{-3}$ [23].

Therefore,

$$\Delta D_{Tal} = 10 \times (-42. \times 10^{-3})$$

$$= \underline{\underline{-.042\text{-in.}}} \text{ contraction}$$

The aluminum contracts with temperature changes at a greater rate than the 347 CRES. The difference is .042-in. - .030-in. or .012-in.

The difference in the change in the unrestrained diameters caused by temperature changes and pressures is approximately equal. The selection of male/female flange materials becomes more complicated. If, in the system application, full operating pressure is applied to the joint before temperatures stabilize, it may be more desirable to make the female flange from the 347 CRES and the male flange from the 6061-T6 aluminum. However, if the system is "bled-in" at very low pressures before full operating pressure is applied, it may be more desirable to make the female flange from the 6061-T6 aluminum and the male flange from the 347 CRES. Assuming that the latter condition exists with this joint, the preliminary design will provide for a female 6061-T6 aluminum flange and a male 347 CRES flange. The "stepped" (.040x.275 C/S) dual seal concept will be used. The inside diameter of the primary (inner) gasket groove is 9.56-in. and the outside diameter of the secondary (outer) gasket groove is 10.59-in. The minimum diameter to the inner edge of the bolt clearance holes (D_C) is 10.90-in. to provide sufficient strength to prevent local distortion in the groove surface at the bolt clearance holes during loading of the gasket. Thus, all bolt clearance holes must be outside the 10.90-in. diameter.

After ascertaining the groove requirements for the preliminary layout, the number of bolts and bolt sizes must be determined. The bolt requirements are determined from the flange preloading requirements, which, in turn, is determined from the known separating forces (or design loads) at

(22) Cryogenic Materials Data Handbook, op. cit.
(23) ibid.

the joint. Also, the initial loads required to properly seat the gasket must be known. If flat crush gaskets were used, the initial preload (which provides the initial sealing stresses) and the allowable reduction in preload (from the allowable reduction in sealing stresses) before leakage occurs would have to be determined. The bolt/flange preloads in the gasket would have to be sufficient to maintain the allowable reduced preload when all separating forces (pressure, axial tension and bending moment) are applied to the joint. With the Conoseal gasket, the primary concern is to prevent excessive separation of the flanges at the groove because the sealing stresses will remain constant if there is no separation or other changes in the preloaded joint when the design loads are applied. Therefore, flange and bolt preloads must be based upon the greater of either gasket preloading requirements or design loads. The total load required to initially seat the gasket is:

$$W_g = \pi (D_p + D_s) w_g \quad \text{(See Appendix A for all symbols)}$$

The required gasket load intensity (w_g) for initially seating the Conoseal gasket is approximately 500 lb/in. of circumference for the 6061-T4 gaskets.

Therefore,

$$W_g = \pi (9.83 + 10.33) (500)$$

$$= \underline{\underline{31,700 \text{ lb}}}$$

The total design loads are the sum of the separating forces caused by proof pressure, axial tension, and bending moment. These are calculated as follows:

<u>Load Caused by Proof Pressure</u>:

$$W_p = \frac{P \pi (D_p)^2}{4}$$

$$W_p = \frac{2250 \pi (9.83)^2}{4}$$

$$= \underline{\underline{171,000 \text{ lb}}}$$

<u>The load applied from external axial tension is known</u>:

$$W_A = \underline{12,000 \text{ lb}}$$

<u>The load caused by external bending moment is</u>:

$$W_B = \frac{4M}{D_{bc}}$$

The bolt circle diameter (D_{bc}) is not accurately known at this point because the bolt size must be determined from the design loads. The tube thickness can be added to the minimum flange clearance diameter (Dc) to gain a first estimate of the bolt circle diameter or:

$$D_{bc} = Dc + t$$

$$D_{bc} = 10.90 + .50 = \underline{\underline{11.40\text{-in.}}}$$

Therefore:

$$W_B = \frac{4\ (18,000)}{11.40}$$

$$= \underline{\underline{6320\ \text{lb}}}$$

This load must be recalculated after the actual bolt size and bolt circle diameter has been determined.

<u>The total design load is</u>:

$$W_T = W_P = W_A + W_B$$

$$W_T = 171,000 + 12,000 + 6320$$

$$= \underline{\underline{189,320\ \text{lb}}}$$

Thus, the flange and fastener design must be based upon the total design load (W_T) because it is larger than the required gasket preload (Wg).

The size and number of bolts required for the joint is a function of the stiffness of the flanges, the permissible combined stress in the selected bolt material, the axial (or tension) load that can be attained in the bolt at the permissible stress value, and many other variables. Selection of the size and number of bolts upon the basis of the cross-sectional area and tension stress in the bolt is not satisfactory because either the required axial preload will not be attained during tightening or the bolt must be over-stressed during tightening to attain the necessary preload. Therefore, the torque-stress-load relationship must be known for the bolt, the bolt and flange materials, and the lubricants to be used before an accurate estimate can be made of the axial tension load attainable in the bolt. The size and number of bolts affects the flange design because the flanges must become stiffer (and heavier) as the bolt spacing increases (fewer, but larger diameter bolts) so as to prevent excessive deflection (bowing, see Figure No. 6) of the flanges between the bolts and the loss of the gasket preload. Larger diameter bolts enlarge the bolt circle diameter (and flange O.D.) while increasing the moments causing rotation of the flange section. Conversely, the flange stiffness (and weight) can be reduced as the bolt spacing decreases (more, but smaller diameter bolts). Smaller diameter bolts minimize the bolt circle

diameter (and flange O.D.) as well as the moments causing rotation of the flange section. A reasonable preliminary estimate of the size and number of bolts required can be gained from the bolt factor (B). The derivation of this factor is detailed in Appendix B. The bolt factor is estimated from:

$$B = \frac{.942\ D_{bc}\ F_b}{W_T} \quad \text{(See Appendix A for all symbols)}$$

Assuming the permissible combined stress in the bolt is 50,000 psi,

$$B = \frac{.942\ (11.40)\ (50{,}000)}{189{,}320}$$

$$= \underline{\underline{2.835\ \frac{\text{psi}}{\text{lb/in.}}}}$$

The bolt selected should have a bolt factor equal to or greater than the estimated bolt factor to maintain the minimum bolt circle diameter. All of the standard thread 5/8-in. bolts will satisfy the requirement (see Table V). The minimum number of bolts required is calculated from:

$$n = \frac{W_T\ B}{.9\ F_b\ d_b}$$

Where:

B = the actual bolt factor from Table V.

The minimum number of 5/8-in. bolts required for each of the thread types is:

	5/8-11	5/8-18	5/8-24
Number of bolts required (n)	29	22	21

The efficiency of the bolt increases as the thread pitch decreases for the same size bolt; therefore, fewer bolts are required to obtain the same total load. In this case, the 5/8-18 bolts will be used and the number of bolts will be increased to 24 to account for unknown (at this point) stresses caused by bending in the bolt. The actual bolt circle diameter will be 10.900 + .625 or 11.525-in. The load on the joint caused by external bending moment (W_B) and the total design load (W_T) should be recalculated using the actual bolt circle diameter.

C-2

$$W_B = \frac{4\ (18{,}000)}{11.525}$$

$$= \underline{\underline{6260\ \text{lb}}}$$

$$\therefore\ W_T = 171{,}000 + 12{,}000 + 6260$$

$$= \underline{\underline{189{,}260\ \text{lb}}}$$

The stress in the bolt at burst pressure design loads must also be considered. This stress must then be compared with the ultimate strength in the bolt. Usually, the joint must withstand these loads without any part of the joint failing, but permanent deformations are permissible that would render the joint unusable. In some cases, it may be required that the joint withstand burst design loads without gross leakage. In this instance, it will be necessary to determine the capabilities of the gasket to follow separations or other deformations in the grooves before gross leakage is encountered. This requirement could then become the limiting design criterion for the flanges and fasteners rather than the proof design loads previously discussed.

The next step in the preliminary design procedure is to determine the approximate thickness of the two flanges. The grooves required for the gasket must first be "layed-in" and the flanges built around these requirements. The nominal groove dimensions for the Conoseal joint is shown on the preliminary layout of Figure No. 29. These dimensions will be used for the preliminary design. However, the supplier must be consulted for all the actual groove dimensions and requirements for any of the proprietary gasket designs.

For the flange analysis, it is more useful to transcribe joint design loads into load intensities (lb/in. circumference). For preliminary analysis, the separating forces (design loads) are assumed to be acting upon the mean tube diameter and the clamping forces (bolt preloading) are assumed to be acting at the bolt circle diameter.

Design (separating) load intensities:

$$w_p = \frac{P\ (D_p)^2}{2\ (D_i + D_t)} \quad \text{(See Appendix A for all symbols)}$$

$$= \frac{2250\ (9.83)^2}{2\ (9.00 + 10.00)}$$

$$= \underline{\underline{57.17\ \text{lb/in.}}}$$

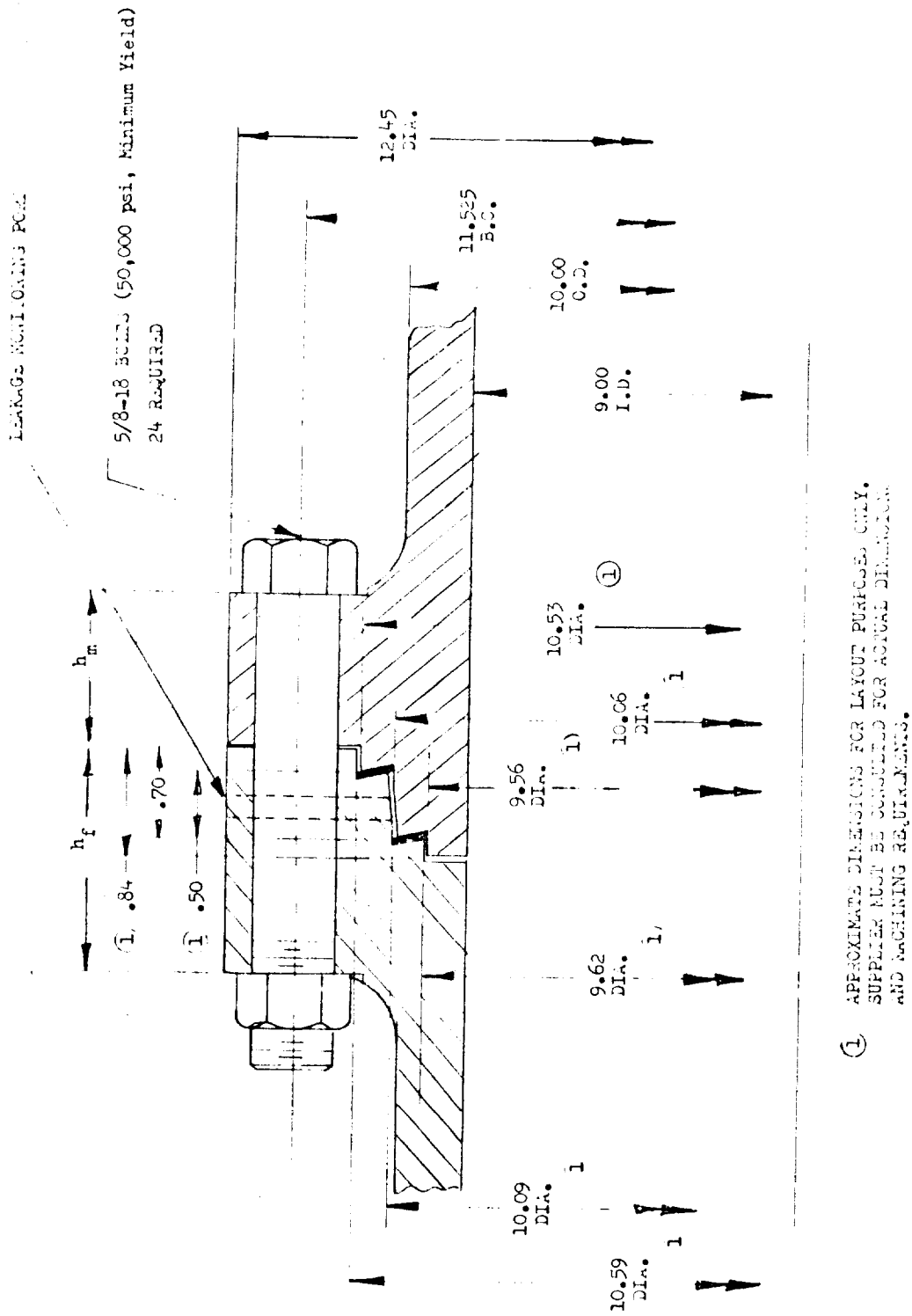


Figure 29
Preliminary Joint Design Layout

$$w_A = \frac{2\ w_A}{\pi\ (D_i + D_t)}$$

$$= \frac{2\ (12{,}000)}{\pi\ (9.00 + 10.00)}$$

$$= \underline{\underline{402\ \text{lb/in.}}}$$

$$w_B = \frac{4\ M}{\pi\ \frac{(D_i + D_t)^2}{2}}$$

$$= \frac{4\ (18{,}000)}{\pi\ \frac{(9.00 + 10.00)^2}{2}}$$

$$= \underline{\underline{127\ \text{lb/in.}}}$$

$$w_T = w_p + w_A = w_B$$

$$w_T = 5{,}717 + 402 + 127$$

$$= \underline{\underline{6246\ \text{lb/in.}}}$$

Preliminary calculations for the flange thickness can be made using the following assumptions:(24)(25)

Each flange is a flat circular ring and the flange thickness necessary for the grooves is disregarded (in this case, a very conservative assumption, but much less conservative than for most face seal applications).

Poissons Ratio = .30 (approximately correct for most materials).

No fixity at the bolted edge of the flange.

The loads resisting design loads (fastener loads) are equally distributed at the bolt circle diameter. (They are actually concentrated loads at each bolt contact point.)

(24) Timoshenko, S., Strength of Materials, Part II, Advanced Theory and Problems, D. Van Nostrand Co., Inc.

(25) Effect of Deflection Under Load on Face Seal Performance, Technical Information Bulletin No. 1015, Harrison Manufacturing Company, Burbank, California

The flange thickness must satisfy two requirements; the maximum stresses must not exceed the allowable stress for the flange material and the flange deflection at the gasket sealing surfaces must not exceed the capabilities of the gasket. If it is assumed in the preliminary estimate that no significant bending rigidity exists in the tube (i.e., the tube wall is thin with respect to the flange), the minimum flange thickness required to satisfy the permissible stress can be calculated from:

$$h_{min_F} = \sqrt{\frac{1.5\ w'_T\ (D_{bc} - D_i - t)\ (D_i + D_F)}{Fa\ D_i\ Log_e\ D_F/D_i}}$$ (See Appendix A for all symbols)

and the minimum flange thickness required to satisfy the permissible deflection (gasket capabilities) can be calculated from:

$$h_{min} = \sqrt[3]{\frac{.75\ w_T\ (D_{bc} - D_i - t)\ (D_i + D_F)\ (D_F - D_{gi})}{\delta\ a\ E\ Log_e\ D_F/D_i}}$$

From the preliminary joint design layout (Figure No. 29) and previous calculations:

w_T = 6373 lb/in.

D_{bc} = 11.525 in.

D_i = 9.00 in.

t = .50 in.

D_F = 12.45 in.

D_{gi} = 9.56 in.

E for 347 CRES (Male Flange) = 28×10^6 psi

E for 6061-T6 Alum (Female Flange) = 10×10^6 psi

F_a for 347 CRES (Male Flange) = 30,000 psi

F_a for 6061-T6 Alum (Female Flange) = 40,000 psi

The total allowable deflection (separation) at the gasket sealing surface for both flanges is .004-in. maximum, or .002-in. maximum for each flange.

The thickness required for the male (347 CRES) flange to satisfy the maximum permissible stress is:

$$h_{min_F} = \sqrt{\frac{1.5(6373)\ (11.525 - 9.00 - .50)\ (9.00 + 12.45)}{(30{,}000\ (9.00)\ \mathrm{Log}_e \frac{12.45}{9.0}}}$$

$$= \sqrt{\frac{1.5(6373)\ (2.025)\ (21.45)}{30{,}000\ (9.00)\ (.3245)}}$$

$$= \sqrt{4.74} = \underline{\underline{2.18\text{-in.}}}$$

The thickness required for the male flange to satisfy permissible deflection (.002-in. maximum) is:

$$h_{min_\delta} = \sqrt[3]{\frac{.75(6373)\ (11.525 - 9.00 - .50)\ (9.00 + 12.45)\ (12.45 - 9.56)}{(.002)\ (28 \times 10^6)\ (.3245)}}$$

$$= \sqrt[3]{33.08} = \underline{\underline{3.21\text{-in.}}}$$

The thickness required for the female (6061-T6 Aluminum) flange to satisfy the permissible stress is:

$$h_{min_F} = \sqrt{\frac{1.5\ (6373)\ (2.025)\ (21.45)}{(40{,}000\ (9.0)\ (.3245)}}$$

$$= \sqrt{3.55} = \underline{\underline{1.85\text{-in.}}}$$

The thickness required for the female flange to satisfy the permissible deflection is:

$$h_{min_\delta} = \sqrt[3]{\frac{.75\ (6373)\ (2.025)\ (21.45)\ (2.89)}{(2.0\text{x}10^{-3})\ (10\text{x}10^6)\ (.3245)}}$$

$$= \sqrt[3]{92.4} = \underline{\underline{4.52\ \text{in.}}}$$

The limiting requirements for both the male (347 CRES) and the female (6061-T6 Alum.) flange is the permissible deflection rather than permissible stress. This is usually the case even when there is considerable bending rigidity afforded by the tubing wall (i.e., the tube wall provides additional stiffness to the flange). It is particularly true of flange materials with low modulus and high strength, such as aluminum.

Where significant bending rigidity exists with the tube wall, a simple calculation for flange thickness is not possible. Any additional stiffness afforded by the tubing tends to lower the stresses and deflections. The following equations can be used to estimate the stresses and deflections where significant bending rigidity exists with the tube.

The maximum stress is:

$$f_{t\ (max)} = \frac{6\ mo}{t^2}$$

where:

$$mo = \frac{5\ w_T\ (D_{bc} - D_i - t)}{1 + \frac{.642h}{\sqrt{.5\ D_i t}} + \frac{.709\ \sqrt{5\ D_i t}}{D_i} \left(\frac{h}{t}\right)^3 (\mathrm{Log}_e\ D_F/D_i)}$$

and the maximum deflection is:

$$\delta\ (max) = \frac{354\ f_{t\ (max)}\ (D_F - D_i)}{E \sqrt{\frac{2t}{D_i + t}}}$$

Because the only unknown at this stage of the preliminary design is the thickness of the two flanges, the equations can be simplified for rapid calculations of stress with various flange thicknesses using the previous estimates as starting points.

In this particular design, the equations for stress reduces to:

$$m_o = \frac{6460}{1 + 428h + .0384\ (2h)^3}$$

$$f_{t(max)} = 24\ m_o$$

The deflection for each of the two flange materials at the indicated stress values is estimated from the simplified equation:

$$\delta_{(max)} = 3.76 \frac{f_{t\ (max)}}{E}$$

The thickness and stress values can then be plotted against deflection to determine the minimal thickness of each flange that will satisfy the permissible deflection. The values were calculated for this particular design and are shown on Figure No. 30. From this figure, it is indicated that the thicknesses required to satisfy the permissible deflection are:

$$\underline{\underline{h_f = 4.42 \text{ in.}}}$$

$$\underline{\underline{h_m = 3.00 \text{ in.}}}$$

The maximum stress in the 3.00-in. male (347 CRES) flange is approximately 16,000 psi, and the maximum stress in the 4.42-in. female (6061-T6 Aluminum) flange is approximately 5280 psi.

A full and detailed analysis must be made after the preliminary design is completed. Flange dimensions may require adjustment, the number of bolts and bolt sizes may require changing, and the seal grooves may have to be relocated dependent upon the results of the analysis. This analysis must include the additional stresses and deflections caused by bending (bowing) between the bolts around the periphery of the flanges. The maximum stress and deflection occurs at the inside diameter of the flange midway between the bolts. Also, the effects of mechanical shock and vibration as well as other considerations (i.e., temperature changes) must be included in the analysis.

A good method for detailed analysis and a good preliminary design method are available in the existing literature.(26) The method used is an iterative process, which is useful for computer work. It also provides detailed analyses for many types of flange (joint) configurations.

8. Handling Procedures and Problems

After the joint design problem has been satisfactorily resolved and all of the machining requirements are specified, the next problem becomes one of properly handling the component parts of the joint up to the time that they are assembled. Handling of the flanges and gaskets prior to assembly affords many possibilities for damage to critical sealing surfaces, particularly if these surfaces are exposed. A certain amount of handling is necessary to inspect, clean, store, and assemble parts after fabrication. The acuteness of the handling problem depends upon the materials involved as well

(26) Rathbun, Jr., F. O., op. cit.

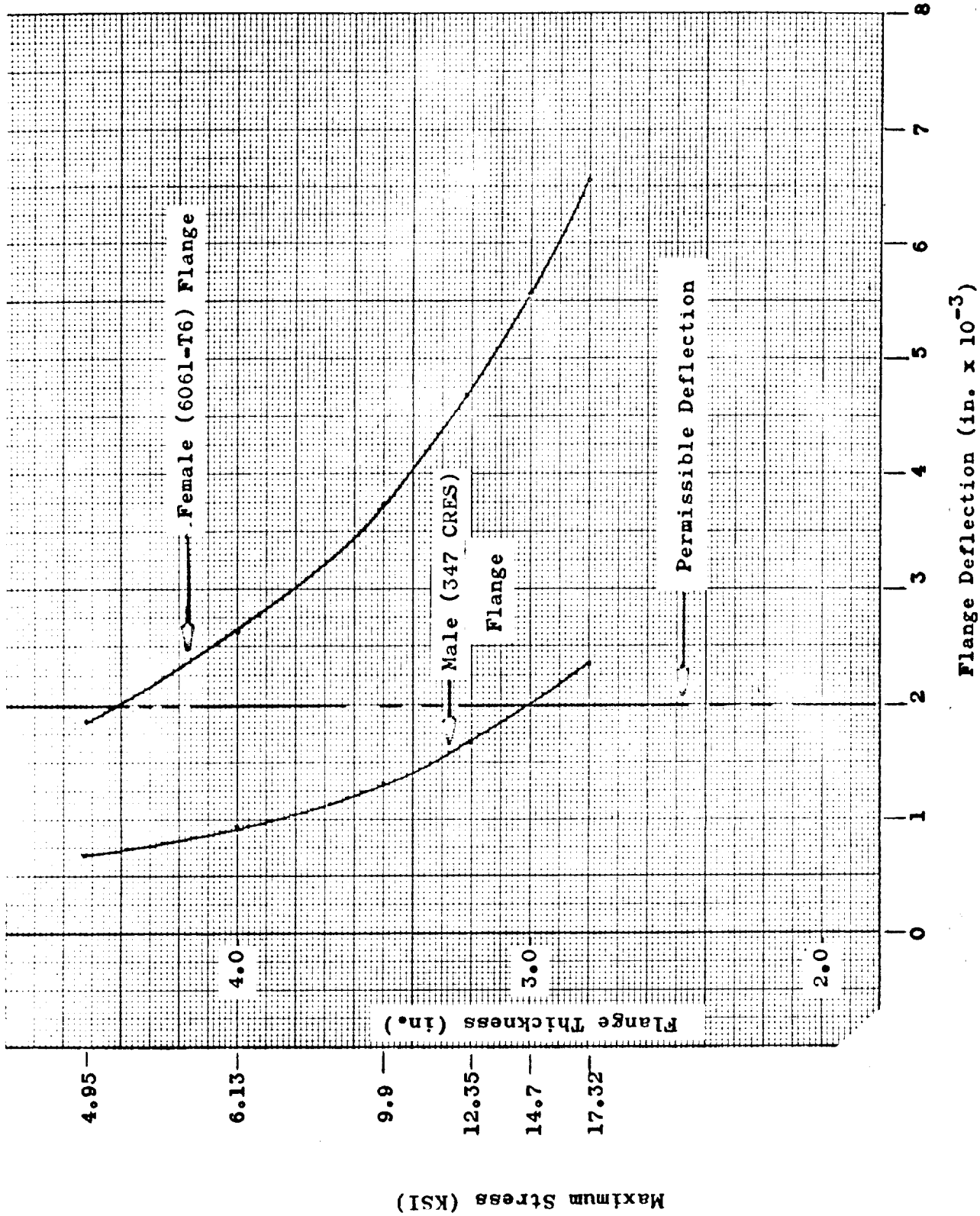


Figure 30
Calculated Flange Stresses and Deflections for Various Flange Thicknesses of the Preliminary Joint Design

as the criticality of the required surfaces. Elastomeric flat gaskets and O-rings can be handled with minimum care without apparent damage, although they cannot be excessively squeezed, cut, or pinched. Plastic gaskets (particularly, the spring-loaded, pressure-energized gaskets that have small protruding sealing edges) are much more susceptible to damage. The metal gaskets are extremely susceptible to damage because the soft metals used at the exposed sealing surfaces are easily scratched or damaged. High strength (hard) steel flanges are usually difficult to damage unless the sealing surfaces are exposed and the parts inadvertently struck with a hard object or rubbed across abrasive surfaces. Exposed sealing surfaces of aluminum and softer stainless steel flanges have greater susceptibility to damage.

There are three precautions to preclude handling damage. The steps in handling should be reduced to an absolute minimum, proper protective packaging and equipment should be provided for all phases of handling up to the assembly of the joint, and necessary handling and assembly procedures should be developed to eliminate the possibility of damage. Special tools and equipment may be required in the inspection and cleaning areas. One method used satisfactorily for the proprietary gaskets is to require that the supplier clean and properly package the parts before shipment. This necessitates either inspection at the source of supply or a destructive sampling by the user, but it provides greater assurance of satisfactory parts. The same procedure, while less desirable, could be used for flanges. It is much easier to provide protective covers or packaging that can be installed immediately after machining and used in all handling operations thereafter.

9. Assembly Procedures and Problems

The final problem in attaining a good joint is the assembly of the component parts. This involves not only proper handling to prevent damage to the parts, but good techniques of assembly to assure proper seating at the sealing interface.

If adequate protective packaging is provided after the cleaning operation, interface surfaces will be free of foreign particles that could interfere with proper sealing surface contact. If cleaning is not previously required, it will be necessary to assure that all interface surfaces are free of foreign particles. This must be accomplished using solvents that are compatible with the system fluids and lint-free (non-sluffing) tissues or cloth. Hard-faced tools must not be used during the cleaning operation as they can easily damage the sealing surfaces. Soft plastic devices with tissue or cloth covers are usually used for cleaning flange surfaces.

Proper lubrication of the joint fasteners will be required. As previously mentioned, this can determine whether sufficient axial load is imparted to the fasteners to provide the essential preload. If spray-on coatings are included in the fastener design and the method for determining the proper loading has been developed, the problem can be reduced or eliminated.

Lubricants, greases, and similar substances should not be used on the gaskets. They may affect the adequacy of the seal during the operating life of the joint. If lubricants are needed so as to pass the initial leak test, the parts should be examined to assure that all machining and surface finish requirements have been satisfied. The joint design should provide sufficient deformation at the sealing interface to preclude the necessity for using lubricants, greases, and similar substances to achieve a satisfactory seal.

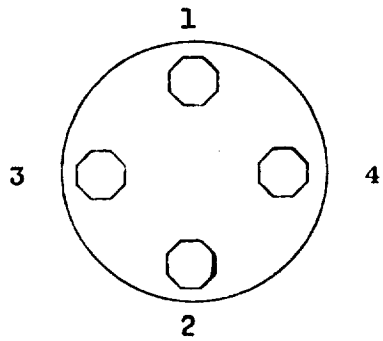
Alignment of the two halves of the joint must also be considered in assembly. If this appears to be a potential problem, it should be considered during the design of the joint. Alignment pins, shoulders, and/or similar means should be provided to accomplish proper alignment before contact is made with the gasket. Otherwise, a relatively non-precise assembly operation can result in an unsatisfactory (leaking) joint.

The flanges must be drawn together as parallel as possible to provide even loading at the sealing interface. This requires a proper tightening sequence on the fasteners. The acuteness of the problem depends upon gasket materials (resiliency and how critical any unequal loading is) as well as how stiff the flanges are. The more resilient elastomer gaskets tend to account for some non-parallelism between the flange faces and minor loading variations around the periphery of the gasket. On the other hand, the metal crush washers and flat plastic gaskets are extremely dependent upon flange parallelism and equal loading around the periphery to obtain and maintain sufficient sealing stresses. The common elastomeric, ambient service O-ring presents the least acute assembly problems because its main requirement is an assurance that no gap exists between the flange faces. However, the loading around the periphery must be reasonably equal to prevent separation of the flange faces, thereby preventing extrusion and nibbling of the O-ring. The plastic-jacketed gaskets are equivalent to the elastomer O-rings as regards assembly. Equal bolt loading becomes more critical with the elastomer O-rings in extremely low temperature service because of the very high squeeze and accompanying high loading required to provide a satisfactory joint.

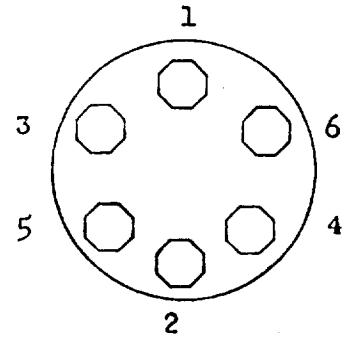
Parallelism between the flange faces and equal loading around the periphery of the gasket is extremely critical for the less resilient, flat plastic gaskets as well as the all-metal crush washers and gaskets. These gaskets depend upon parallelism and loading to both obtain and maintain equal sealing stresses around the periphery of the gasket.

A suggested tightening sequence pattern for the elastomer O-rings and plastic-jacketed gaskets is shown on Figure No. 31. The more critical tightening sequence patterns for the other gaskets are shown on Figure No. 32. The "1/2-turn" value shown on Figure No. 32 has proven to be satisfactory for most all-metal gasket joints. However, it is dependent upon the fastener thread pitch and the load compression characteristics of the gasket; therefore, a smaller value may be necessary for some joints. In these latter instances, the permissible value must be determined by actual tests and this value must then be specified on the joint assembly drawing and specifications.

EXAMPLE A



EXAMPLE B



ALLOWABLE BOLT PATTERNS		
MULTIPLES OF 4	MULTIPLES OF 6	
4	6	
8	12	72
16	24	90
	36	120
	48	180
	60	240

1. WHEN ASSEMBLING FLANGED JOINTS, INSTALL BOLTS AND TIGHTEN IN SEQUENCE, AS SHOWN IN EXAMPLES A OR B, AS APPLICABLE*, UNTIL THE FLANGES AND BOLTS ARE SEATED.

   * ON FLANGES THAT HAVE BOLT PATTERNS WHICH ARE MULTIPLES OF EXAMPLES A OR B (SEE ABOVE TABLE), SELECT THE NEXT BOLT MIDWAY BETWEEN TWO BOLTS THAT HAVE ALREADY BEEN TIGHTENED AND REPEAT THE SAME CYCLE UNTIL AT LEAST 50% OF THE BOLTS THAT HAVE BEEN TIGHTENED - THE REMAINING BOLTS ARE TIGHTENED IN RANDOM SEQUENCE.

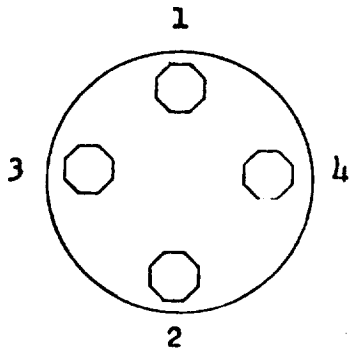
2. TIGHTEN ALL BOLTS TO 1/3 OF SPECIFIED TORQUE VALUES IN THE SEQUENCE INDICATED IN 1 ABOVE.

3. TIGHTEN ALL BOLTS TO 2/3 OF SPECIFIED TORQUE VALUES IN THE SEQUENCE INDICATED IN 1 ABOVE.

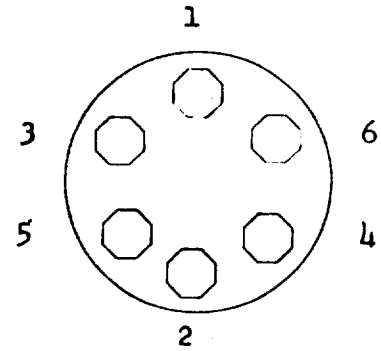
4. TIGHTEN ALL BOLTS TO 100% OF SPECIFIED TORQUE VALUES IN THE SEQUENCE INDICATED IN 1 ABOVE.

Figure 31
Suggested Tightening Sequence Patterns, O-Rings (Standard Squeeze) and Plastic-Jacketed Gaskets

EXAMPLE A



EXAMPLE B



ALLOWABLE BOLT PATTERNS	
MULTIPLES OF 4	MULTIPLES OF 6
4 8 16	6 12 72 24 90 36 120 48 180 60 240

1. WHEN ASSEMBLING BOLTED JOINTS, INSERT THE BOLTS AND TIGHTEN THEM FINGER-TIGHT.

2. TIGHTEN ALL BOLTS 1/2 TURN[a] IN THE SEQUENCE SHOWN IN EXAMPLES A OR B, AS APPLICABLE[b]. BEFORE APPLYING THE WRENCH FOR THE 1/2 TURN[a], ASSURE THAT THE BOLT IS FINGER-TIGHT.

   (a) Except where smaller values are specified on the Engineering Drawing.

   (b) On flanges that have bolt patterns which are multiples of examples A or B (see above table), select the next bolt midway between two bolts that have already been tightened and repeat the same cycle until all bolts have been tightened to the specified value.

3. CONTINUE THE TIGHTENING PROCEDURE UNTIL THE SPECIFIED TORQUE IS APPLIED TO EACH BOLT.

Figure 52
Suggested Tightening Sequence Patterns, High-Squeeze O-Rings, Flat Elastomer and Plastic Gaskets, and All-Metal Gaskets

B. TEST PROCEDURES AND TEST EQUIPMENT

There were two types of testing used as data sources. These were the laboratory tests, wherein seal performance evaluations were accomplished, and the component development tests, which encompassed the static seals at the interfaces.

There were two phases to the Laboratory Testing; the preliminary tests and the pre-qualification tests. Table VI is a listing of the tests conducted in each phase.

The flanges and fasteners used in the preliminary test series were designed so as to eliminate, insofar as practicable, deflections and distortions. Typical preliminary, heavy-duty, test flanges are shown on Figure No. 33. Grooves, .002-in. to .005-in. deep by .030-in. wide, were machined radially across the face of the female flange to assure that a seal was not formed between the flat faces making contact between the flanges.

The purpose of the preliminary test series was to obtain design data concerning the preparation of the mating surface, the seating load and assembly procedures needed for a particular gasket configuration, and the minimum leakage rate capability of the design. In addition, preliminary data was gained pertaining to the ability of the gasket to follow deflections and distortions in the flanges.

Test cells were then designed and fabricated for a particular gasket based upon the preliminary test data. These test cells were of a minimum weight construction and suitable for a particular application of design pressures, temperatures, fluids, and environments. They also simulated the system joints proposed for the application. These test cells were used for the pre-qualification test series listed on Table VI. Figures No. 34 and No. 35 show typical test cells that were designed for two different applications.

The Pre-Qualification Test Series had a twofold purpose; to verify, as closely as possible, the design parameters gained from both the preliminary tests and other available data as well as to define any potential problems that might be encountered with the system joints. Combination tests (i.e., low temperature vibration) were also conducted as part of this series when the combinations tested were considered to be potential problems.

The data accumulated during the laboratory testing was eventually used for actual component joint design.

The data gained from the component development tests is included herein along with the procedures and test equipment used for each of the tests.

TABLE VI

LABORATORY TEST SERIES AND SEQUENCE

TEST SERIES	FLANGE TYPE	TESTS CONDUCTED
Preliminary	Heavy Duty	1. Detailed Flange and Gasket Inspection. 2. Loading Tests. 3. Assembly Requirements (Procedure from Figure No. 31 or No. 32) - Modified as necessary to properly seat the particular gasket configuration. 4. Proof Tests. 5. Leak Tests. 6. Flange Axial Separation and Angular Misalignment Tests. 7. Flange Reusability Tests. 8. Preliminary Burst Tests.
Pre-Qualification	Minimum Weight	1. Detailed Flange and Gasket Inspection. 2. Proof Tests. 3. Leak Tests. 4. Temperature Tests (Low and High). 5. Thermal Shock and Thermal Cycling. 6. Fluid Resistance and Compatibility. 7. Salt Spray and Storage. 8. Vibration Tests. 9. Burst Tests.

Figure 33
Typical Preliminary Test Flanges

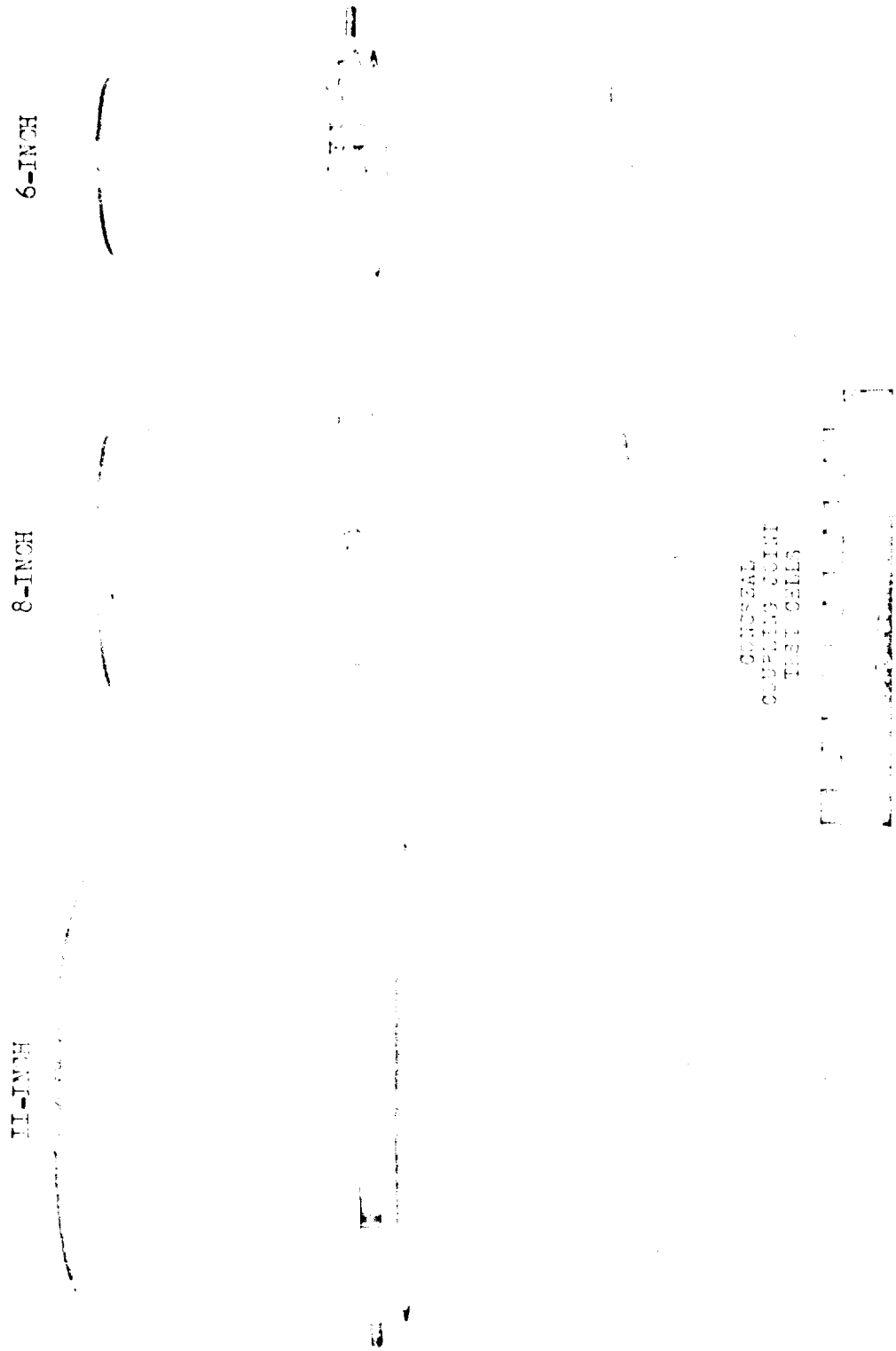


Figure 34
Typical Pre-Qualification Test Cells

DUAL CONICAL JOINT TEST CELLS

Figure 35
Typical Pre-Qualification Test Cells

1. Laboratory Evaluation Tests

a. Flange and Gasket Inspection

The actual critical surface finishes, flatness, and dimensions of both the mating flanges and gaskets were inspected and recorded. The critical sealing surfaces on the flanges and gaskets were first visually examined (using microscopes if required) for nicks, dents, scratches, or other imperfections that would be detrimental to the effectiveness of the seal. The gasket and flange surface finishes were determined using a direct-reading surface analyzer (Taylor Hobson Talysurf Model 3) when the surface hardness and the analyzer stylus loads were such that no scratches were introduced on the surface by the stylus. For the softer surfaces, surface replicas were analyzed. However, in many instances, the surface finishes were determined by visual comparison using standard surface gauges. This was particularly true for the softer gasket materials, such as Teflon. The accuracy of this method is limited, because it largely depends upon interpretation by the inspector. The free height and critical diameters of the gasket and the groove depths, diameters, and other critical dimensions of the flanges were determined using instruments that were either modified or designed to prevent any extraneous scratches, scores, or other damage to the critical surfaces during the inspection.

b. Loading Tests

The loading tests were conducted using the preliminary (heavy duty) flanges. The test set-up for the loading tests is shown on Figure No. 36. The weight of the upper flange (usually the male) was determined, the gasket and the upper flange was installed, and the flange gap determined for the "no load" condition. The "no load" condition was estimated from the known gasket free height and groove dimensions for those gaskets designed for very light seating loads. The flange gap obtained with the upper flange weight was used as the initial point for those gaskets requiring higher seating loads because this load was essentially the same as the zero load point. The dial indicators were then "zeroed." The load was then applied through the calibrated load cell in even increments and the flange gap measurements and dial indicator readings recorded at each increment until the flange faces "bottomed-out" (in the case of the pressure energized gaskets) or until a predetermined compression was obtained (flat gaskets, crush washers, and some diametral gaskets). The load was then relieved in even increments and the flange gap measurements and dial indicator readings recorded at each increment until the load became zero. After completion of the loading tests, the gasket was inspected to determine the permanently deformed sealing edge width, mean sealing edge diameter, and gasket free height. In some cases, the gasket was carefully sectioned and photomacrographs made to further analyze the deformed configuration. Also, the critical surfaces on the mating flanges were examined for any detrimental distortion or scoring.

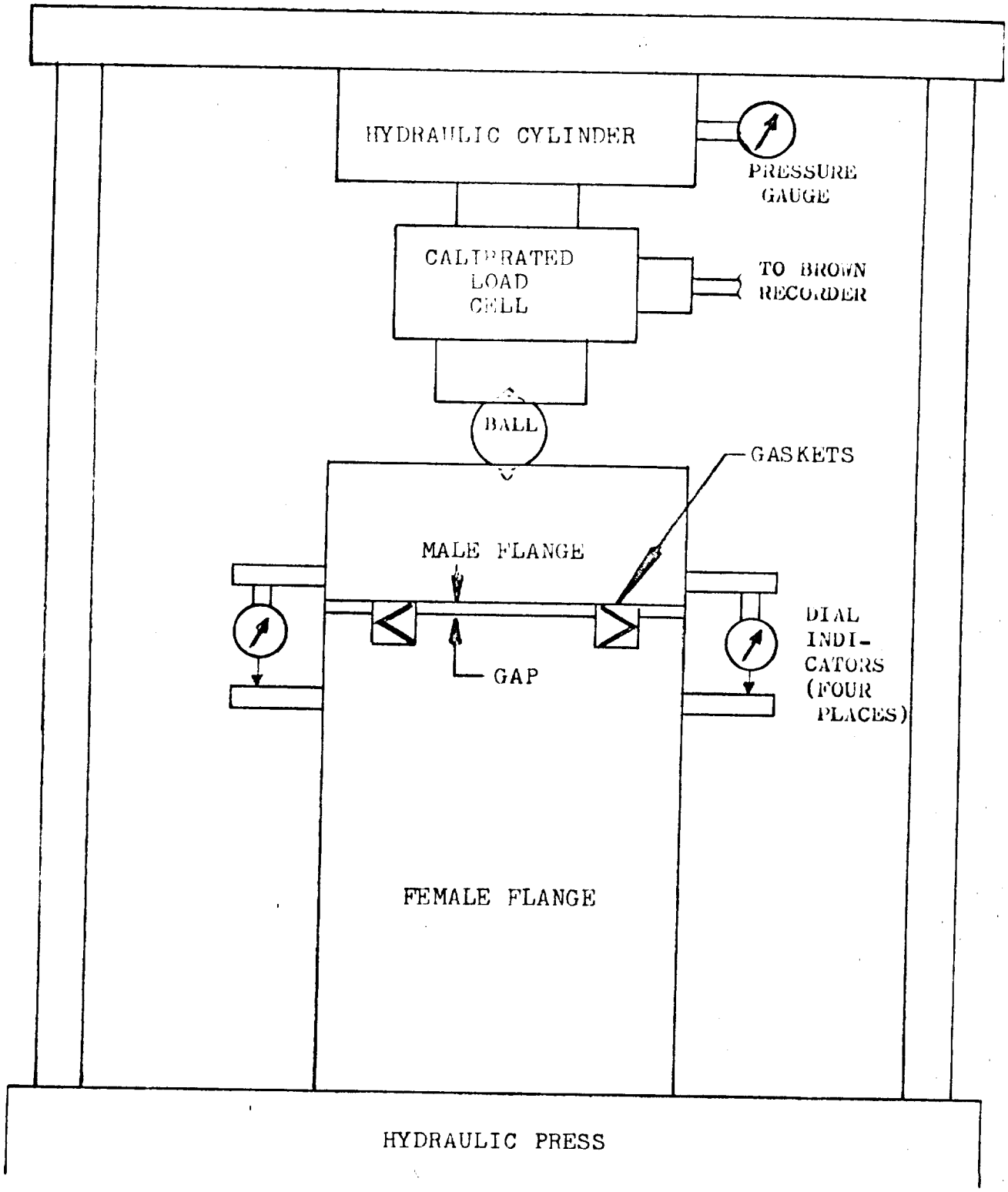


Figure 36
Loading Test Set-Up

c. Assembly Procedures and Requirements

The assembly procedures used depended upon the flange and fastener configuration as well as the particular gasket being tested. Initially, one of the sequences shown on Figure No. 31 or No. 32 was used for bolted joints after the bolts were lubricated and installed "finger-tight" while maintaining flange parallelism. Modifications were made to this procedure if necessary to properly seat the particular gasket configuration. The other joint configurations (i.e., V-band coupling joints) were assembled in accordance with the manufacturer's recommendations.

d. Proof Tests

Each joint was proof tested before the initial leak tests were conducted to ascertain if any detrimental effects were introduced at proof test pressures. Proof pressures were 150% of the maximum expected operating pressure for the particular application or 200% of the nominal operating, if the maximum operating pressure was not known. However, most of the proof tests were conducted at 2000 psig. Pressure was applied in increments of 10%, 25%, 50%, 75% and 100% of proof pressure and held for five minutes at each increment, during which time, the joint was visually examined for leakage. The test media used was water, gaseous helium, or gaseous nitrogen depending upon the application. Gas proof tests were conducted with the joint submerged in water to facilitate detecting any leakage. Pressures were then lowered in the same increments and held for five minutes at each increment. If leakage was noted, the joint was disassembled and the parts were reinspected to determine what permanent distortions or other detrimental changes had occurred.

e. Leakage Tests

Inert gas was used to pressurize the test joint in increments of approximately 10%, 25%, 50%, 75%, and 100% of final leakage test pressure and held for a minimum of five minutes at each pressure level. Final leakage test pressure was 75% of proof pressure (usually 1500 psig). Two methods for detecting leakage was used; the test joint was submerged in water and leakage was collected in a graduated beaker or leakage was detected using a mass spectrometer leakage detector. Figure No. 37 illustrates a typical water submersion leakage test set-up for the single seal. The set-up for dual seal joints is shown on Figure No. 38. Leakage was collected in the water-filled inverted graduated beaker at each test pressure. The primary seal leakage monitoring line for the dual seal joints was sufficiently pressurized to emit gas bubbles from the submerged end of the line before leakage test pressures were applied to the test joint. The low-pressure gas line hand-valve was then shut-off, checked for leakage, and the beaker was installed over the collector tube end. Then, the water level in the beaker was noted. The test pressure was applied to the test cell and the change in the water volume within the beaker was recorded during the hold period at each pressure level. Although helium gas was used for most of the tests, some of the tests were conducted using nitrogen gas. If there was no leakage detected using this method, the joint was then subjected to a leakage check using the mass spectrometer (Consolidated Electronics Corporation Model 24-210B Helium Leak Detector).

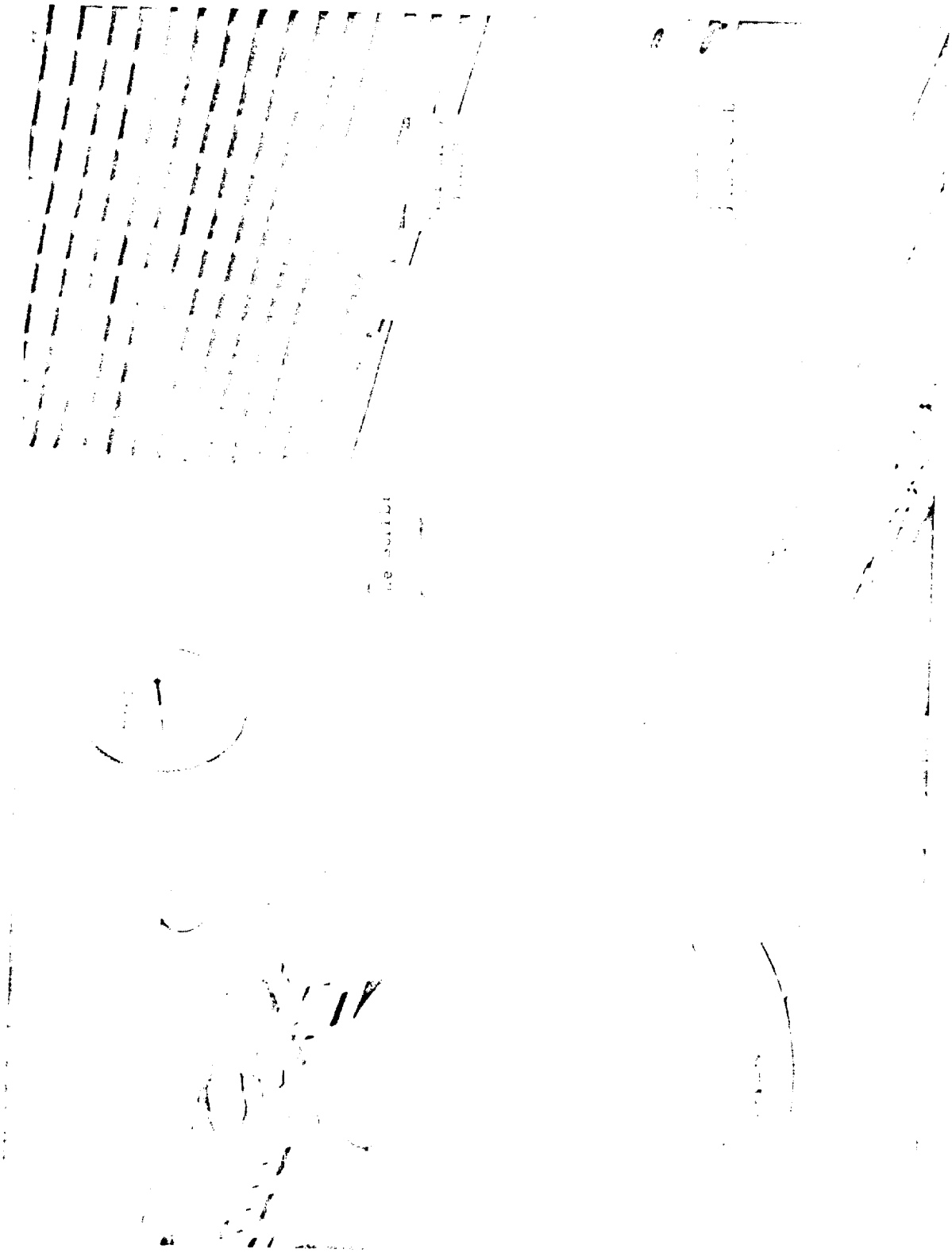


Figure 37
Leak Test Set-Up for Single Seal Joints
(Water Submersion Method)

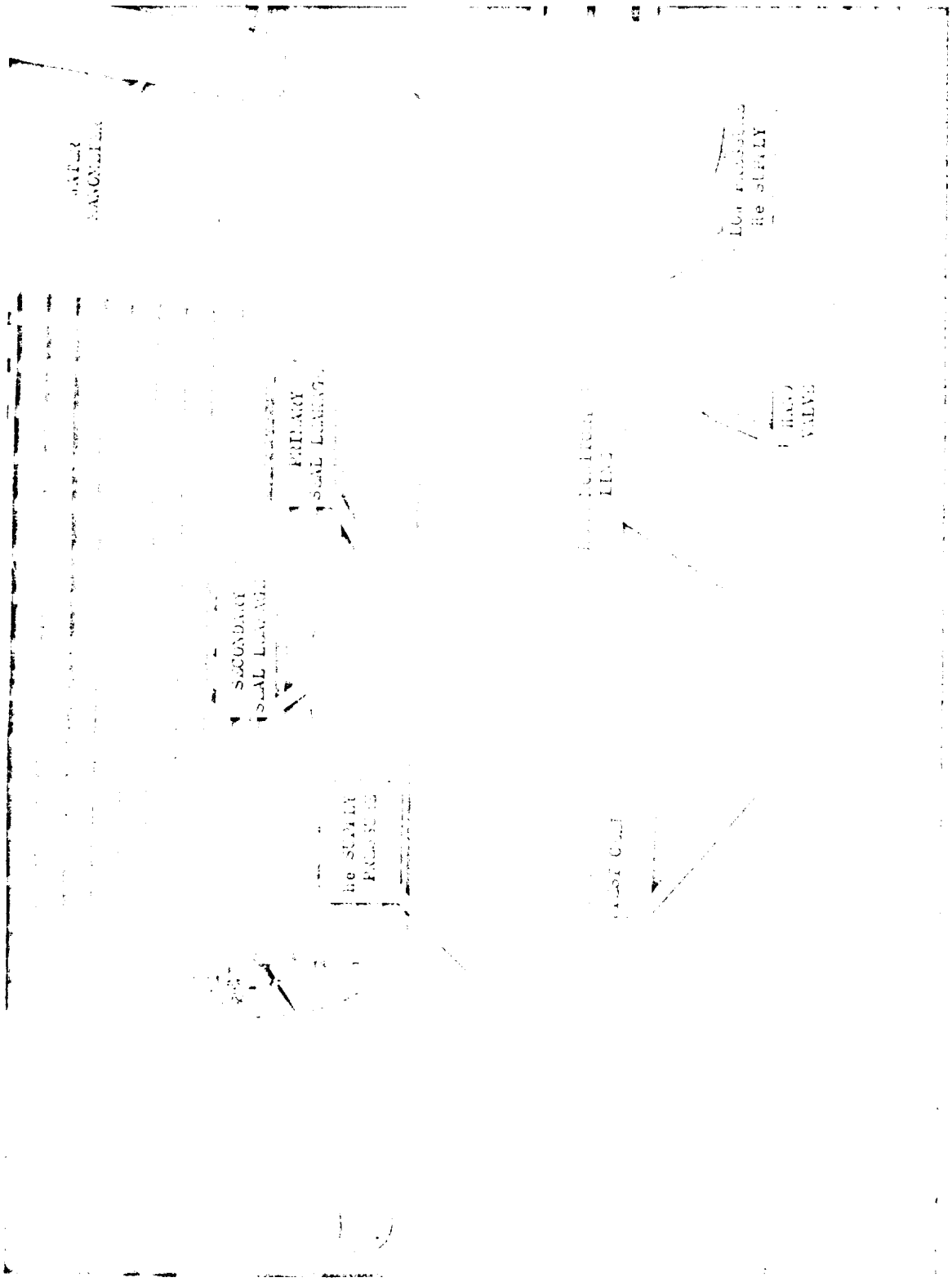


Figure 38
Leak Test Set-Up for Dual Seal Joints
(Water Submersion Method)

Two methods of leakage detection with the mass spectrometer were used for the single seal joints. In one method, the sampling probe attachment of the mass spectrometer was used as illustrated on Figure No. 39. First, the leakage detector was calibrated. Then, the sample probe, which is more commonly known as the "sniffer," was connected and laboratory background readings were determined. The joint was then scanned by moving the "sniffer" around the outer periphery of the flange interface to determine if any leakage existed above the background reading at each pressure level. This method of leakage detection poses a problem in that leakage can exist but be below the background reading (usually approximately 1.0×10^{-7}) of the detector and it is assumed that leakage is less than the background value. Should leakage be detected above the background value, it is difficult to either assign or calculate a true leakage value because the probe collects from only one small area at a time and the background value can be different for each area. This is particularly true if the reading varies around the periphery of the joint. In addition, the detector is analyzing a pre-set constant flow of gas into the mass spectrometer.

The second method applied to single seal joints involved the use of the mass spectrometer in conjunction with a vacuum chamber as illustrated on Figure No. 40. The test joint was installed into the vacuum chamber and connected to the gas pressurization line. The leakage detector was connected in parallel with the vacuum chamber pump. Initial chamber evacuation was accomplished with the vacuum chamber pump and completed by using the leakage detector system. After the system was evacuated to the level required for the mass spectrometer, the background reading was determined (helium leakage into the chamber from the external environment), the test cell was pressurized, and leakage measurements were made at each pressure level. The total leakage from the test joint was assumed to be the difference between the readings at each pressure level and the background reading. The accuracy of this method depends upon the number of other pressurized joints (i.e., tube connectors) that are inside the vacuum chamber and upon whether the background reading is above the lower limit of the leakage detector. Accuracy can be greatly improved by eliminating all of the pressurization line connectors from inside the vacuum chamber.

A typical leakage test set-up for the dual seal joints using the mass spectrometer is shown on Figure No. 41. The leakage sampling line of the mass spectrometer is attached to the leakage monitoring port of the test cell. This leakage monitoring port vents the interseal cavity, which is evacuated to the level required for the leakage detector and any background reading is noted. The test cell is then pressurized and readings are taken at each pressure level. The total leakage from the primary (inner) seal is assumed to be the difference between the background reading and the reading at each pressure level. The accuracy of this method depends upon the characteristics and capabilities of the secondary (outer) seal as well as the deflections encountered in the flanges and fasteners at the various pressure levels. This is the most accurate of the leakage test methods used because the pressure differential across the secondary outer seal is constant providing that

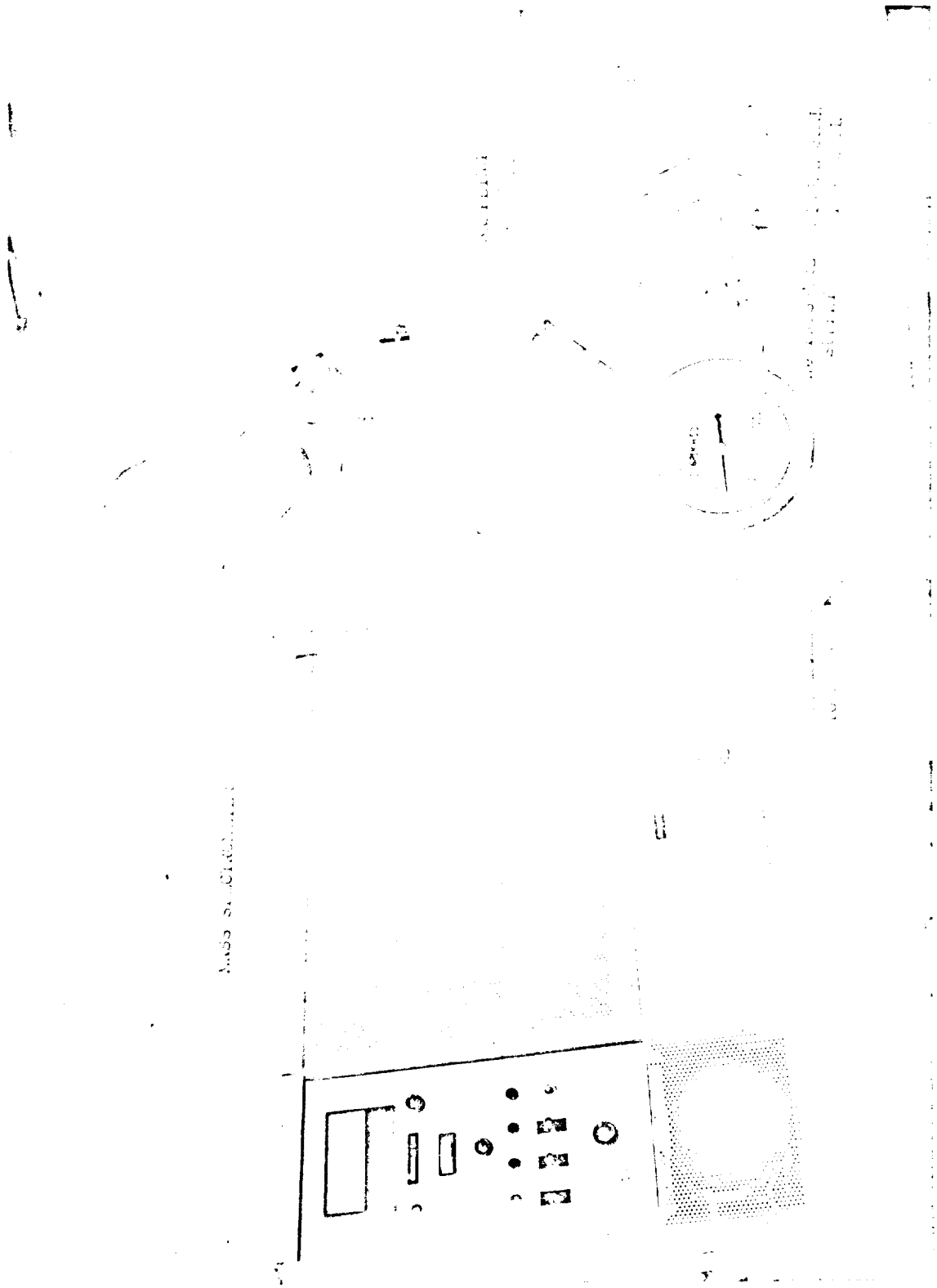


Figure 39
Leak Test Set-Up for Single Seal Joints
(Mass Spectrometer-Probe Method)

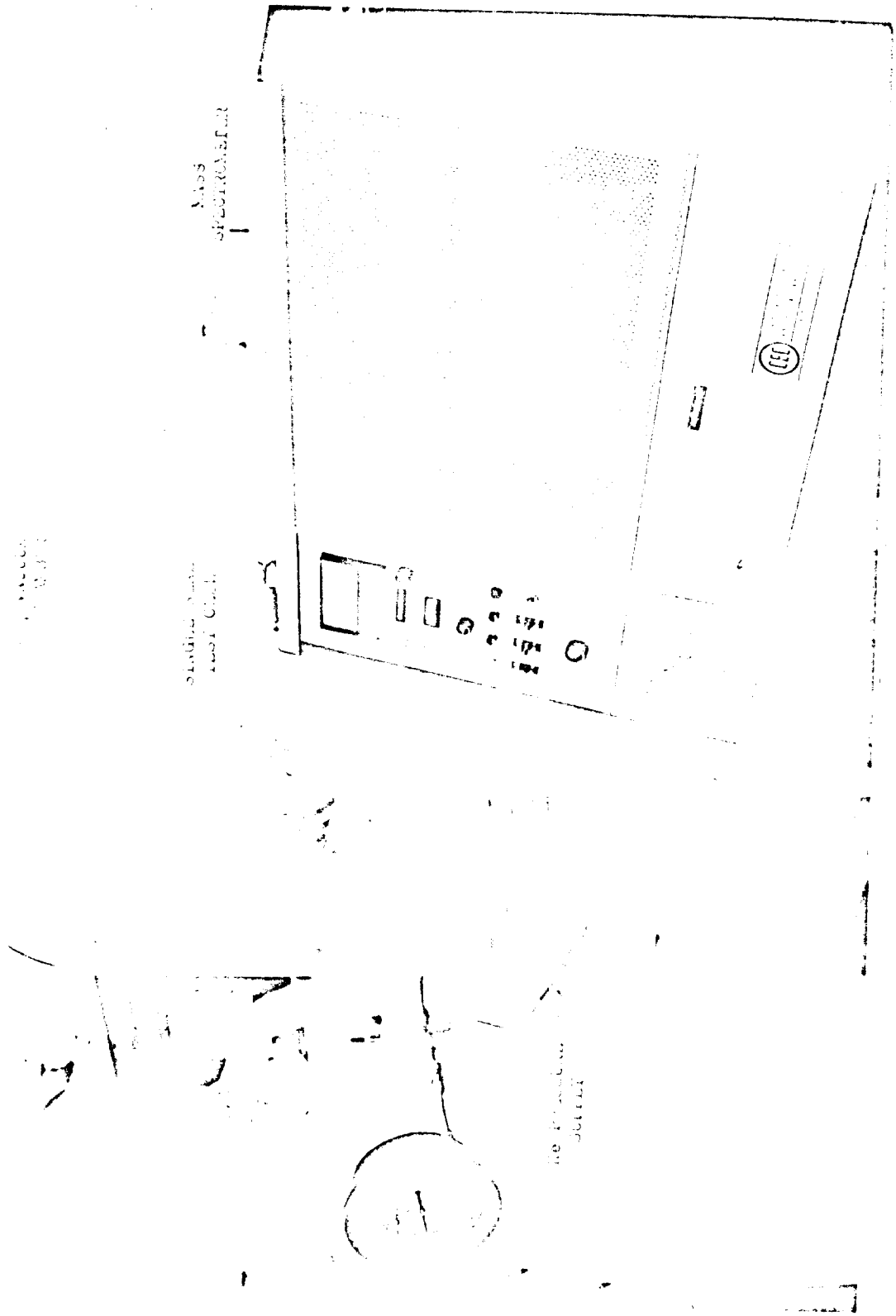


Figure 40
Leak Test Set-Up for Single Seal Joints
(Mass Spectrometer-Vacuum Method)

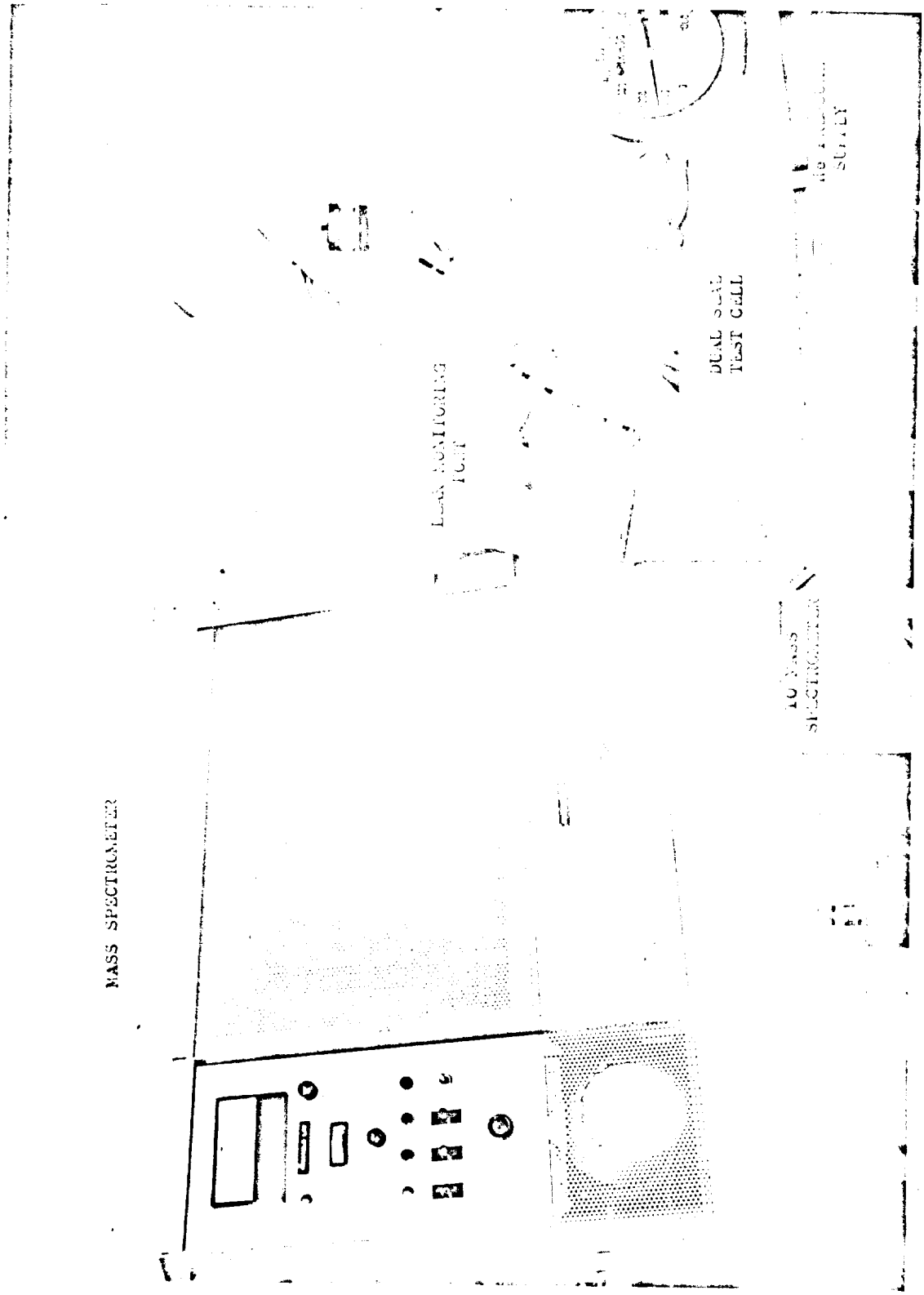


Figure 41
Leak Test Set-Up for Dual Seal Joints
(Mass Spectrometer Method)

no deflections occur in the flanges at the various pressure levels or the secondary seal is capable of maintaining the seal even if deflections are encountered.

f. Flange Angular Misalignment Tests

Tests were conducted to determine the necessary parallelism which must be maintained between the flange faces. Only the heavy duty flanges were used. These test results were used as an indicator of permissible joint deflections resulting from externally-applied bending moments and/or the permissible angular misalignment between the flange faces during the assembly of a particular gasket configuration. The joint was first assembled in the normal manner. It was then leak tested, proof tested, and again leak tested to determine the initial leakage rate. The bolts were then loosened only to the extent needed to permit the insertion of a .002-in. thick shim (or a shim .002-in. thicker than the normal flange gap at the bolt circle) between the flanges for either a one-third or one-quarter segment of the bolt circle as shown on Figure No. 42. The bolts were then retightened in proper sequence and the joint was again leak tested, proof tested, and leak tested. Additional .002-in. thick shims were added after each test cycle until the leakage increased sharply or the seal failed.

g. Flange Axial Separation Tests

Straight axial separation tests were conducted to determine the effects of axial deflections, or to gain an indication of the axial deflection permissible for a particular gasket configuration. Only heavy duty flanges were used. Two methods were used. In the first method, the joint was assembled in the normal manner and leak tested to determine the initial leakage rate. The bolts were then loosened just enough to insert a .002-in. thick shim in one-third the bolt circle segments, similar to the assembly shown on Figure No. 42 except this time all around between the flange faces. With gapped flanges, shims were .002-in. thicker than the flange gap. If the joint again satisfactorily passed the leakage test, additional shims were added in .002-in. increments until leakage increased sharply or the seal failed. The second method was to decrease the bolt (or fastener) torque values in predetermined increments and measure the change in flange gap with indicating gauges connected to the flanges at each incremental pressure.

h. Flange Reusability Tests

Flange reusability tests were conducted with most of the all-metal gaskets, particularly if the gasket base material hardness was either approximately equivalent or greater than the flange material hardness. The joint was properly assembled, leak tested, proof tested, and leak tested. The joint was then disassembled and reassembled (using a new gasket at each assembly) for a total of 30 assembly cycles or until leakage became excessive. The flange surfaces were then reinspected to determine the cause of leakage. If necessary and possible, the flanges were re-machined to the acceptable requirement for the gasket. Gasket reusability tests were not conducted because it was previously determined that reusability was not desirable.

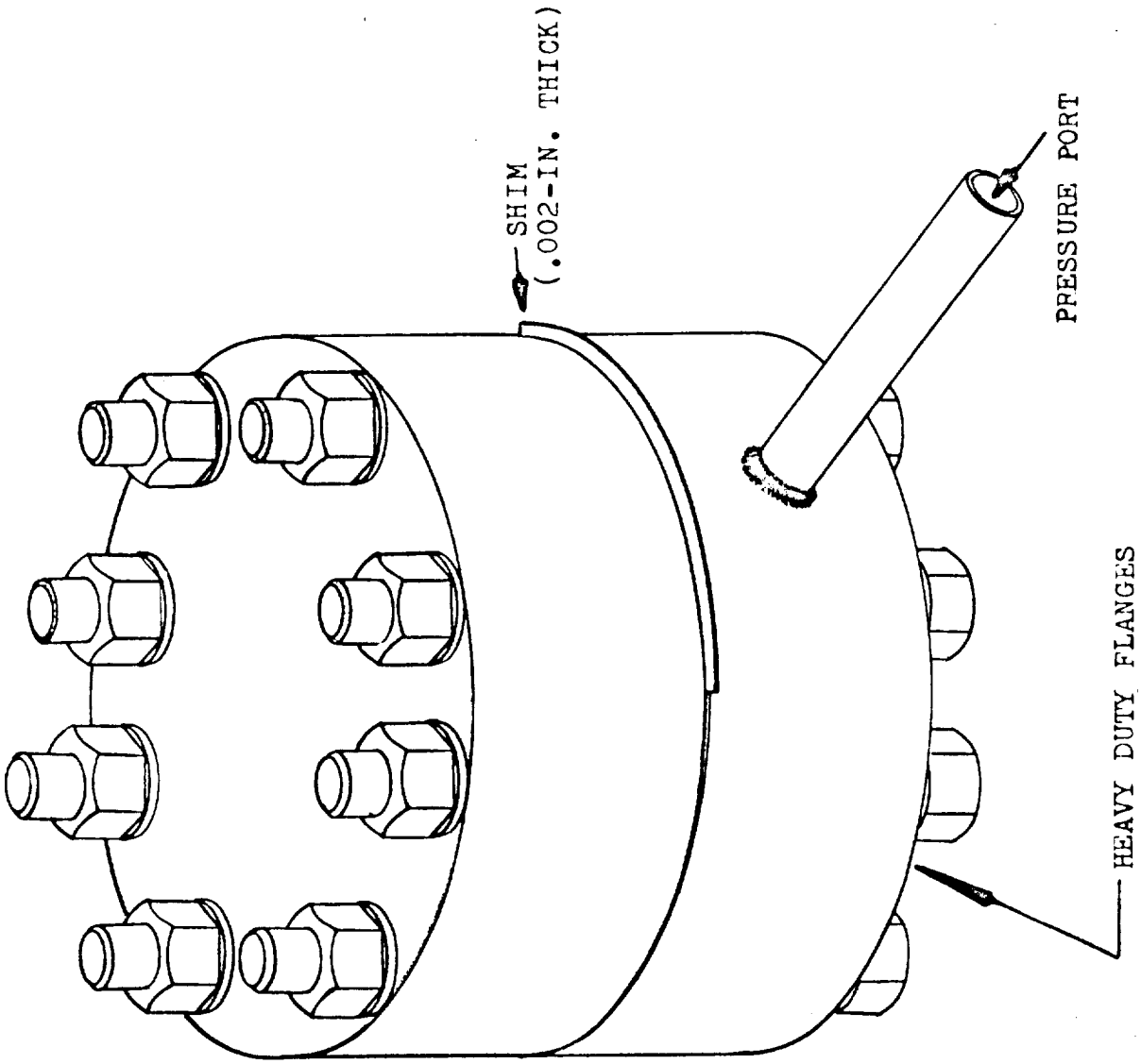


Figure 42
Flange Angular Misalignment Test Set-Up

It was determined to be extremely difficult to reassemble the gasket in the previous position and the possibility of damage in removal precluded the acceptability of re-using the gasket. However, some gasket reusability data was obtained under laboratory conditions when the gasket was deemed suitable for additional tests.

i. High Temperature Tests

High temperature tests were conducted with some test cells, primarily the all-metal gaskets, at the maximum temperature expected for a particular application. The joint was either soaked in an oven (if the maximum desired temperature was less than 300°F) or flowed with hot gases (if the maximum desired temperature was above 300°F) until the joint temperature stabilized. Leakage tests were made at ambient temperature before the test, at stabilized high temperature (if feasible), and again at ambient temperature. The maximum temperature used depended upon the application for which the particular seal was being tested. The joint was then disassembled. All parts were examined and reinspected for any detrimental changes. Figure No. 43 is a schematic of a typical test set-up for test temperatures above 900°F.

j. Low Temperature Tests

Low temperature (cryogenic) tests were conducted in two different ways to simulate conditions expected with actual hardware joints depending upon their locations in the various systems. The first method involved low temperature bleed-in with subsequent pressurization, and the second procedure included simultaneous pressure and temperature application. Ambient temperature leak tests were conducted on all joints before the low temperature tests were performed.

Low temperature bleed-in was performed at low pressure until the temperature stabilized in the joint. The maximum leakage pressure was then applied and the leakage rate was determined with a mass spectrometer or by visual observation if the rate was above the mass spectrometer capabilities. The test media was liquid nitrogen (for liquid oxygen system applications) or liquid hydrogen during bleed-in, and low temperature helium for the actual leak test. Figure No. 44 shows a typical test set-up for dual seal test cells.

The second method for low temperature testing consisted of flowing cryogenic fluid through the test cell at design pressure (i.e., pressure and temperature applied simultaneously) until the cell temperatures stabilized. If no visible leakage was observed, low temperature helium was applied to determine the actual leakage rate with the mass spectrometer if the leakage rate was within the measuring range of the instrument.

In most cases, the fastener torque value was determined both before and after the tests.

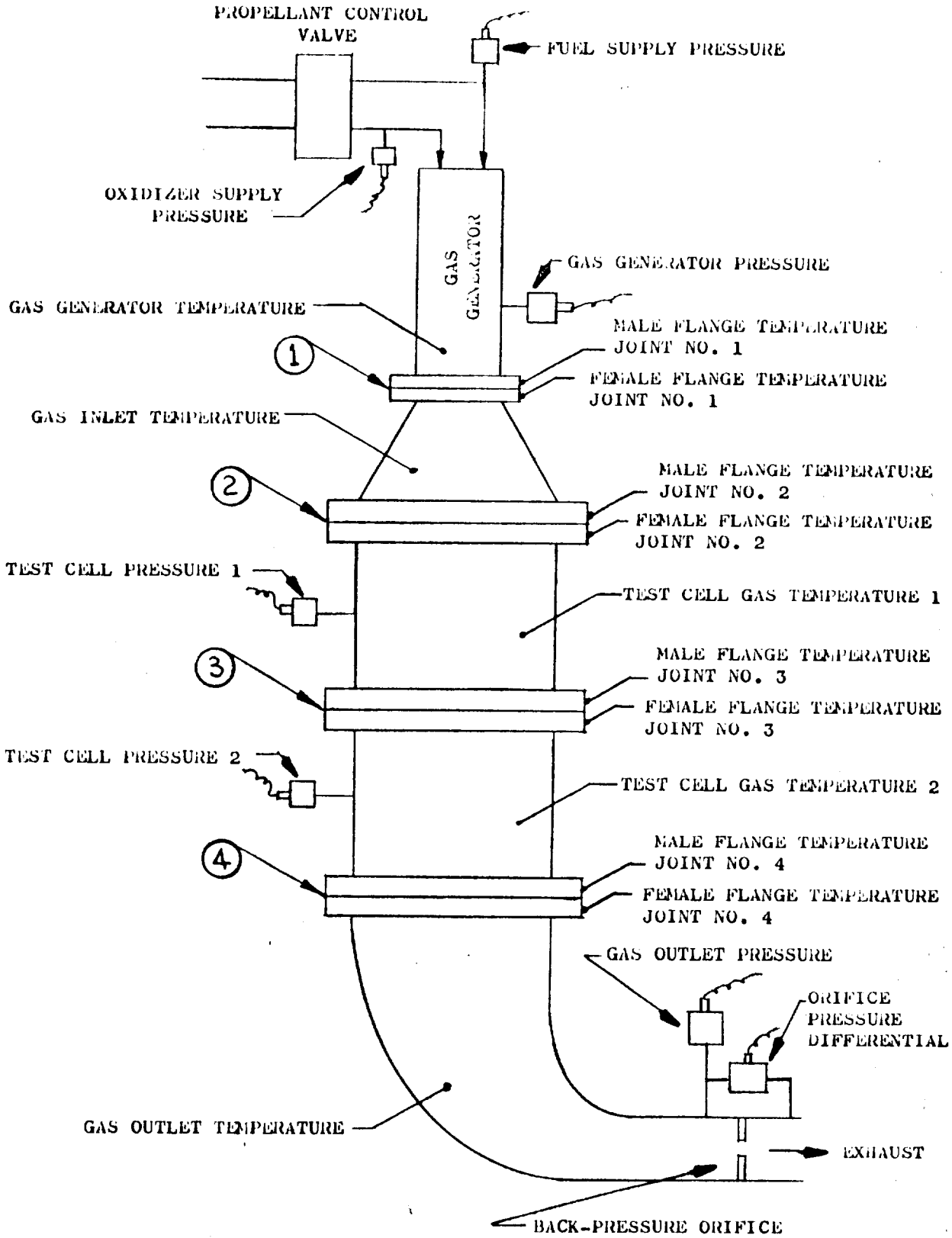


Figure 43
Typical High Temperature and Thermal Cycling Test Set-Up

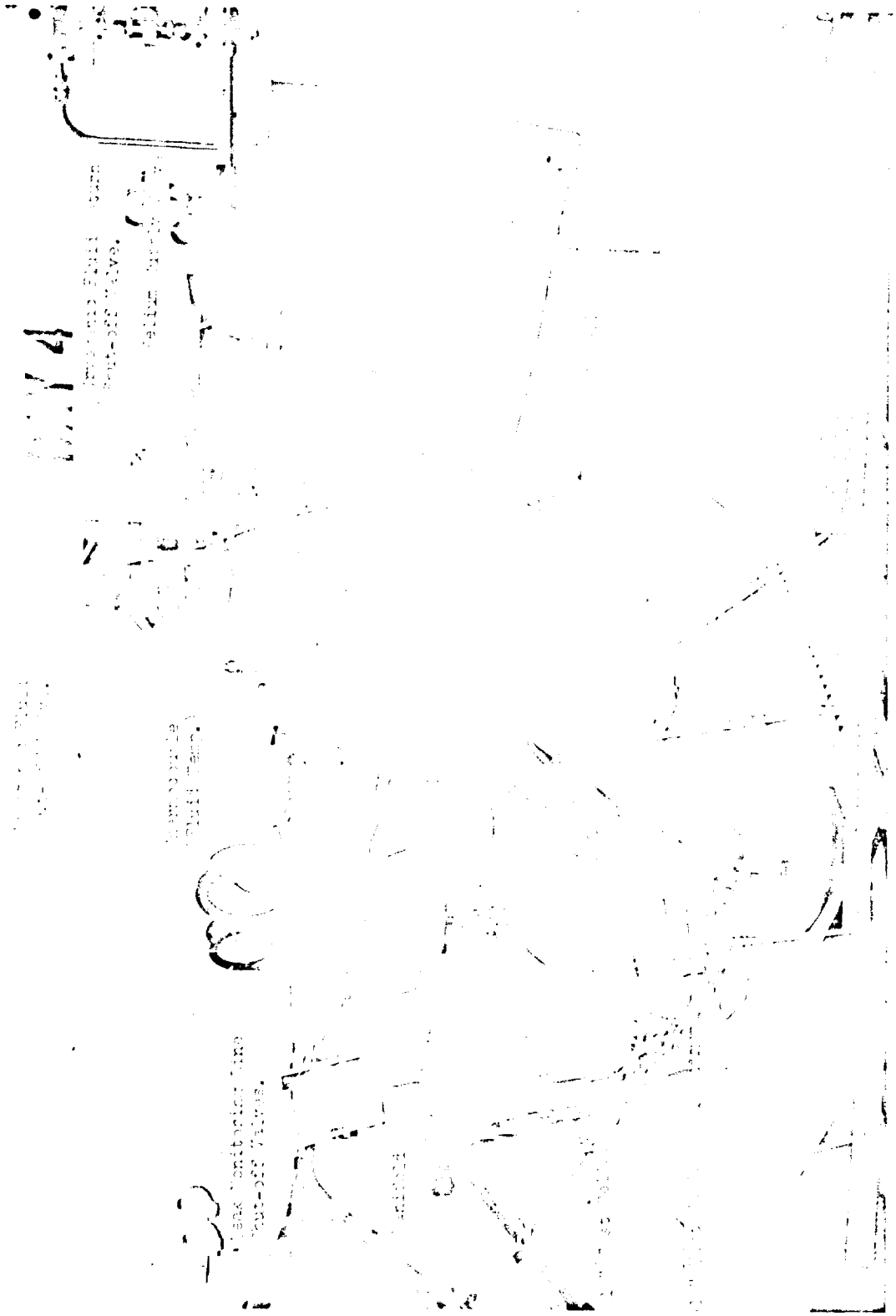


Figure 44
Typical Low Temperature and Thermal Cycling Test
Set-Up (Dual Seal Joints)

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k. Thermal Cycling Tests

If the joint passed the high and/or low temperature tests, thermal cycling tests were conducted to ascertain the affects of cycling upon leakage rates, changes in fastener tension, and other parameters. These tests were conducted in two different ways depending upon the application desired. With joints for low temperature applications, the joint was: soaked at the highest expected environmental temperature and leak tested; flowed with cryogenic liquid until the temperatures stabilized and leak tested; and then, returned to the highest ambient environmental temperature and leak tested. The test was continued for a total of 30 temperature cycles, or until excessive leakage was encountered, whichever occurred first. With joints for high temperature applications, the joint was: soaked at the lowest expected ambient environmental temperature and leak tested; flowed with high temperature gases until the temperatures stabilized and leak tested (if feasible); and returned to the lowest environmental temperature and again leak tested. These tests were repeated for a total of five cycles, or until excessive leakage was encountered, whichever occurred first.

l. Fluid Resistance and Compatibility Tests

Fluid resistance and compatibility tests were conducted to ascertain the storage capability with system fluids and simulated conditions of the actual system. This test applies primarily to joints for use in storable propellant systems. The joint was: leak tested with gaseous nitrogen; filled with system fluids for a specified time and again leak tested; flushed with the solution to be used in the system and leak tested with gaseous nitrogen; and then stored at the highest expected ambient environmental temperature (usually +160°F) for a period of 15 days with appropriate plugs installed at the inlets and outlets to the joint. The ambient gaseous nitrogen leakage test, fluid leakage test, flushing, and subsequent ambient leakage tests were repeated after one month. The joint was then disassembled and examined for corrosion, permeation, or other detrimental affects.

m. Salt Spray and Storage Tests

Tests were conducted with some joints to determine the affects of long term ambient storage conditions and salt spray. These tests were devised to ascertain any salt environment or electrolytic corrosion problems. Salt spray tests were conducted in accordance with MIL-STD-810, Method 509, or MIL-E-5272C, Procedure I, as appropriate for the particular application. Leakage tests were conducted before the test and after the test was completed. Fastener torque values were also determined before and after the test. The joint was then disassembled, inspected, and visually examined to determine any deleterious effects.

n. Vibration Tests

Vibration tests were conducted with some joints to determine the suitability of the joint for various applications and to ascertain any detrimental affects upon the joints and gaskets. These tests were conducted in

accordance with MIL-STD-810, Method 514 for ground launched missiles, or MIL-E-5272C, Procedure XIV, as appropriate for the particular application. Leakage tests were conducted before the test and after the tests were completed. Fastener torque values were also determined before and after the tests. The joint was then disassembled and all parts were inspected and visually examined to determine any detrimental distortion, deterioration, or other affects.

o. Vibration-Low Temperature Tests

Vibration tests were also conducted at low temperature with some joints to be used in cryogenic applications to determine the affects upon the joint and gasket. Vibration levels were as specified in the previous section and the test fluid was liquid nitrogen. The leakage tests were conducted before the test, at specified time intervals during the test, and after the test was completed. The fastener torque values were also determined before and after the test. The joint was then disassembled and all parts were inspected and visually examined to determine any detrimental distortion, deterioration, or other affects.

p. Burst Tests

Preliminary burst tests, using heavy duty flanges, were conducted with some gaskets to determine the ability of the gasket to withstand burst pressure at various flange separations without extrusion, gross leakage, or gasket failure. The test joint was properly assembled with appropriate final bolt torque values and instrumented to determine any separation of the flange faces as shown on Figure No. 45. Hydrostatic pressure was then slowly applied to the joint up to the nominal burst pressure (1.33 or 1.50 times proof pressure depending upon the application) or until either failure or gross leakage occurred. When either failure or gross leakage occurred, the pressure and any separation of the flange faces were noted. The joint was then disassembled. The gasket was examined for detrimental distortions or the cause of failure or gross leakage and the flange surfaces were examined for damage. If the flange faces were damaged, they were resurfaced to the requirements of the gasket. If there was no apparent damage to the flange surfaces, a new gasket was installed and the bolts were properly tightened to 75% of specified torque. The flange was then instrumented as shown on Figure No. 45 and hydrostatic pressure was slowly applied until failure or gross leakage occurred. This sequence was repeated using a new gasket and properly surfaced flange faces until failure, gross leakage, or extrusion occurred at very low pressures. Final bolt torque values were decreased in increments of 50%, 25%, 10% and zero.

The pre-qualification burst tests with minimum weight flanges were conducted to determine the ability of the test cell or joint design to withstand the design burst pressure without failure or gross leakage. All these tests were conducted with the joint assembled with the specified final torque values applied to the bolts or fasteners. Hydrostatic pressure was slowly applied up to nominal burst pressure. If no gross leakage or permanent distortion was observed, the pressure was relieved and the fastener torque values were noted. The joint was then disassembled and all parts

TEST NO.	FINAL BOLT TORQUE VALUE
1	100% of Specified
2	75% of Specified
3	50% of Specified
4	25% of Specified
5	10% of Specified

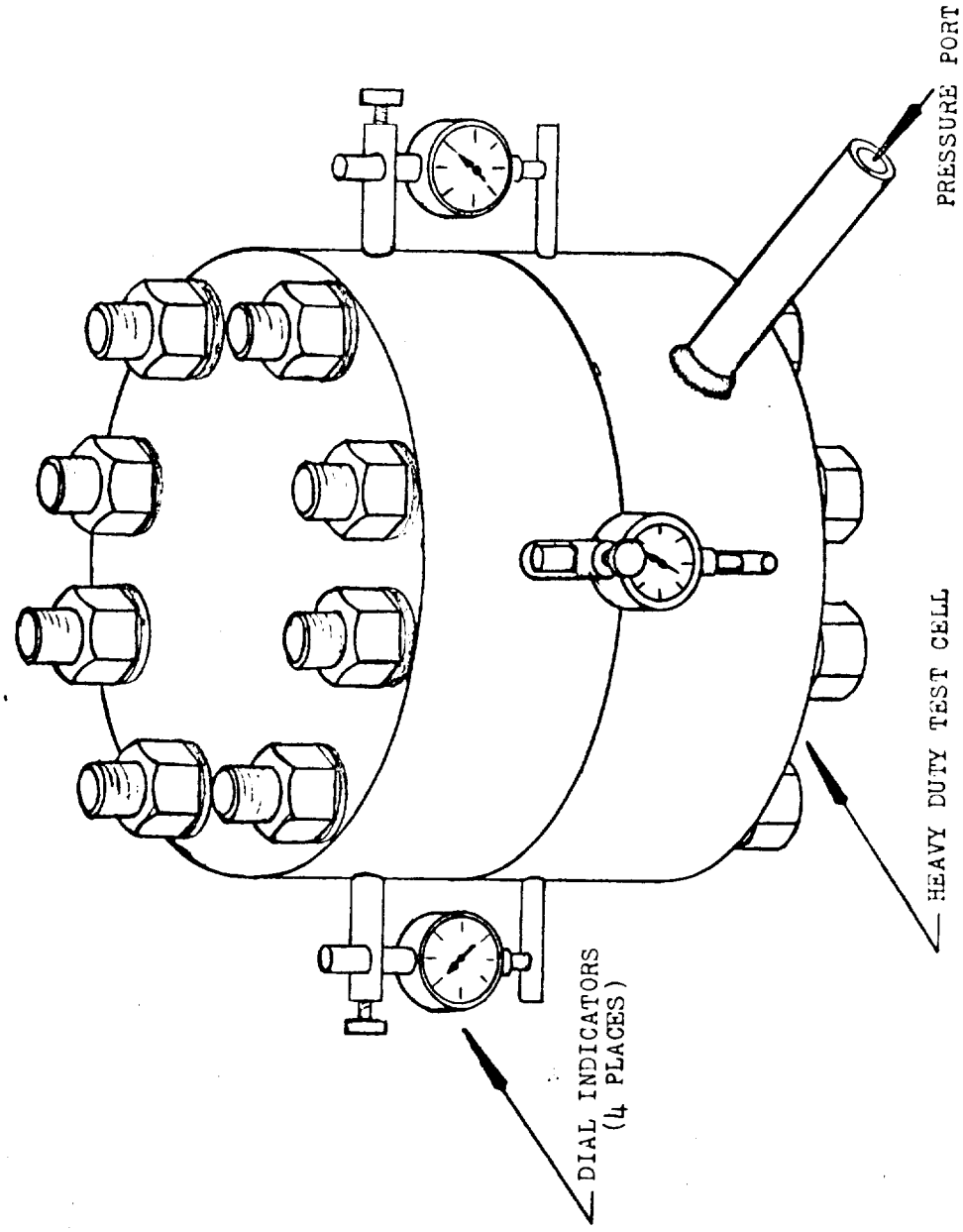

Figure 45
Preliminary Burst Test Set-Up

examined to determine any permanent distortions or other detrimental effects. If none were present, the joint was reassembled with a new gasket and hydrostatic pressure was applied slowly until the joint failed or gross leakage occurred. Subsequently, all parts were examined to determine permanent distortions or the causes of gross leakage.

2. Component Tests

Data were collected for joints in actual hardware during component development and system tests. The component development tests were devised for the particular component to determine its acceptability for the particular application. Data collected from system tests are more in the form of reliability data and do not locate or define the problem area. The component development tests conducted were as follows:

a. Proof Tests

Proof tests were conducted to determine any detrimental permanent distortion. Test media was water in some tests and dry gaseous nitrogen in other tests, depending upon the application. Proof test pressures were also as specified for the particular application and the location of the component in the system. Leakage was determined by submersion in water (for nitrogen proof tests) or by visual observation (for water proof tests). If no leakage or detrimental distortions were noted during this test, tests as defined in the following discussions were conducted. If leakage or detrimental distortions were noted, the component was disassembled and all parts were inspected and visually examined to determine the extent of permanent distortions or the cause of leakage.

b. Leakage Tests

Leakage tests were conducted to determine the effectiveness of the various sealed joints in the components. The test media used was dry gaseous nitrogen or gaseous helium, depending upon the application. Leakage test pressures were also dependent upon the application, but were usually 75% of proof pressure. Leakage test methods were dependent upon the component design and the application. Two leakage test methods were used for the single seal joint designs; these were water submersion, and a vacuum chamber with a mass spectrometer attached. In the water submersion method, the component was submerged in water and pressurized in even increments to the leakage test pressure for five minutes at each incremental pressure and the leakage was collected into a calibrated inverted beaker. The second method consisted of installing the component in the vacuum chamber, to which the mass spectrometer was attached, pressurizing the component in increments to the leakage test pressure for five minutes at each incremental pressure and measuring leakage with the mass spectrometer. Two leakage detection devices were used for the dual seal joint designs. After the external seal was satisfactorily tested (by submersion method with pressure between seals), the inter-seal cavity was connected (through a leak test port) to either a flowmeter or a mass spectrometer, depending upon the component and the application.

c. Low Temperature Leak Tests

For those components designed for cryogenic applications, low temperature leakage tests were devised to determine the affects of low temperature upon maintaining the initial seal at each joint in the component. These tests usually consisted of low temperature (cryogenic) "bleed-in," then pressurization to leakage test pressures with gaseous helium or nitrogen depending upon the temperature of the liquid used in the application. Leakage tests were conducted at ambient, low temperature, and again, at ambient temperature.

d. Thermal Cycling Tests

Thermal cycling tests were conducted with the components to determine if any deterioration of leakage rate capabilities occurred as a result of cycling. The temperatures involved in the thermal cycling tests depended upon the expected operating temperature of the component. For components designed for low temperature applications, the component was cycled between the highest expected environmental temperature and the lowest operating temperature. Leakage tests were conducted at each temperature during the cycle. The number of cycles was determined by the particular application, but usually consisted of a minimum of five cycles. For the components designed for high temperature applications, the component was cycled between the lowest expected environmental temperature and the highest expected operating temperature. Leakage tests were conducted at each temperature during the cycle. Again, the number of cycles was determined by the application, but usually consisted of a minimum of five cycles.

3. Systems Tests

After the component was sufficiently developed, it was installed into the actual system. Leakage tests of the complete system were conducted at low pressures to be reasonably assured that all joints were satisfactory before an actual test of the system. The method used for leakage detection depended upon the system, application, and joint design. The test usually consisted of applying 50 psig to the system and measuring pressure decay for a 10 minute period. If pressure decay was within pre-established limits, the system was considered satisfactory for the actual tests. If the pressure decay was above pre-established limits, each joint in the system was checked further to locate the predominant leakage points. Corrective measures were taken to reduce or eliminate the predominant leakages and the system was rechecked. Leakage was determined during the actual system test by visual observation. After the actual test, the system was again leak tested as previously described.

C. TEST RESULTS

1. Laboratory Tests

a. Loading Tests

Results of the loading tests conducted with the gaskets tested are presented on Figures No. 46 through No. 61. Where possible, the calculated sealing stresses are also included in the figures. The stress values at final loading conditions are based upon the deformed width and mean diameters of the sealing area of the gaskets when removed from the assembly. The primary concern was to determine whether sufficient sealing stresses are obtained during loading and at what point of flange separation the sealing stresses dropped below the minimum required levels necessary to maintain sufficient sealing characteristics. Loading values are presented in load per unit length of sealing surface (lb/in. circumference) because the proprietary gaskets are designed for a range of diameters using the same cross-sectional configuration and should provide the same unit loading if the effects of the radius of curvature are discounted. The loading test results justifies the use of these units. Variations in load values (i.e., total load/πx sealing surface diameter) for different diameters of the same cross-sectional configuration was found to be much more a function of machining tolerances (groove depth, gasket free height, and gasket leg or spring thickness variations) than diameters. This was valid at least for the range of diameters tested of a particular cross-sectional configuration.

Loading data for normal squeeze on elastomeric O-rings were previously discussed. The data were available in existing literature[27] and were presented on Figure No. 12. For temperatures within the useful range of the material (where the elastic properties are still retained), the loading values are low in comparison with the other types of seals. The O-rings are not satisfactory with normal squeeze for very low temperature service because the elastomeric materials become brittle at very low temperatures (below approximately -100°F). It has been proposed[28][29] that high squeeze (70% to 90%) could be used to provide static seals for very low (cryogenic) temperature applications. Loading tests were conducted using 0.139-in. cross-section Silicone and Teflon O-rings and the data are included on Figure No. 12. With the high squeeze (90%), most of the elastic capabilities are lost. Permanent set (compression set) can be equal to the squeeze with some elastomers and

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(27) Seal Compound Manual, Parker Seal Co. Catalog C5702, July 1964

(28) Weitzel, D. H., Robbins, R. F., Ludtke, P. R., and Obori, Y., Elastomeric Seals and Materials at Cryogenic Temperatures, National Bureau of Standards (This is a continuing series of reports initiated in 1960 that are available through the Defense Documentation Center for Scientific and Technical Information (DDC), Cameron Station, Alexandria, Virginia 22314)

(29) Timmerhaus, K. D. (Editor), International Advances in Cryogenic Engineering, Annual Proceedings of the Cryogenic Engineering Conference, Plenum Press, 1960 through 1965

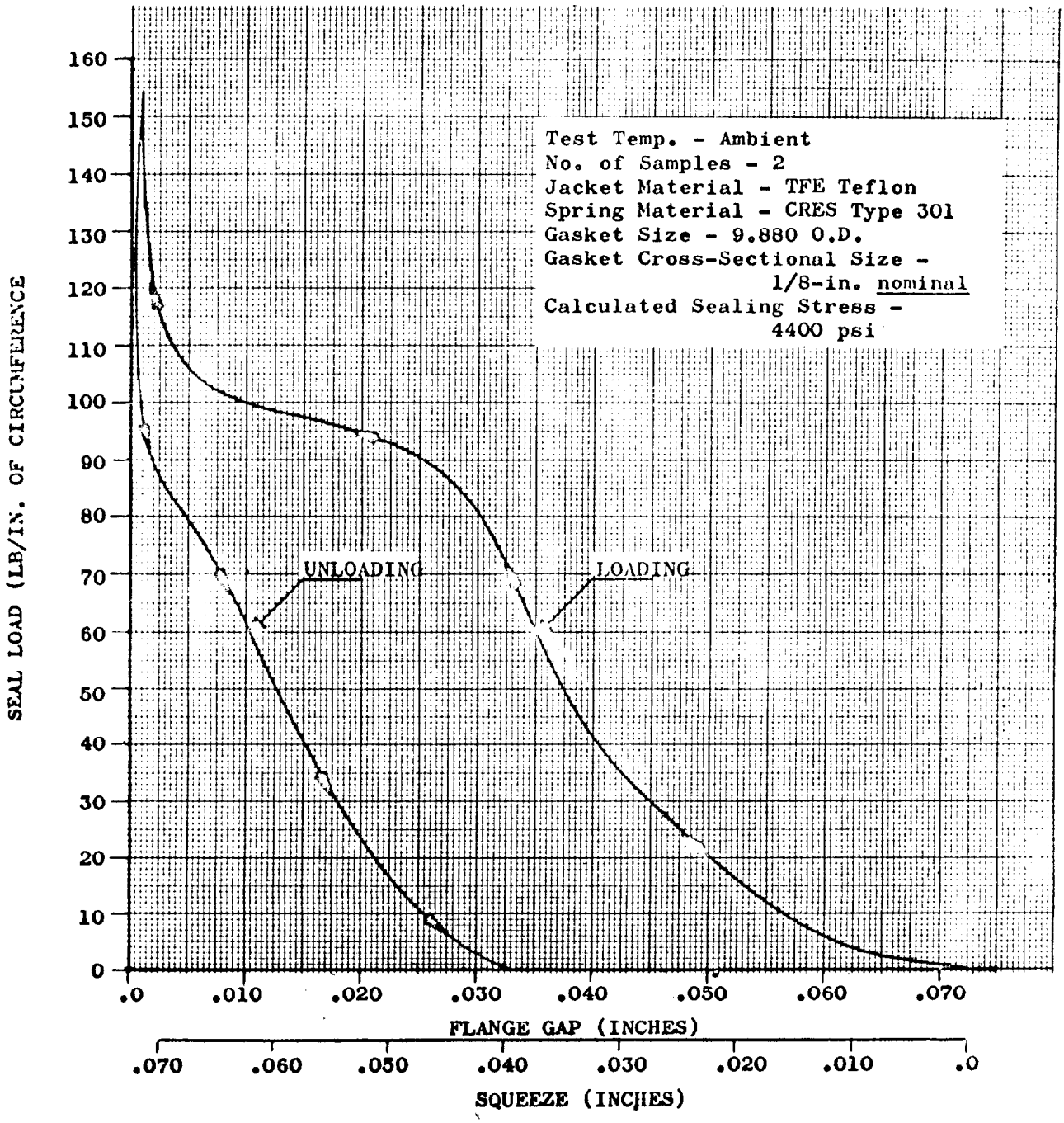

Figure 46
Load vs. Compression, Raco Engineering Division,
R. E. Creath Co., P/N 10030

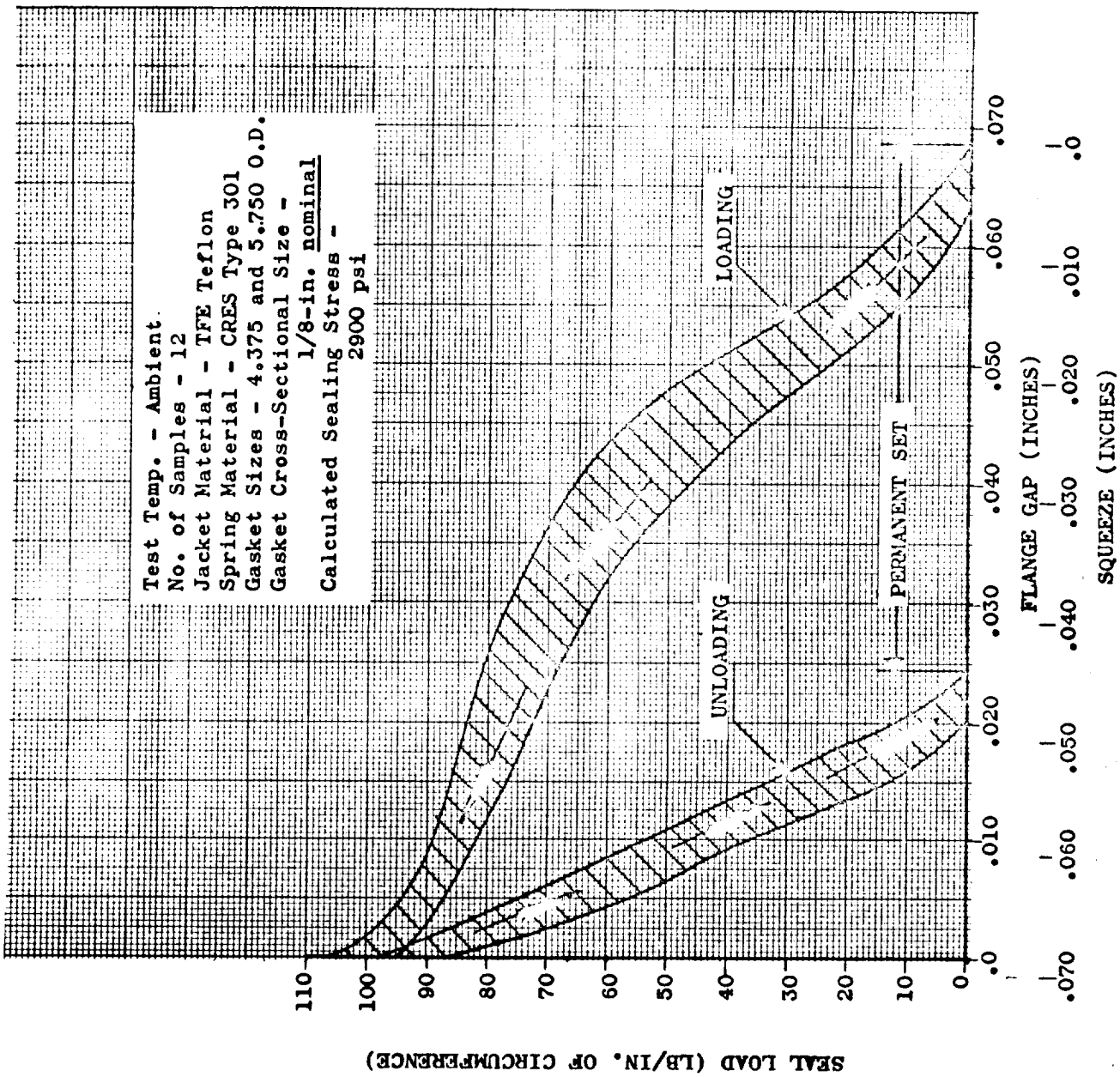

Figure 47
Load vs. Compression, Raco Engineering Division,
R. E. Creath Co., P/N 10011-4.375 through -5.750

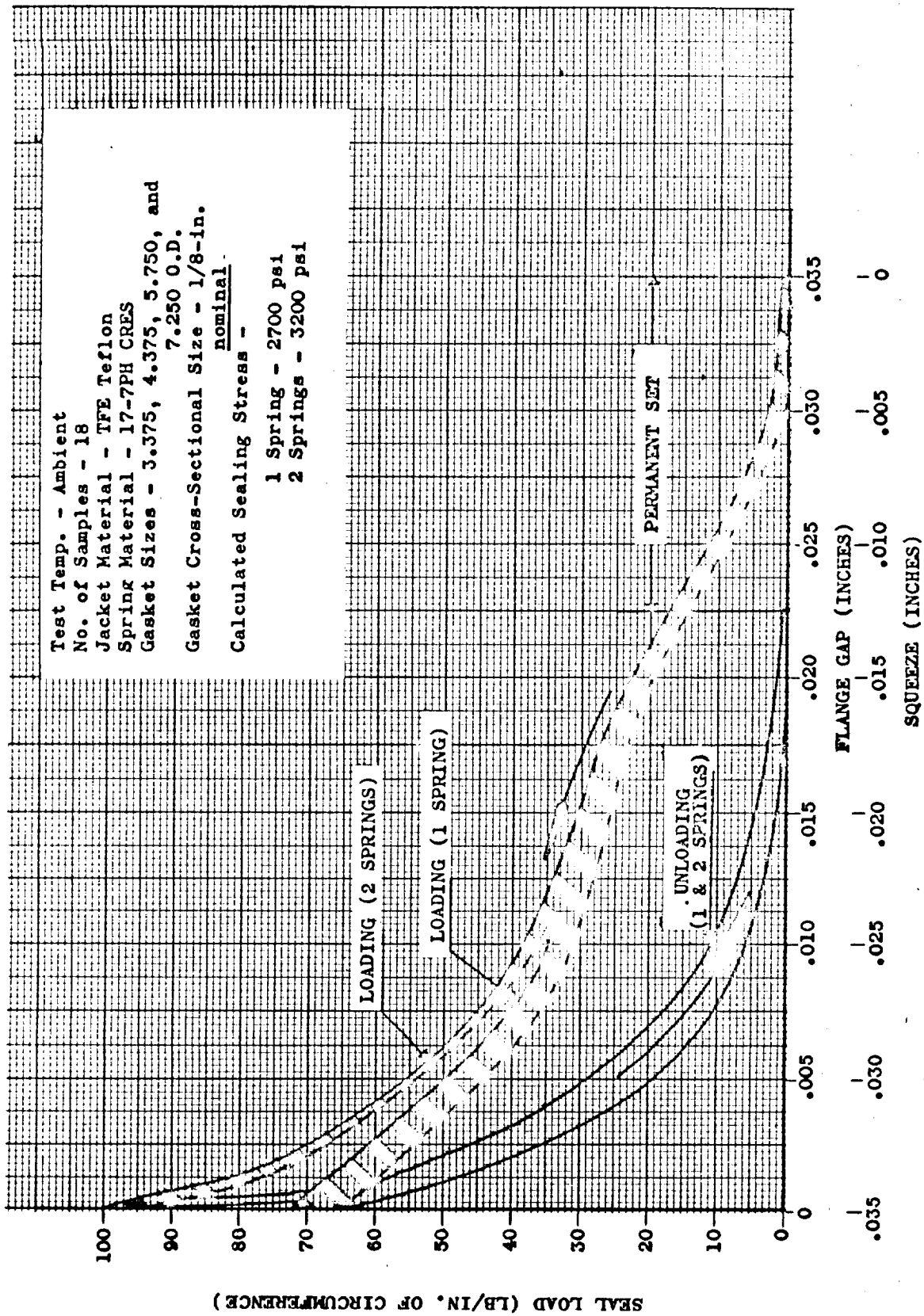

Figure 48
Load vs. Compression, Reid Enterprises, Inc. (Omniseal)
P/N R10205 (Internal Pressure) and R10215 (External Pressure)

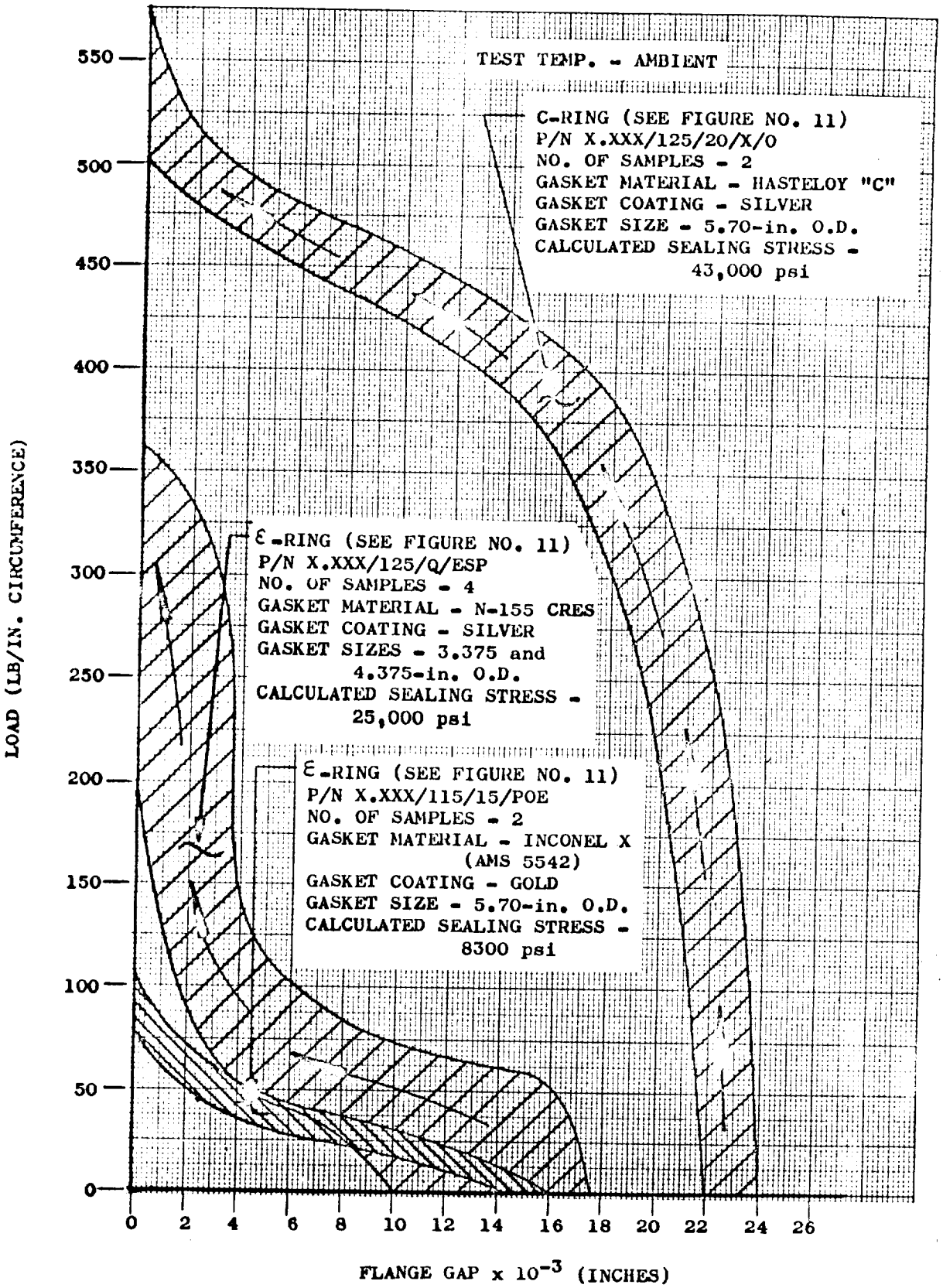

Figure 49

Load vs. Compression, Pressure Science, Various P/N's

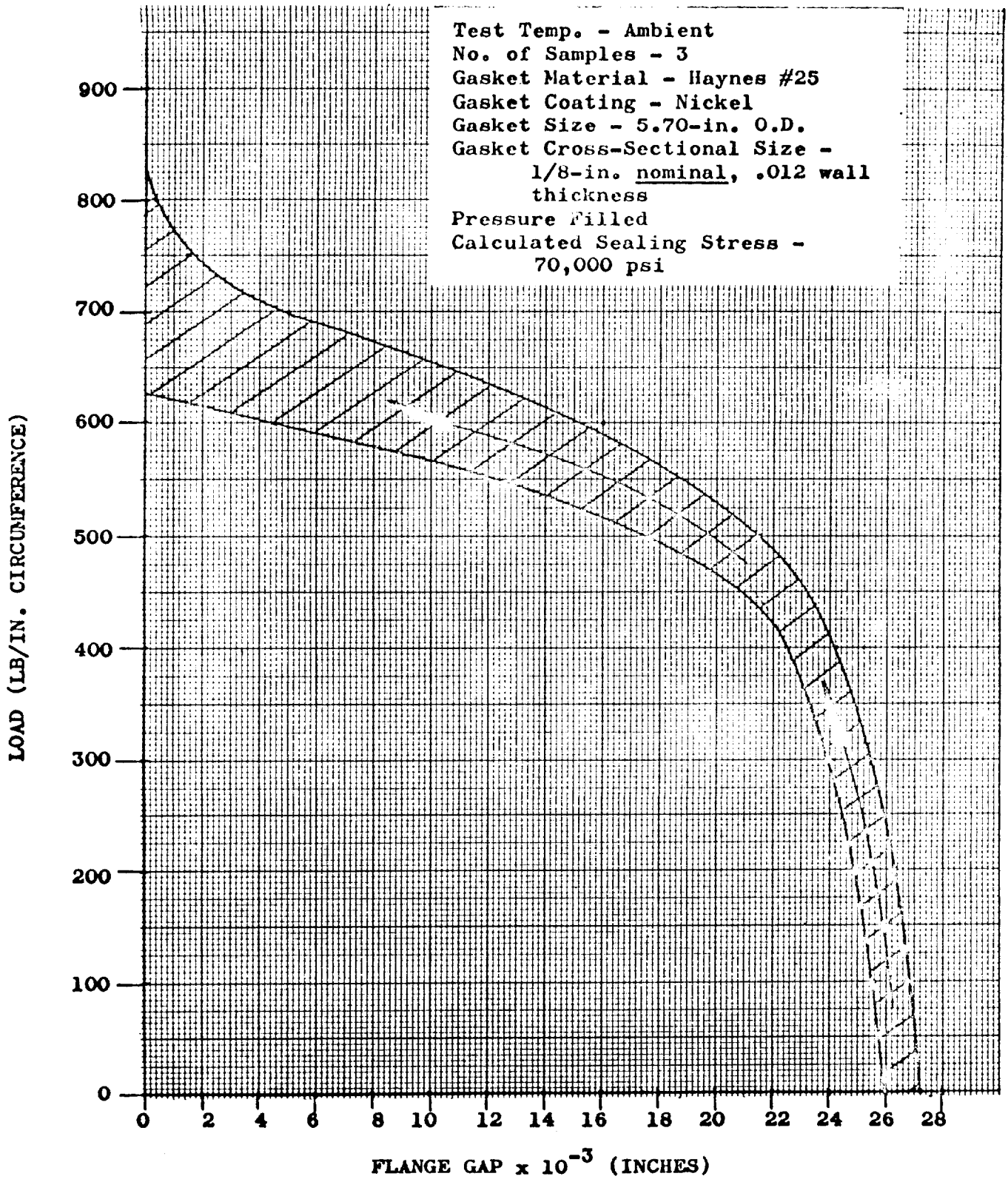

Figure 50
Load vs. Compression, Advanced Products Co., Metal "O"-Ring, P/N 5.695-7/012-9-NP2025PF

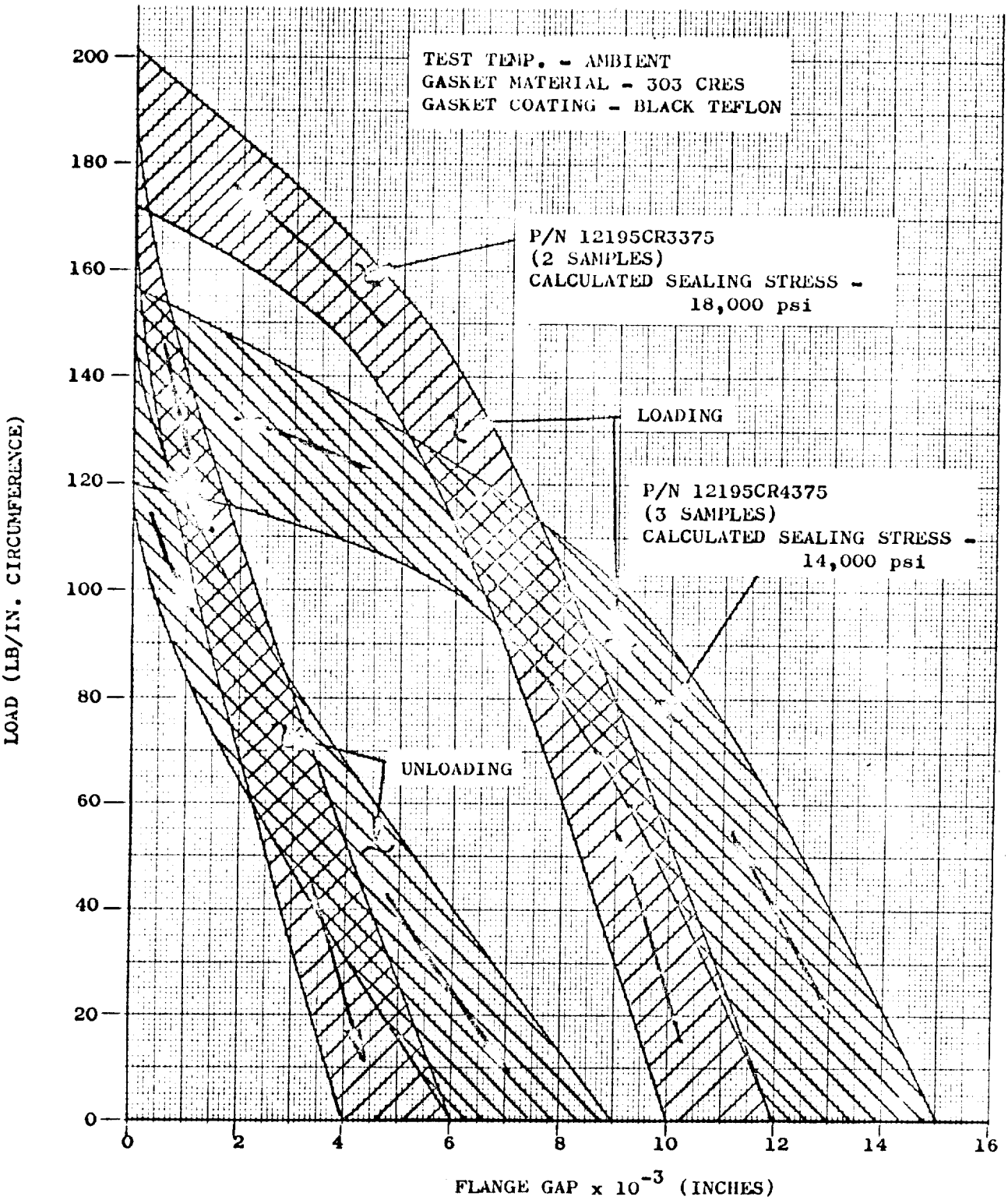

Figure 51
Load Vs. Compression, Harrison Manufacturing Co., K Seal

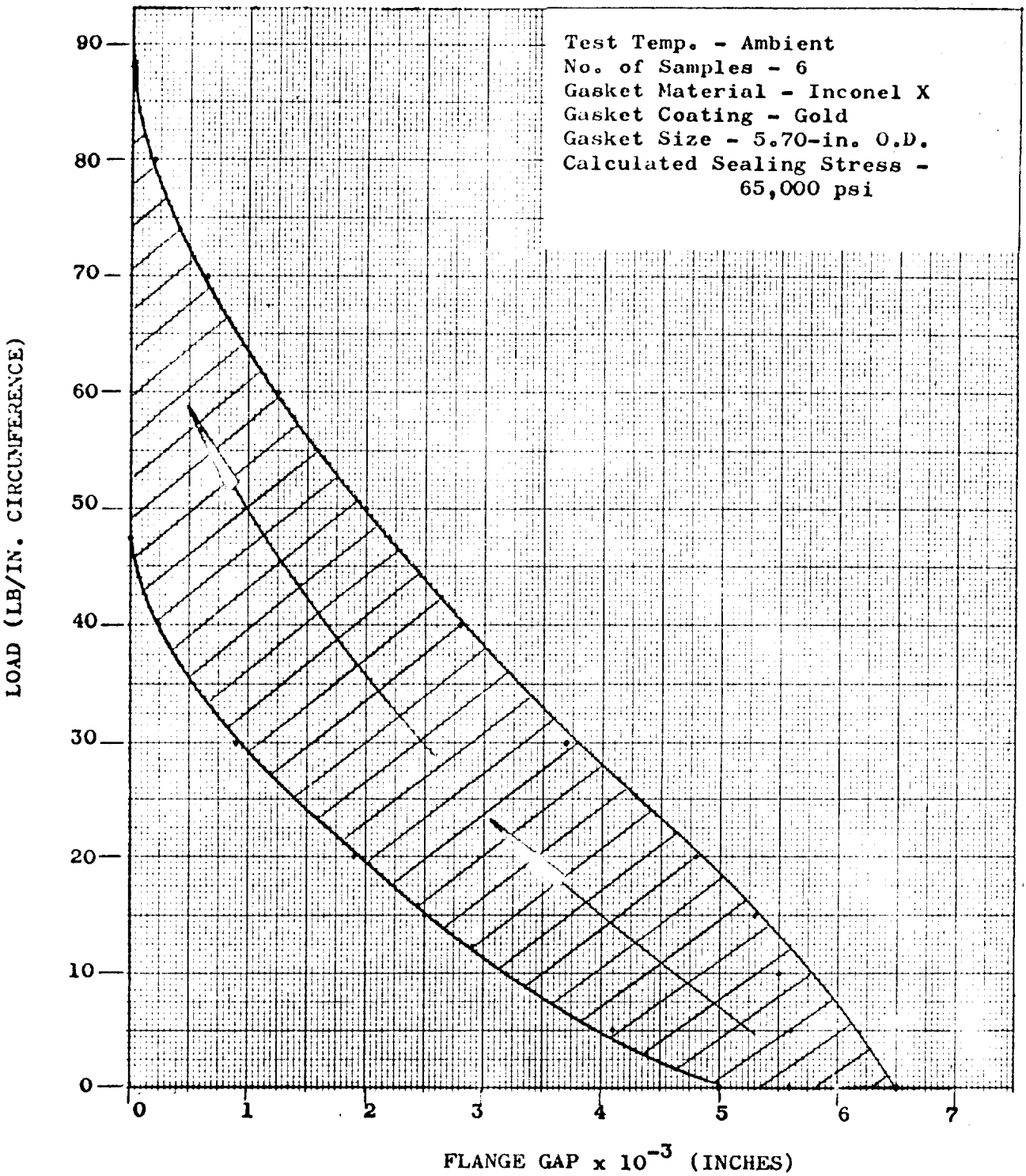

Figure 52
Load vs. Deflection, Hasket Seal Division P/N 18111

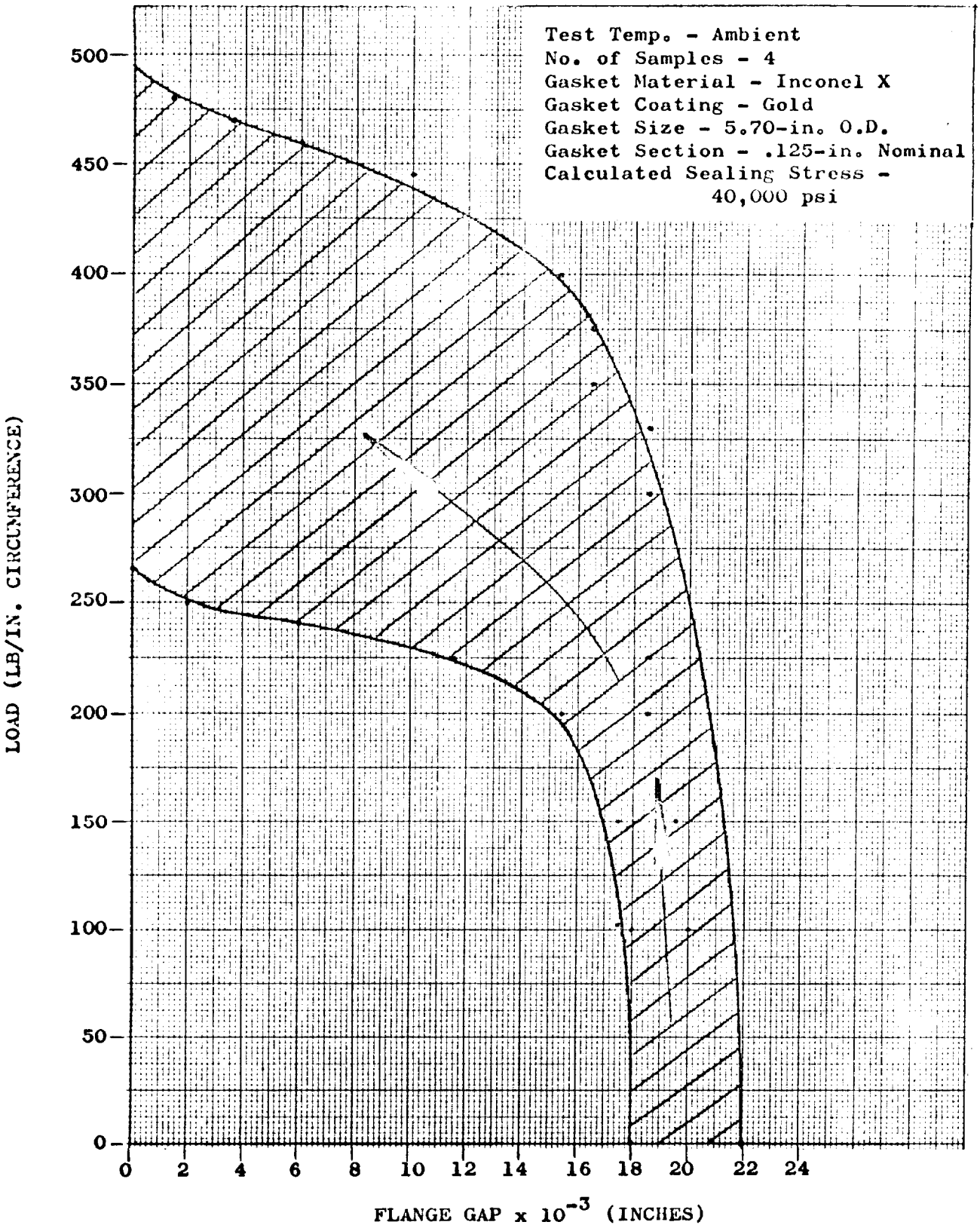

Figure 53
Load vs. Compression, Tetrafluor "TV" Seal,
P/N TVM-4-182-8-5

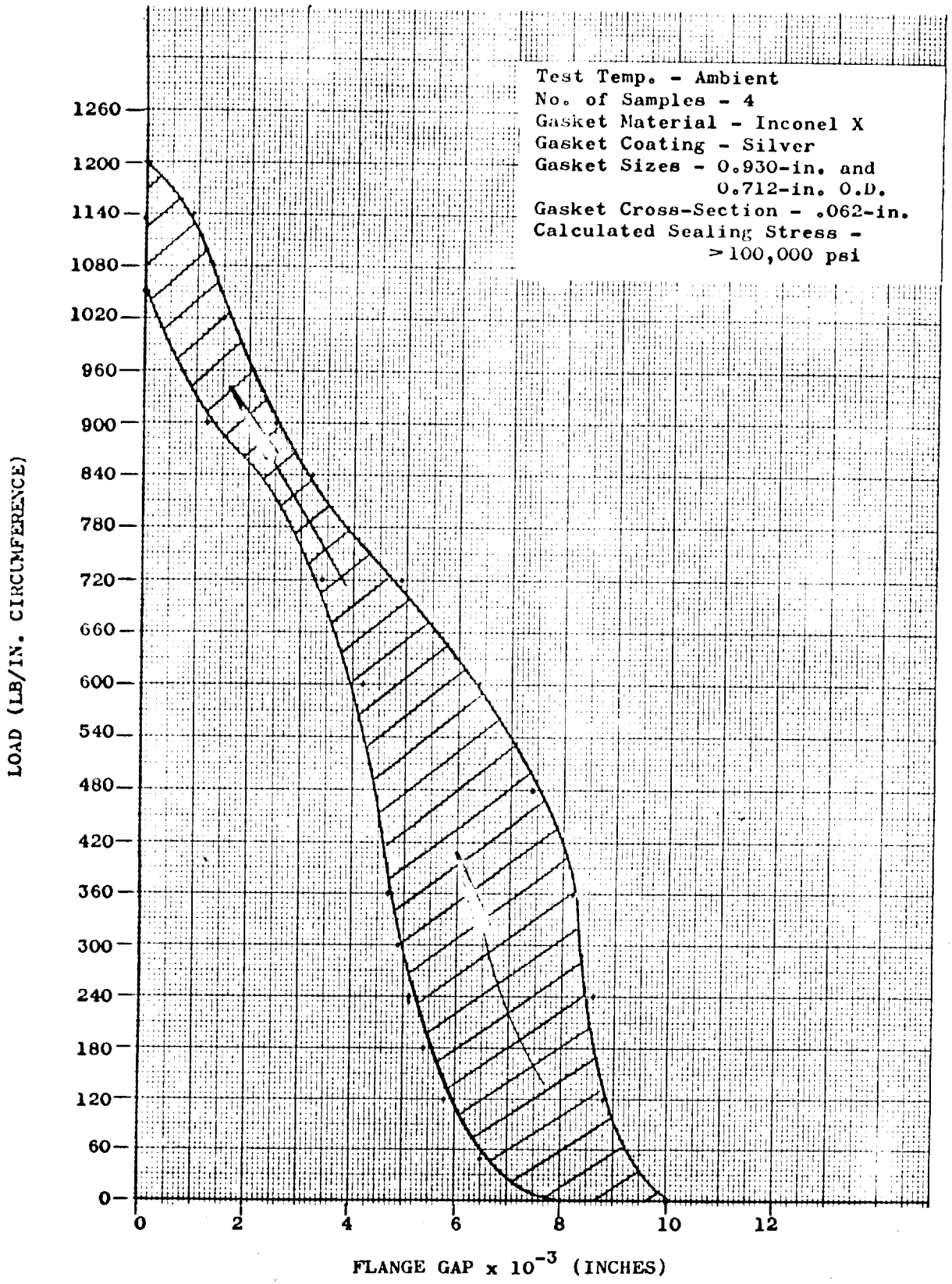

Figure 54
Load vs. Compression, Tetrafluor "TV" Seal,
P/N TVR-19-8-4 and -16-8-4

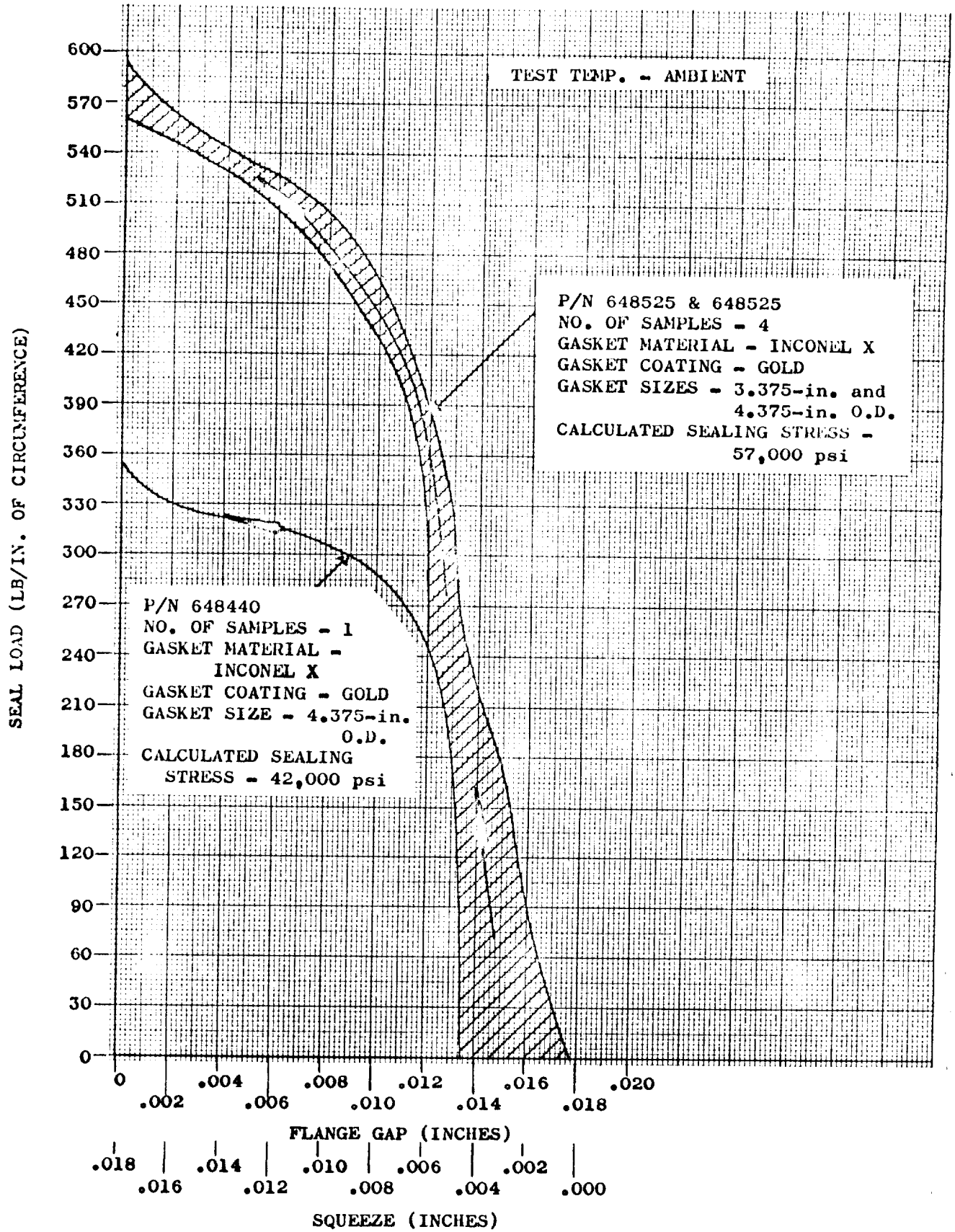

Figure 55

Load vs. Compression, Parker Seal Co., Meta-V-Seals, P/N 648524, 648525 and 648440

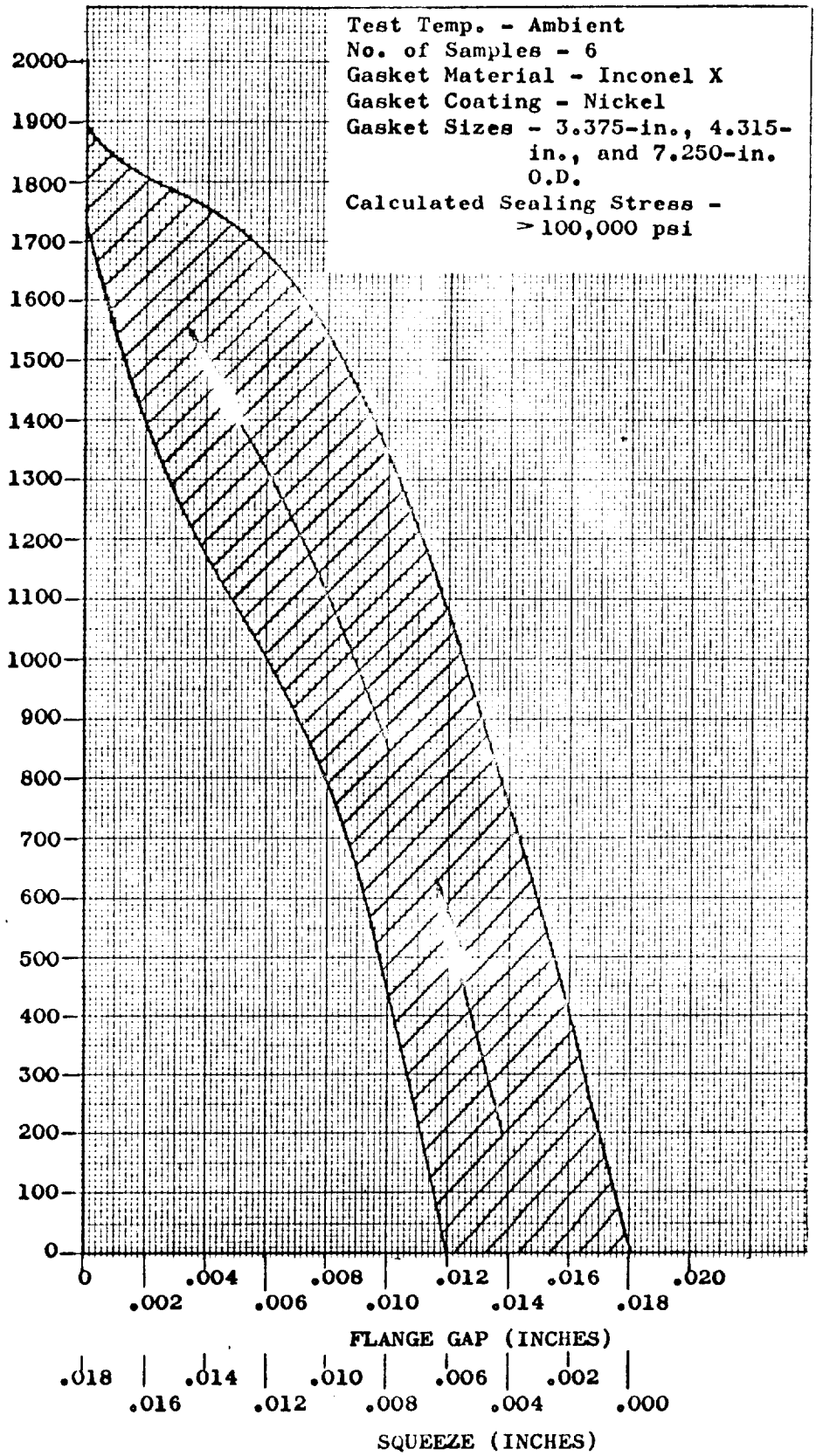

Figure 56
Load vs. Compression, Astrodynamics Seals Division of Aerospace Components Corp., P/N S-9033, S-9041 and S-9060

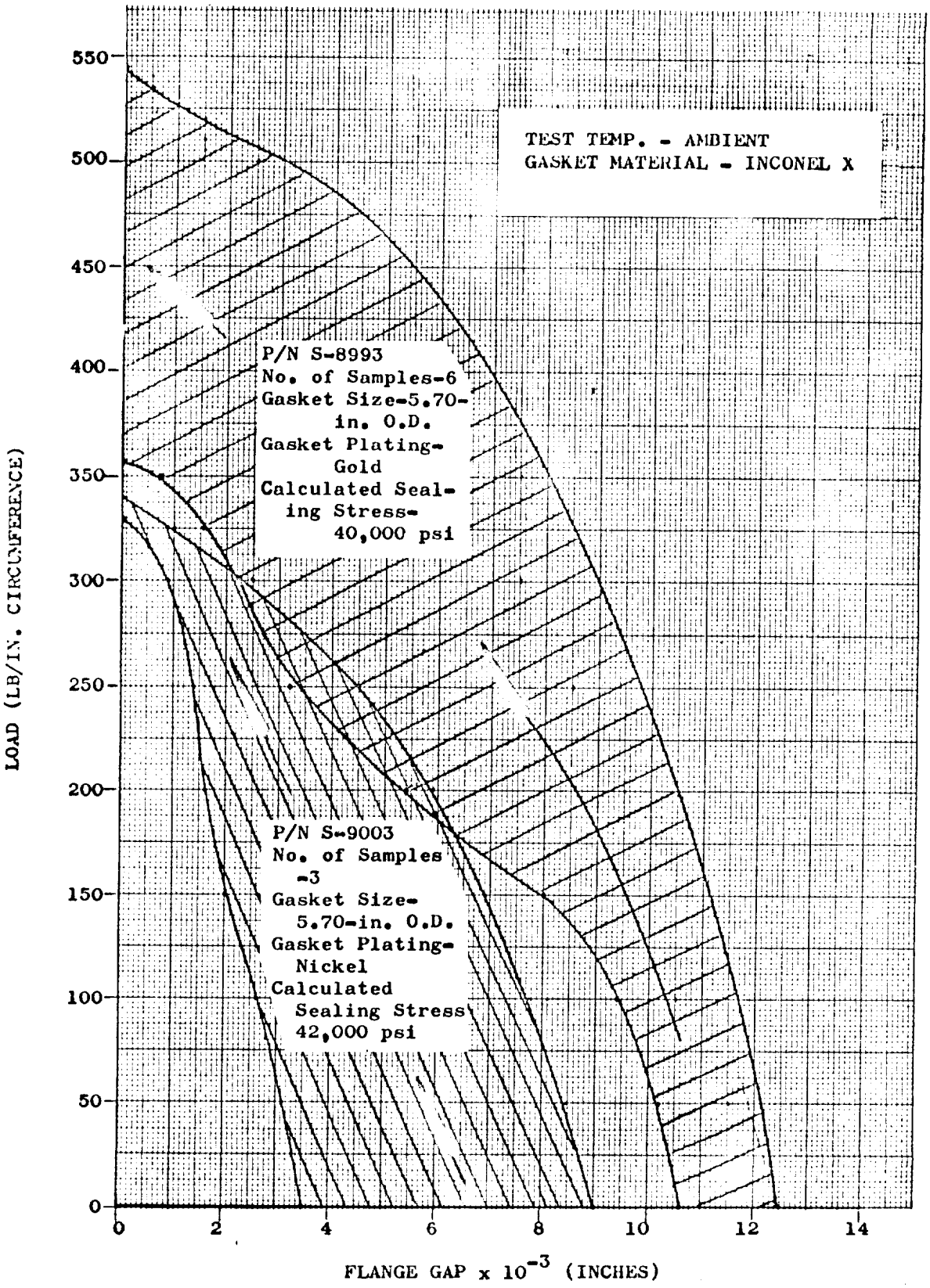

Figure 57
Load vs. Compression, Astrodynamics Seals Division
of Aerospace Components Corp., P/N S-8893 and S-9003

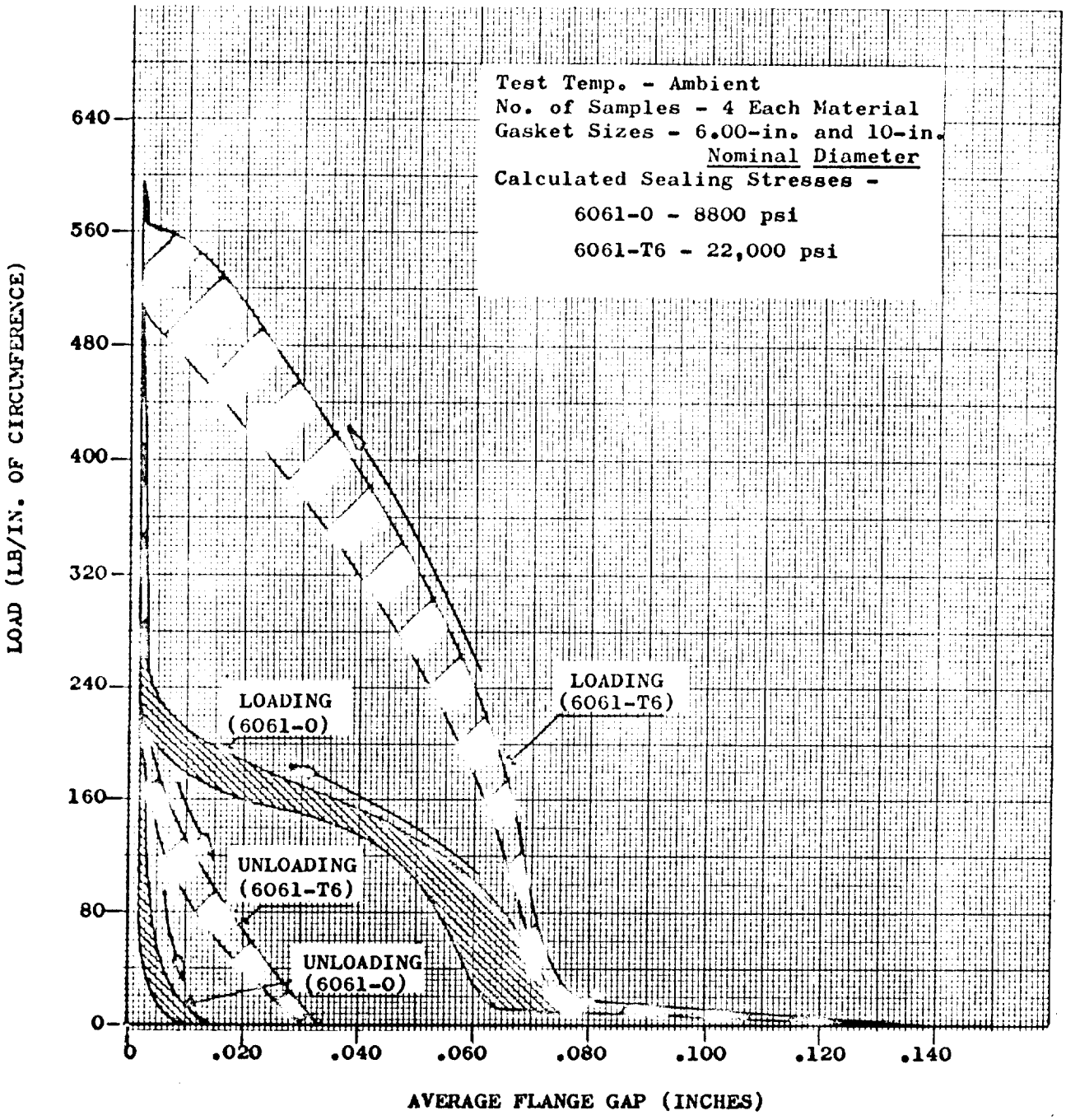

Figure 58
Axial Load vs. Compression, Single Conoseal Gaskets
.040 x .275-in. Cross-Section (6061-O and 6061-T6)

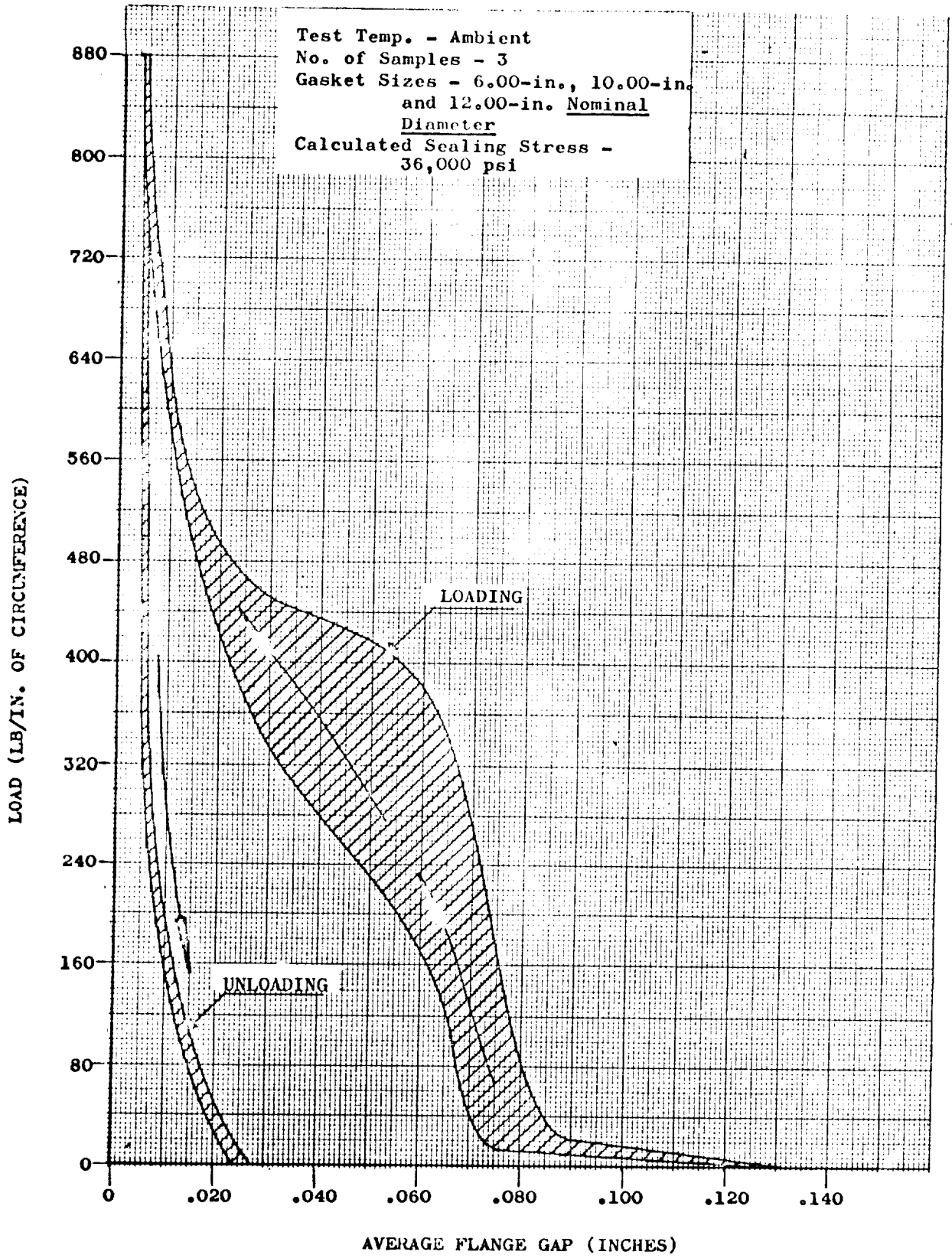

Figure 59
Axial Load vs. Compression, Single Conoseal Gaskets, .040 x .275-in. Cross Section (321 CRES Annealed)

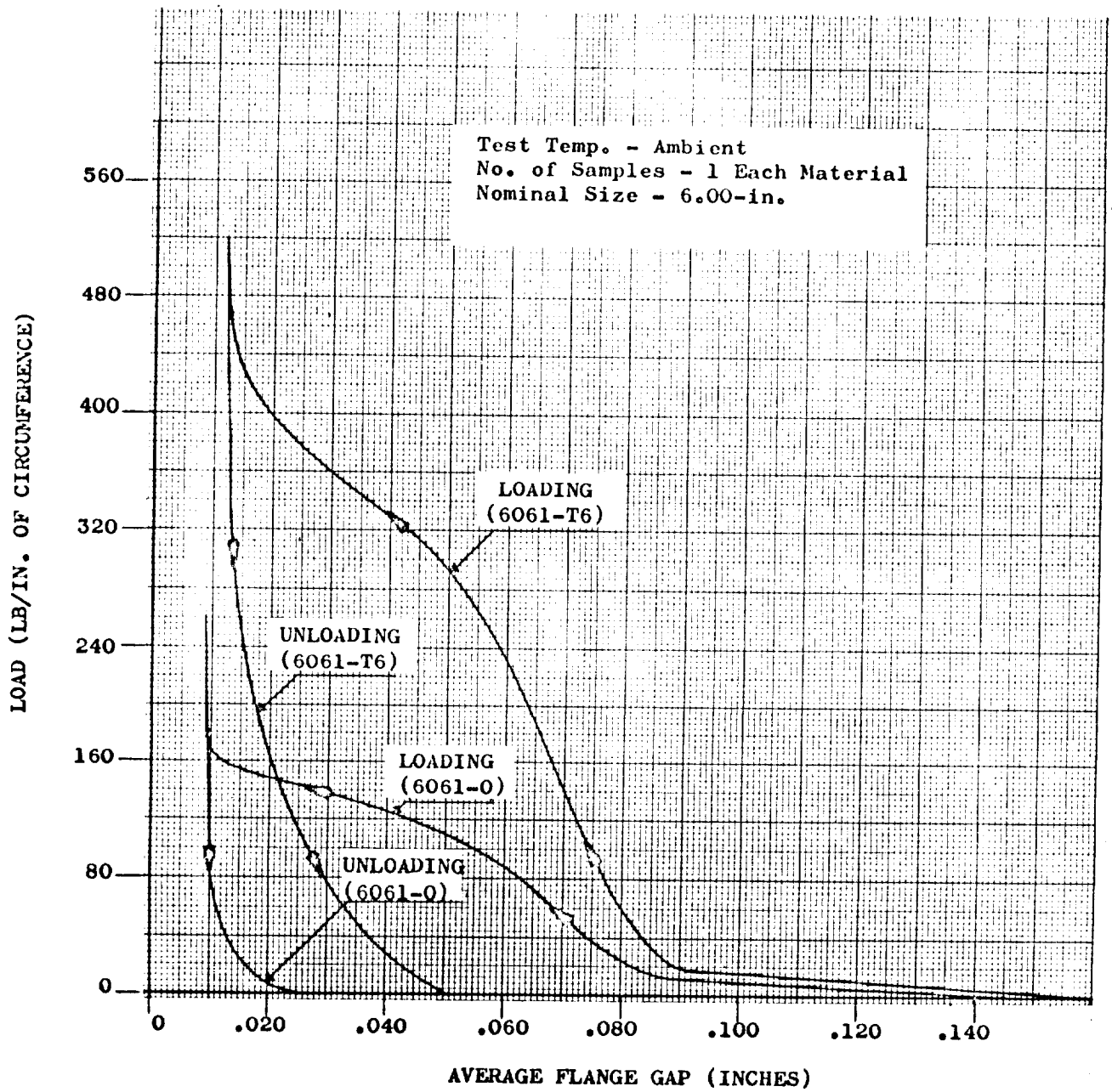

Figure 60
Axial Load vs. Compression, Dual Conoseal Gaskets, .040 x .275-in. Cross-Section (6061-0 and 6061-T6)

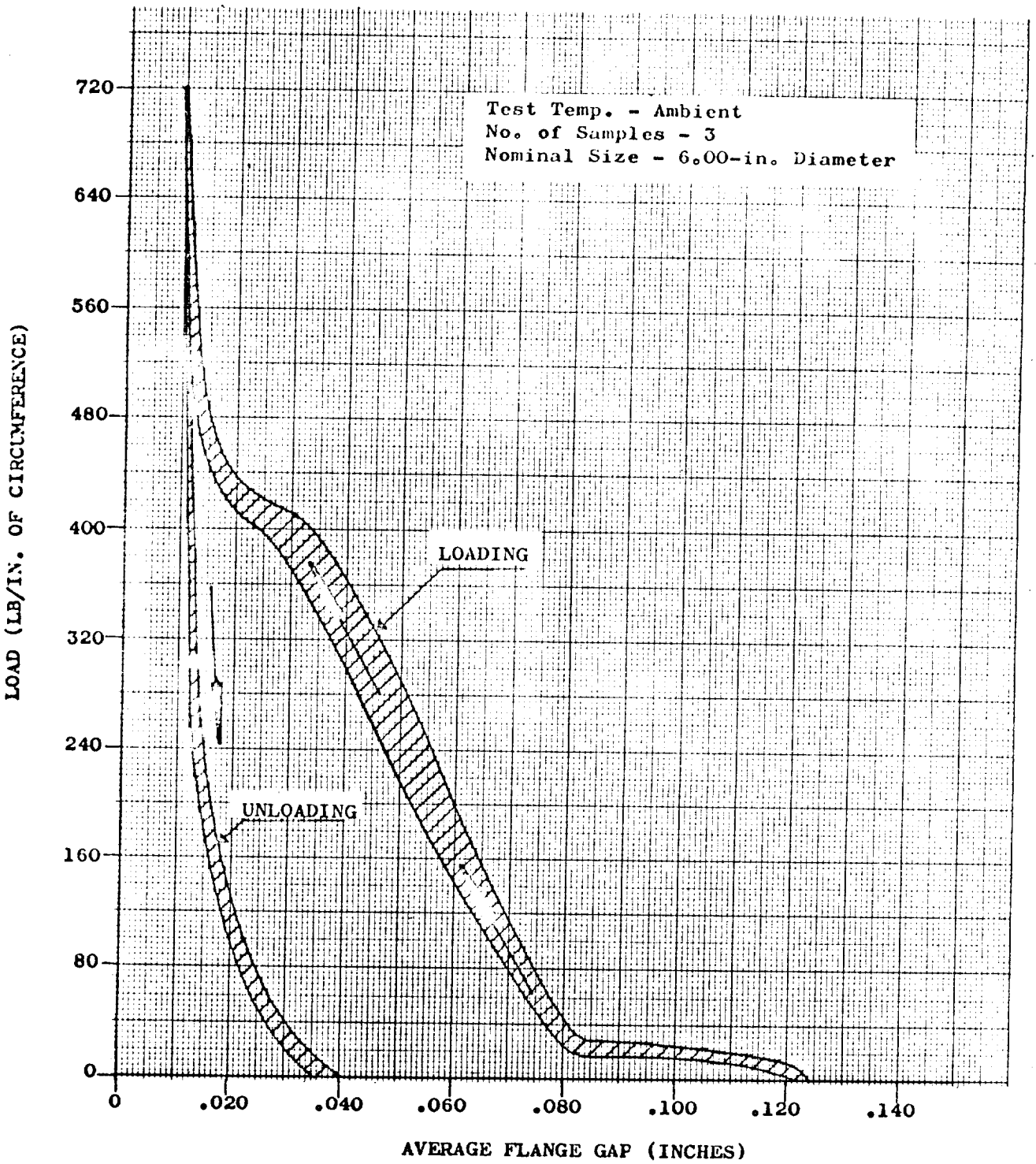

Figure 61
Axial Load vs. Compression, Dual Conoseal Gaskets, .040 x .275-in. Cross-Section (321 CRES Annealed)

plastics. The compression set with the Teflon O-rings with 90% squeeze was 95% and with the Silicone O-rings, 90%. The Teflon breaks down at approximately 75% squeeze.

All the "pressure-energized" gaskets were permanently deformed (set) during the loading tests. Typical examples are shown on Figures No. 47, No. 48 and No. 51. None of these gaskets returned to their initial free height when they were unloaded. Springback varied from approximately 30% to 60% (of initial squeeze) for the plastic-jacketed gaskets and from approximately 10% to 90% for the all-metal gaskets. This is not a serious problem with the plastic-jacketed gaskets because sufficient sealing stresses are maintained up to .010-in. separation of the flanges.

The initial squeeze on the Raco seals varied from .075-in. to .060-in. and, after loading, the minimum springback was .020-in. The initial squeeze on the Omniseal was .035-in. to .030-in. and, after loading, the minimum springback was .017-in. The springback is usually very important with the all-metal gaskets, because the initial squeeze is small. Initial squeeze varied from .003-in. for one version of the Astrodynamics seal to a maximum of .025-in. for one version of the metal O-ring and the Pressure Science C-Ring. Usually, the gaskets with very low initial squeeze had the higher springback, which is essential if any flange deflection capabilities are to be attained. The gaskets with the highest initial squeeze had the least amount of springback. The metal O-rings and the Pressure Science C-Ring had the highest initial squeeze (.025-in.) but returned only .0025-in. (10% springback). The initial squeeze on the Pressure Science E-Ring, Parker V-seal, and Tetrafluor TV-Seal was lower, but these seals returned almost to their initial free height (90% springback).

The gaskets with the higher springback have a better chance for maintaining sufficient sealing stresses when flange deflections occur. However, the initial squeeze should be as high as possible and the loading must be sufficient to gain the initial seal. Ideally, the best gasket capabilities are attained with high squeeze and high springback. Unfortunately, this is very difficult to achieve without very high initial loading because of the characteristics of the materials.

The Omniseal jacket extruded between the spring coils in the single spring configurations. This would tend to cause variations in sealing stresses and sealing edge width around the gasket sealing edges. Compensation for this condition was possible by either increasing the preload so that the minimum sealing stress was attained, or the thickness of the plastic jacket under the sealing edge could be made thicker to "bridge" the gap between coils. The units tested apparently did this satisfactorily because the variations in compressed sealing edge widths were very small. However, this may require further investigation with other single spring configurations (cross-sections) of the Omniseal if problems arise in obtaining and maintaining a satisfactory seal. The multiple spring configurations contain opposite wound springs installed inside the outer spring. These

configurations eliminated the problem of extruding the jacket between coils, but there is some increase in the required loading (see Figure No. 48). Both of the plastic-jacket gaskets provided a sealing edge width greater than .020-in. (see Figure No. 62) and they were the best of the "pressure-energized" gaskets.

The all-metal, pressure-energized gaskets designed with flat sealing areas (i.e., Haskel seals and V-seals) seldom provided a sealing edge width equal to the width of the flat. The tolerance stack-up between the groove depth and the gasket free height prevents this unless all actual dimensions are on, or near the nominals. Therefore, the actual sealing edge width can be much less than the sealing edge flat. The range of deformed sealing edge widths are shown on Figure No. 62 for the gaskets tested. The E-ring actually provides a double sealing edge, but the loading in the second sealing edges is not applied until the flanges are nearly face-to-face.

The cross-sectional squeeze on the all-metal diametral gaskets varied from 0.002-in. to 0.018-in. depending upon the type of gasket, tolerance stack-up, and materials. The deformed sealing edge width varied from 0.002-in. to 0.030-in. depending upon gasket type, the gasket cross-sectional size, and the gasket materials. The average compressed sealing edge width of the Conoseal gasket was equal to approximately one-half of the gasket thickness. All of the diametral gaskets tested, except for the "Infundibular" gasket, start compression with the sealing edge a finite width (sharp corner on the gasket); therefore, the actual sealing stresses are assumed to be equal to the compressive strength of the gasket material from the start of compression until the gasket is seated. Axial springback characteristics in these gaskets are not as important as radial springback in most cases. However, they varied from 0.001-in. to 0.020-in. (5% to 30%). The radial springback characteristics are a function of the gasket cross-section and materials. This radial springback determines the capability of the gasket to accept radial and/or differential radial deflections in the mating flanges. Radial and/or differential radial deflection capabilities are a function of the column strength, the flexibility of the gasket section or legs, and/or the compression modulus of the gasket section.

The Gamah seal configuration must rely mostly upon the compressive modulus of the gasket material while the Nucoseal relies upon the column strength of the gasket legs and the compressive modulus of the support ring. The Infundibular gasket relies mostly upon the bending strength of the gasket section and the modulus of the material. The Conoseal gasket relies upon the column strength of the gasket material and the cross-section. For a particular Conoseal gasket cross-section, increasing the material strength will increase the radial (and axial) springback characteristics, but it also increases the radial (and axial) preloading required to gain the initial seal as was shown on Figure No. 58. In addition, the section tends to buckle as a column when the load is applied; therefore, radial springback is also dependent upon the buckling strength of the cross-section. The Grayloc seal would gain some capabilities in differential radial deflections from pressure application.

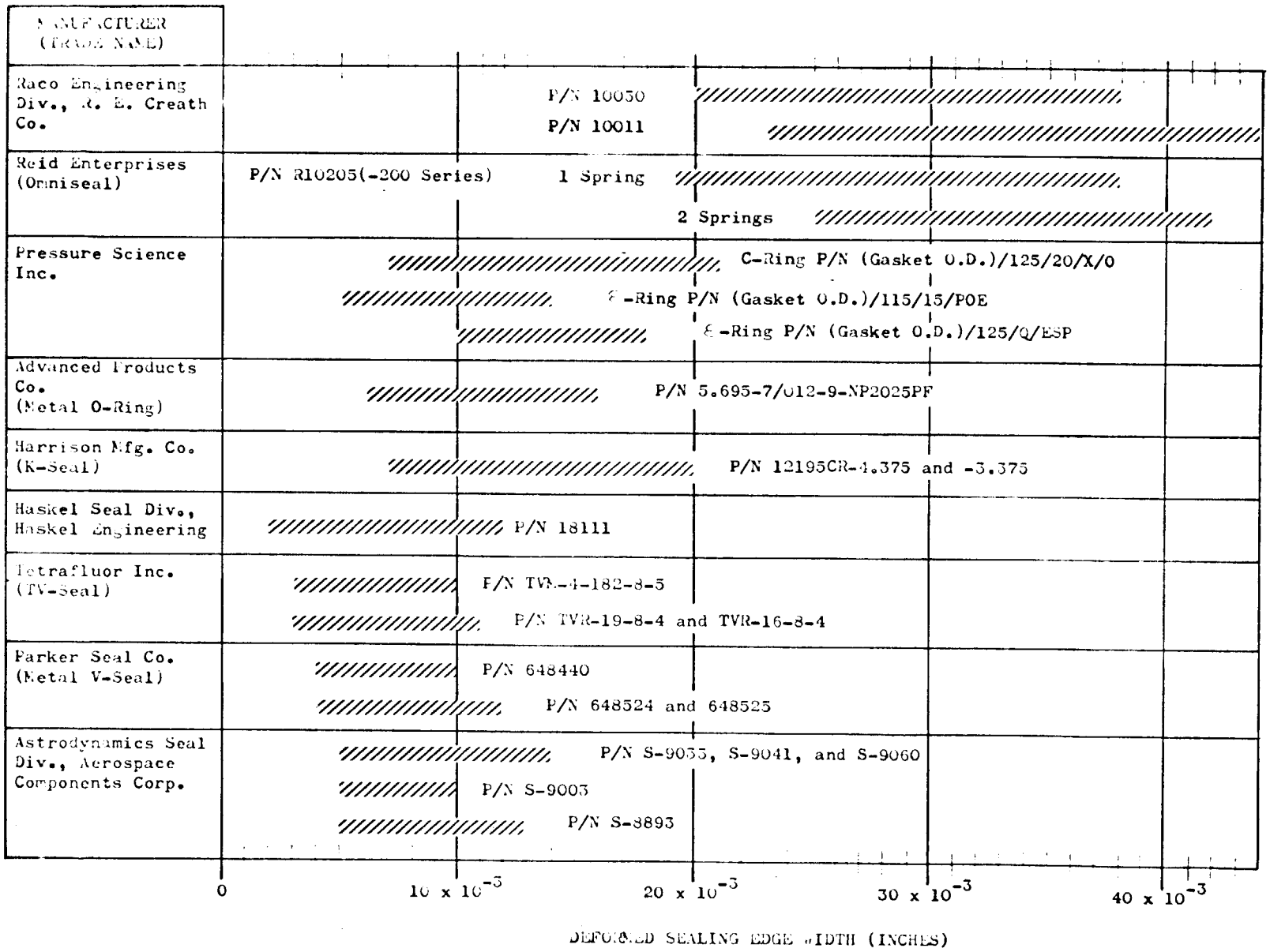

Figure 62
(Sheet 1 of 2)
Range of Deformed Sealing Edge Widths (In Direction of Leakage Flow) (Post Loading Test)

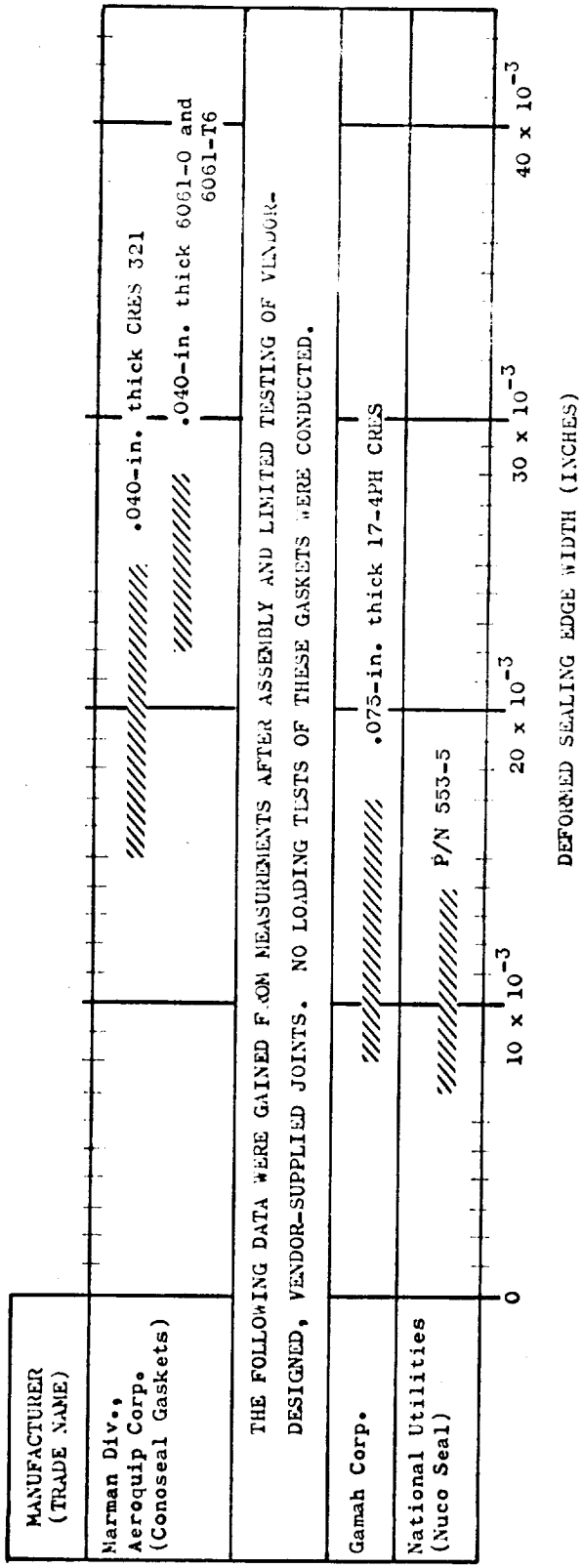

Figure 62
(Sheet 2 of 2)
Range of Deformed Sealing Edge Widths (In Direction of Leakage Flow) (Post Loading Test)

However, no laboratory tests were conducted with the Grayloc seal and the effects are not known. It is assumed that the Grayloc seal is approximately equal to the best of the diametral gaskets in this regard.

The squeeze, sealing edge width, and sealing stresses in the flat gaskets (includes spiral wound gaskets) are a function of the load, the groove configuration, and other variables. The spiral wound gaskets tested provided reasonably good squeeze (with accompanying high load) and springback characteristics with the groove depths used for the loading tests. It was determined later in the testing that the squeeze (and accompanying loading) was insufficient to provide a good seal with the configuration tested because when sufficient loading was applied to achieve a seal, springback was considerably reduced. Data was obtained for Figure No. 16 by discussions with suppliers as well as surveys of catalogues and literature. It is understood that very broad adjustments can be made in these values depending upon the method of manufacturing the gasket. It is not known what sealing characteristics would be provided with these configurations. No attempt was made to determine squeeze, springback, and other performance factors in the flat gaskets and solid section metal rings. However, it is known that the values for springback are very minute when used in standard flat faced flanges. They can be improved somewhat by using groove configurations that provide shear deformations in the gasket[30]. Also, increasing or decreasing the sealing edge width requires that the gasket loading be increased or decreased as a function of the contact area.

b. Proof Tests

One thing was common to all of the seals tested. If gross (visual) leakage occurred at proof pressures, very seldom would resealing occur as pressures were released. The O-rings and the plastic-jacketed materials would reseat more often than the all-metal gaskets. However, once the intimate sealing surface contact was lost, it usually required a new seal to pass the leakage test. In some cases with the all-metal gaskets, sufficient motion (wiping) occurred between sealing edges and the mating surfaces to require re-surfacing of the flange to regain sealing with a new gasket. This was necessary because the "wiping" caused radial scratches on the mating surfaces.

With the elastomeric O-rings and plastic-jacketed gaskets, the gasket material tends to "pack" into the groove as pressure is applied. This is a good condition in most applications. However, if deflections in the flanges cause a gap between the flanges and sufficient pressures are applied, the material will extrude into the gap. If the gap is sufficiently large, the gasket can extrude between the flange and "blow-out," resulting in gasket failure. If the gap is small, the material can partially extrude between the flanges. Thus, when pressure is subsequently released, a piece of the gasket material can be cut off as the flange surface comes together and evenutal failure of the gasket can occur. Extrusion of the O-rings was previously discussed and these characteristics were shown on

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[30] Rathbun, Jr., F. O., op. cit.

Figure No. 26. Extrusion characteristics of the plastic-jacketed gaskets is a function of the material strength, thickness, cold flow characteristics (developed stress and time), pressures, and flange gap. Differential radial deflections in the flanges can cause the shearing (wiping) of the seal contact area, but these gaskets will sometimes re-seal if the condition is not too severe.

With the all-metal, pressure-energized gaskets, gross leakage is usually caused by exceeding yield stresses in the gasket material. This can cause diametral distortions as well as permanent distortion in the legs. Both can cause the loss of intimate contact at the interface sealing surface and radial scores or scratches on the mating surfaces. If the maximum stresses developed in the gasket are limited to the material yield strength, only the differential radial deflections in the flanges can cause loss of the seal. The limiting pressure is dependent upon the hoop stresses developed in the gasket cross-section or the bending stresses developed in the legs. Tests of some of the gaskets disclosed that the leg stiffness for most gaskets is sufficient to take at least the pressures equal to the limiting pressures caused by hoop stresses in the gasket. The maximum hoop stresses developed depend upon the cross-sectional area and the mean diameter of the gasket if the gasket is not "backed-up" by the groove diameter in the flanges. Assuming that clearances between the gasket and the groove are sufficient to allow the gasket to develop stresses equal to the yield strength of the material (which is usually the case), the permissible pressures can be determined for a particular gasket cross-section and diameter. The curves of the hoop stress limit pressures shown on Figure No. 63 are based upon the minimum yield strength of the materials shown. Differential radial deflections between the gasket and the flanges should be further investigated to assure that the allowable maximum for the gasket is not exceeded. The maximum expected hoop deflection in the gasket should be calculated and compared to those in the flanges to assure that the difference is within the differential radial deflection capabilities of the gasket.

The Pressure Science E-ring and C-ring seals are an exception to this because the thin section is usually overstressed at pressure. These gaskets tend to "pack" into the groove (initially, as the gasket is compressed during assembly and finally, as pressure is applied) similarly to the plastic-jacketed gaskets and they rely upon the back-up diameter in the flange groove to prevent the ultimate strength of the section from being exceeded. The outside diameter of the metal V-seal expands as the gasket is loaded and reduces (or eliminates) the free clearance between the gasket outside diameter and the groove diameter. Thus, the limiting pressures for these gaskets is the pressure that would cause excessive bending in the legs. No attempt was made to determine these values for the gaskets tested. However, it is assumed that a curve very similar to that shown on Figure No. 63 would apply for a particular gasket cross-section, but the allowable pressures could be somewhat higher.

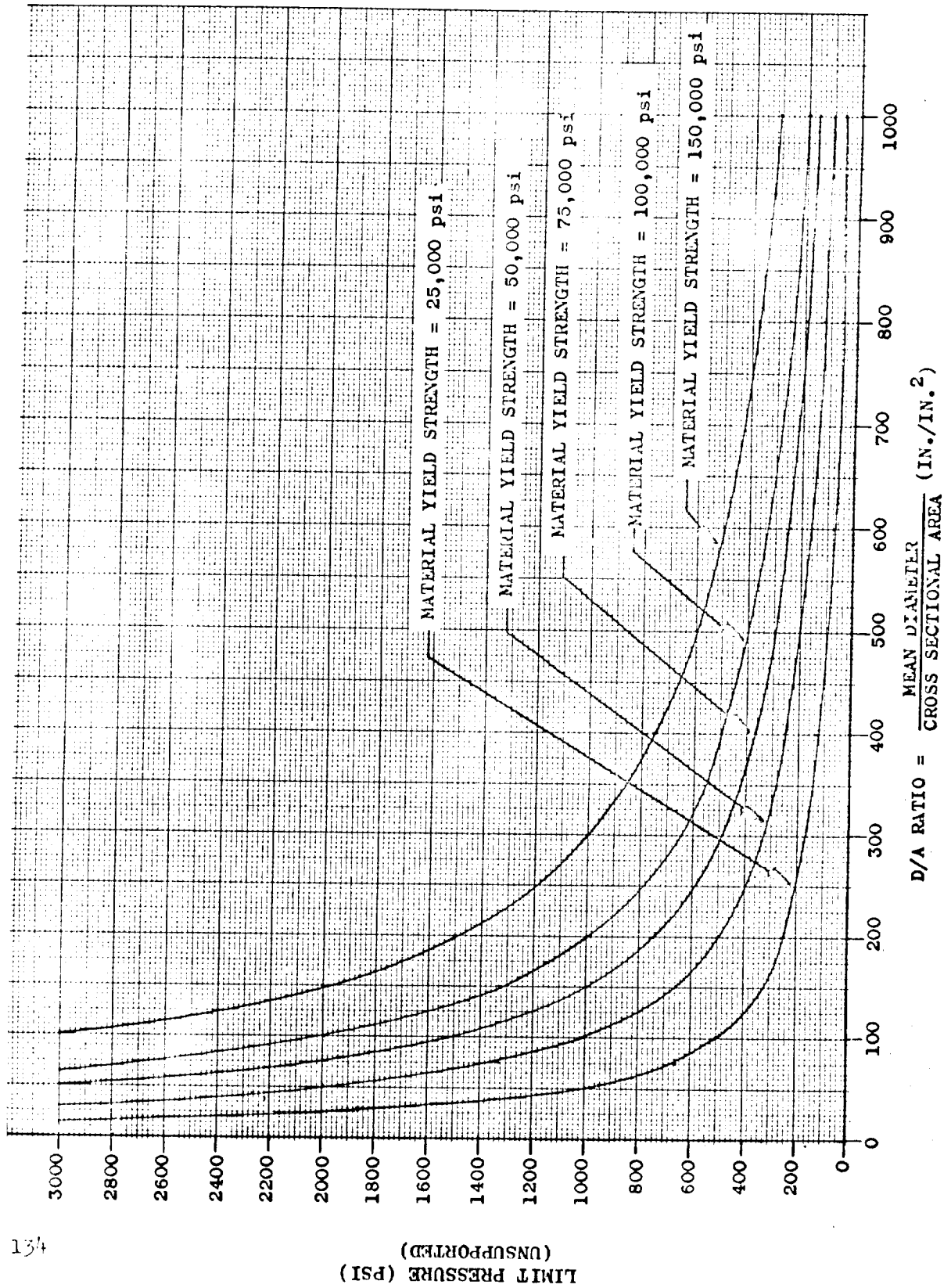

Figure 63
Unsupported Hoop Stress Limit Pressure vs. D/A Ratio
for Metal "Pressure-Energized" Gaskets

With the all-metal diametral gaskets, most failures were caused by differential radial deflections in the flanges when the axial and circular deflections were properly controlled. The gaskets with heavier cross-sections (i.e., Gamah and Nucoseal) were particularly poor in this respect. The Nucoseal could probably be improved, particularly in the larger diameters, because the diametral compressive deflections accrued during installation could be used to good advantage. The Conoseal gaskets depend upon column stresses in the thin cross-section (including some buckling) to provide some springback in the gasket. However, the radial springback in Conoseals made with softer metals, such as annealed copper and soft aluminum, was relatively poor when compared to those made from harder materials (CRES, -T4 and -T6 aluminum). Conoseals still provide the best capabilities in differential radial deflections of all the diametral gaskets tested. They also have the best axial and circular deflection capabilities of all the metal gaskets, including the face gaskets. However, when limiting seal deflection capabilities are used at higher pressures, the problem of excessively bowing the thin section and sliding the sealing edge axially exists. For this reason, it is best to select Conoseal gaskets with the largest available cross-section for the particular diameter desired.

c. Leakage Tests

Results of the leakage tests are shown in Figure No. 64. The data were selected from tests where the design requirements (i.e., loading, deflections, and distortions) and the machining requirements (i.e., surface finish and flatness) were within the values shown on Figure No. 11 as well as other requirements specified by the supplier. No peripheral testing data (determination of deflection, distortion, misalignment capabilities) are included. Also, no data are included from tests where gross leakage was encountered during the proof test.

Permeability of the elastomeric and the plastic materials was the limiting factor in helium leakage rate capabilities. Permeability is a function of the exposed area, leakage path thickness, pressure, and time. Permeability of the O-rings can be reduced by high (80% to 90%) squeeze. This reduces the exposed area and increases the leakage path thickness. The helium permeability of the Raco and Omniseals tested (1/8-in. cross-section) was approximately 1.0×10^{-5} standard cc/sec per inch of circumference. It is claimed that a new configuration of the Raco seal (with a solid machined spring replacing the normal perforated sheet metal spring) reduces helium permeability by a factor of 100 or more. This configuration reduces the exposed area of the Teflon by better than 90% and increases the leakage path thickness. No laboratory tests were made with this configuration.

If gross leakage occurred during proof tests, the joint would rarely pass the leakage test. The elastomeric O-rings were a predominant exception to this. However, they still presented some problems, particularly if extrusion of the O-ring had occurred at proof pressures. The spring-loaded, plastic-jacketed gaskets also would occasionally re-seal after gross leakage during the proof tests. This depended upon the clearance between the groove

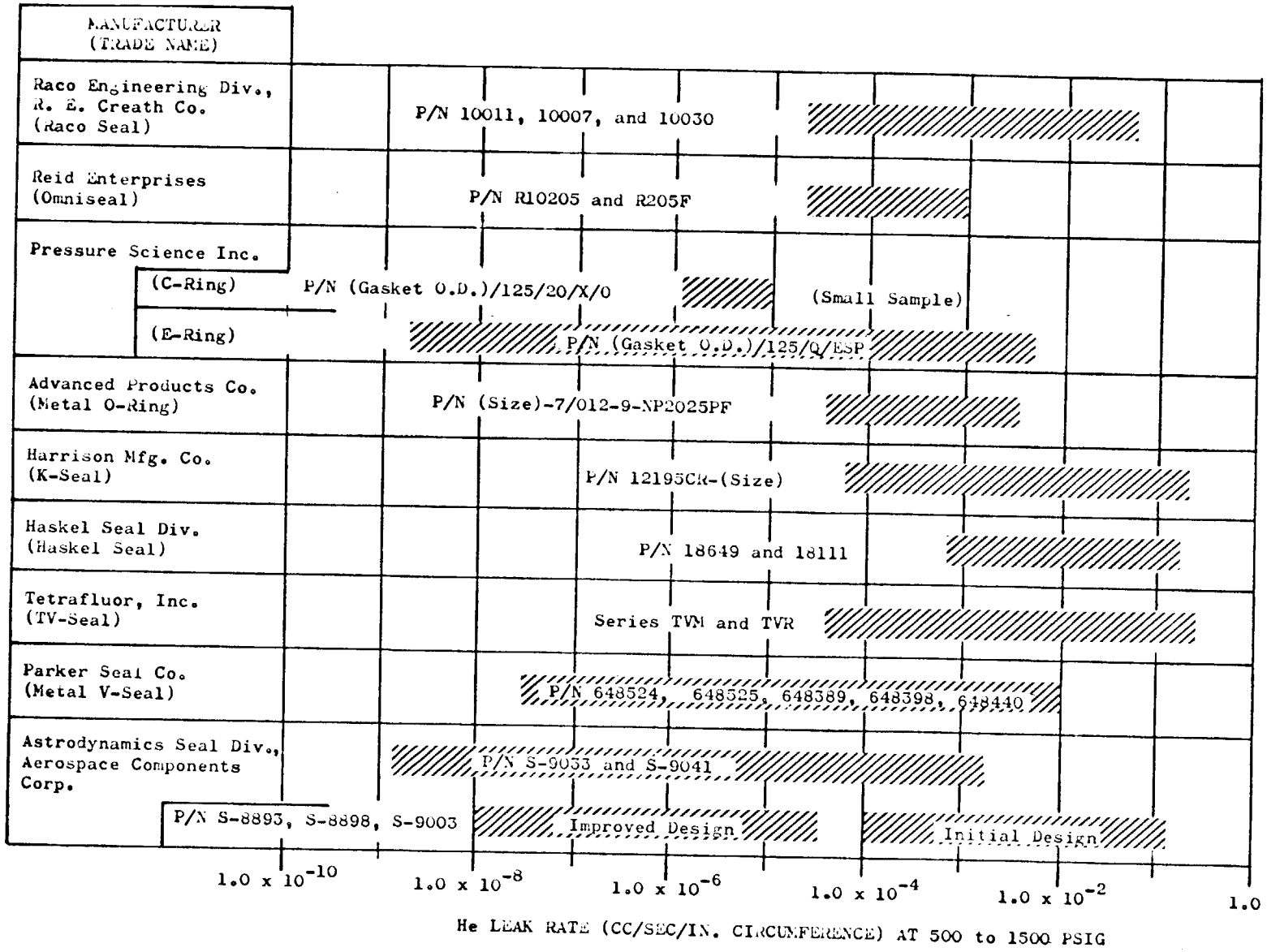

Figure 64
(Sheet 1 of 2)
Helium Leak Rate Ranges for Various Proprietary Gaskets
at Ambient Temperature

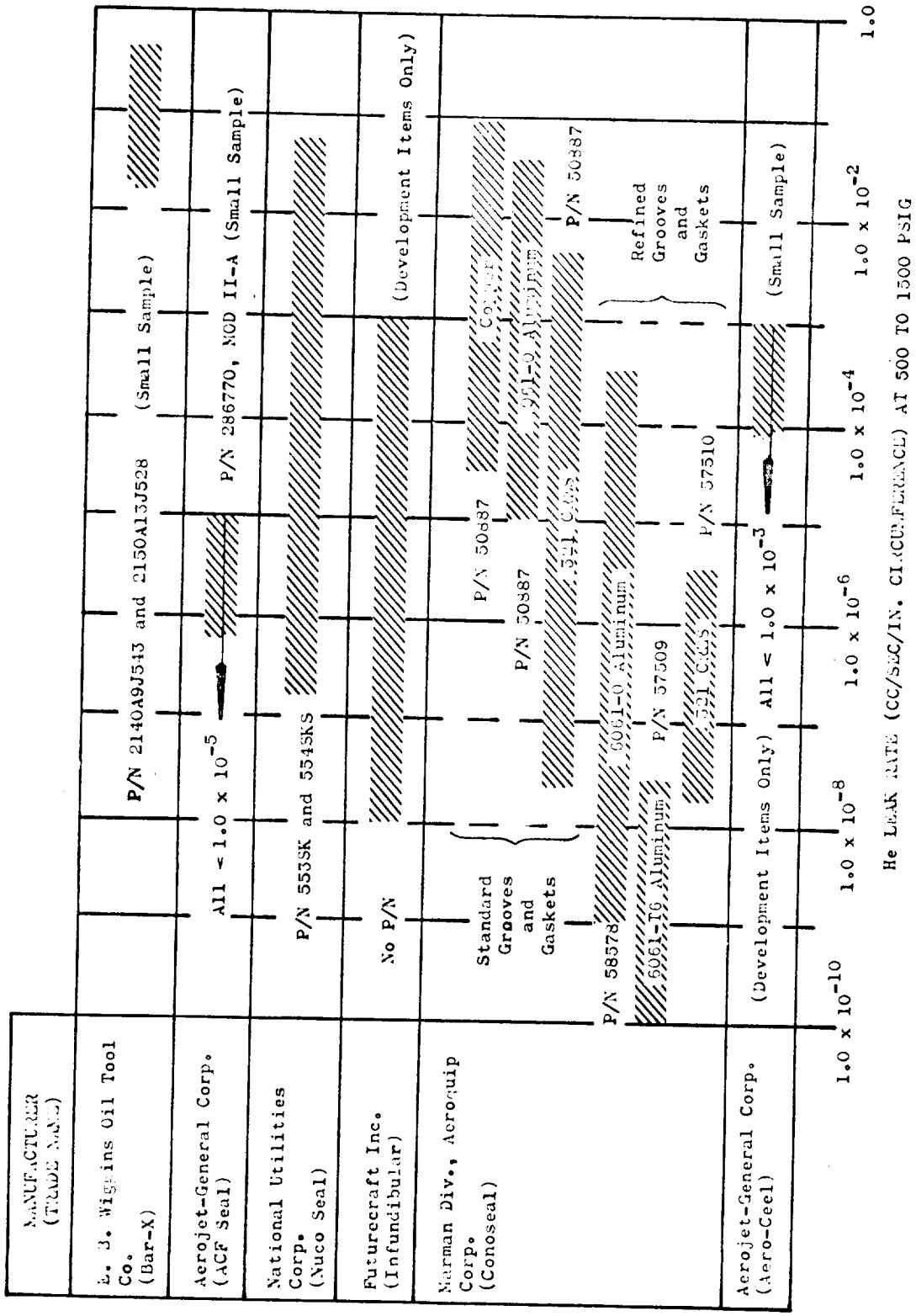

Figure 64
(Sheet 2 of 2)
Helium Leak Rate Ranges for Various Proprietary Gaskets
At Ambient Temperature

diameter and the gasket outside diameter. Gross leakage of the metal, pressure-energized and flat gaskets was usually caused by a loss of the intimate sealing contact resulting from the differential radial (hoop) deflections or by leg distortions. Gross leakage with diametral seals was also caused by loss of the intimate contact resulting from the differential radial (hoop) distortions or bowing of the gasket section.

Groove machining requirements proved to be one of the most important factors in reducing leakage rates. Groove depth had to be held to very close tolerances to provide sufficient and proper squeeze for the pressure-energized face gaskets. Clearances between the gasket outside diameter and the backup groove diameter are also critical, particularly with the thin cross-section gaskets. These gaskets are designed to expand the outside diameter as the mating surfaces are drawn together. When the proper squeeze (groove depth) is attained, the gasket diameter contacts the back-up groove diameter for support. The original Conoseal joints tested (see the standard Conoseal grooves and gaskets, Figure No. 64) were designed primarily for liquid systems. They provided a satisfactory seal for this purpose; however, helium leakage rates were relatively high. As a result, the groove and gasket finishes as well as the tolerances were made more exacting and these refined joints provided a greatly improved helium leakage rate. In this connection, they were the best of any of the joint configurations tested.

Excessive waviness of the mating surfaces causes a high leakage rate with the pressure energized gaskets. This depends upon the resiliency or circular flexibility of the gasket sealing edge and the loading provided by the legs. Out-of-roundness of the sealing diameters also caused high leakage rates with the diametral gaskets. The required roundness is a function of the gasket flexibility. Generally, out-of-roundness tolerances are much more rigid for gaskets with heavier cross-sections than they are for the gaskets with thinner cross-sections.

Surface finishes are extremely important in reducing leakage rates. The O-rings and plastic-jacketed gaskets are the least critical of the pressure-energized types as these materials will flow into the irregularities. However, radial and spiral scratches and/or scores will cause problems if they are relatively deep. The metal gaskets are extremely susceptible to radial and spiral scratches in the mating surfaces. The sealing edge width is small and loading of the metal gaskets usually provides for minimal sealing stresses that cannot always force the sealing material into the scratch to seal effectively.

Surface treatments for corrosion protection that etch the mating surfaces also cause relatively high helium leakage rates. This was a particular problem with anodized aluminum flange surfaces. Relative leakage rates for bare surfaces and anodized surfaces with the all-metal gaskets are shown on Figure No. 65. Apparently, the anodizing process causes surface porosity that the gaskets cannot compensate for. The higher loaded, plastic-jacketed gaskets (increased spring stiffness) could compensate to some extent, but leakage rates were not consistently as good as those obtained with the

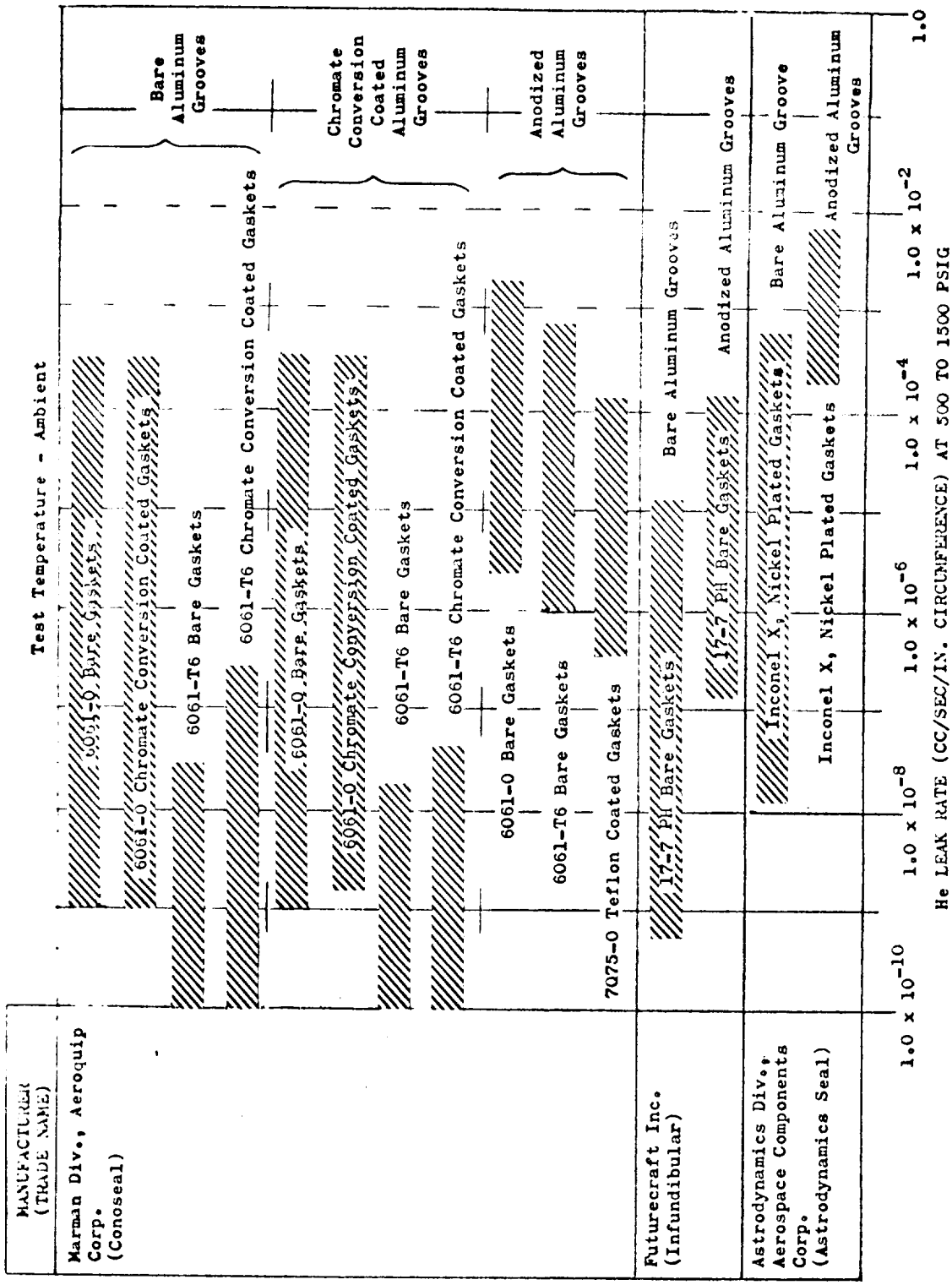

Figure 65
Leak Rate Comparisons - Bare Chromate Conversion Coated and Anodized Aluminum Groove Surfaces

bare aluminum flange surfaces. Higher loading was attempted with the metal gaskets, but the reduction in leakage rate was negligible. Various other corrosion protection surface treatments were tried on the aluminum flange surfaces. These included spray-on coatings, chromate conversion coatings, and a patented anodizing process of the Martin Metals Company. The chromate conversion coatings provided leakage rates that were equivalent to the leakage rates of the bare aluminum surface as shown on Figure No. 65. This is a Government Standard Process acceptable for surface treatment for many applications. The Martin Metals anodizing process also showed good promise, but was not pursued after the chromate conversion coating was found to be acceptable. The same problem was encountered with a passivated 303 CRES flange.

High leakage rates were also caused by grit, dirt, and other particles on the mating flange surfaces. If the flange surfaces were thoroughly cleaned before the joint was assembled, low helium leakage rates could be consistently attained. However, when assemblies were made with flanges that were left open for a few days, very high leakage rates would frequently result.

One problem was encountered with the Omniseal; the "anti-shrink" ring installed at the spring inside diameter would occasionally move out of the Teflon jacket during assembly. When this happened, interference with the jacket would prevent proper seating of the sealing edges and leakage would result. The purpose of the "anti-shrink" ring is to prevent the spring from shrinking under the jacket sealing edges at low temperature (below approximately -100°F). This problem was resolved by improvements in the manufacture of the ring and by increasing the length of the skirt (sealing edge-to-inside diameter) on the jacket.

The original E-ring was designed for a nominal squeeze of 0.006-in. Leakage rates with this design were between 6.0×10^{-3} cc/sec/in. circumference. The redesigned version provided for 0.012-in. nominal squeeze at the inner sealing contact point and 0.006-in. at the outer sealing contact point. The redesigned version provided much lower initial leakage rates (as low as 3.0×10^{-9} cc/sec/in. circumference).

There was a direct correlation between leakage rate capabilities, the method, and the care used to assemble the joint. If the groove area was properly cleaned before assembly and the joint properly tightened in accordance with Figure No. 31 or Figure No. 32 as applicable, leakage rates were consistently lower by a factor of 10 to 10^3. This was somewhat less critical for O-rings for relatively narrow temperature service (standard squeeze) and for the plastic jacketed gaskets for narrow temperature service range. Proper assembly was extremely important for the all-metal gaskets, the "oversqueezed" (low temperature cryogenic) O-ring designs and the plastic-jacketed gaskets.

d. Flange Angular Misalignment Capabilities

Misalignment capabilities of the gasket are dependent upon operating pressures and temperatures, squeeze on the gasket, the gasket

materials and cross-sectional size, preload and stress relationships, as well as other similar considerations. All of the misalignment tests were conducted at ambient temperature. It should be noted that the misalignment capabilities can change drastically if temperatures are increased or decreased. Also, any angular misalignment in the mating surfaces after assembly has been completed will drastically reduce the flange deflection capabilities during operation. Typical results from these tests are shown on Figure No. 66.

The average misalignment capability of the pressure-energized gaskets tested was approximately 0.004-in. maximum for the 1/8-in. cross-section gaskets. The range was 0.010-in. maximum at 1500 psig for the plastic-jacketed gaskets to less than a 0.001-in. maximum at 1500 psig for some of the metal gaskets. The affects of larger or smaller diameters is negligible. This means that regardless of diameter, the maximum "out-of-parallelism" between flange faces (total run-out) must be within these values for the particular cross-sectional size tested. Also, if misalignment is expected during assembly, the flanges must be designed for deflection capabilities which are greatly reduced from those shown on Figure No. 11.

The angular misalignment capabilities of the diametral gaskets are approximately equal to those of the face gaskets. It was assumed at the beginning of the test that the diametral gaskets would be capable of allowing greater angular misalignment than the face gaskets. Testing proved that only the single Conoseal joints were as good in this respect as the better of the all-metal face gaskets (approximately .006-in. maximum at 1500 psig with the .032 x .275 cross-section gaskets). The other diametral gaskets were approximately equal in performance to the average, all-metal, face gaskets. However, the best configuration of the dual Conoseal concepts reduces this capability to approximately 0.004-in. maximum because of the tolerance stack-up and the interaction between the two grooves and gaskets.

e. Axial Separation Capabilities

Separation (or axial deflection) capabilities of the gaskets is dependent upon the pressure and temperature, gasket materials, and gasket cross-section. The separation tests were designed to gain an indication of the gasket capabilities with respect to flange rotation and bolt elongation. The flange rotation (bending of the flange section from the bolt circle or flange contact point to the gasket sealing surface) actually results in a small angle between the surfaces. Straight separation tests usually approximate the condition sufficiently for the data to be useful. However, with some gaskets, the angle resulting from flange rotation will improve the gasket capabilities over those obtained from straight axial separation tests. With other gaskets, the angle can be detrimental to the gasket capabilities over those of straight axial separation tests. Typical results from these tests are shown on Figure No. 67.

The average axial deflection capability for the pressure-energized face gaskets was approximately .004-in. for the gaskets with 1/8-in.

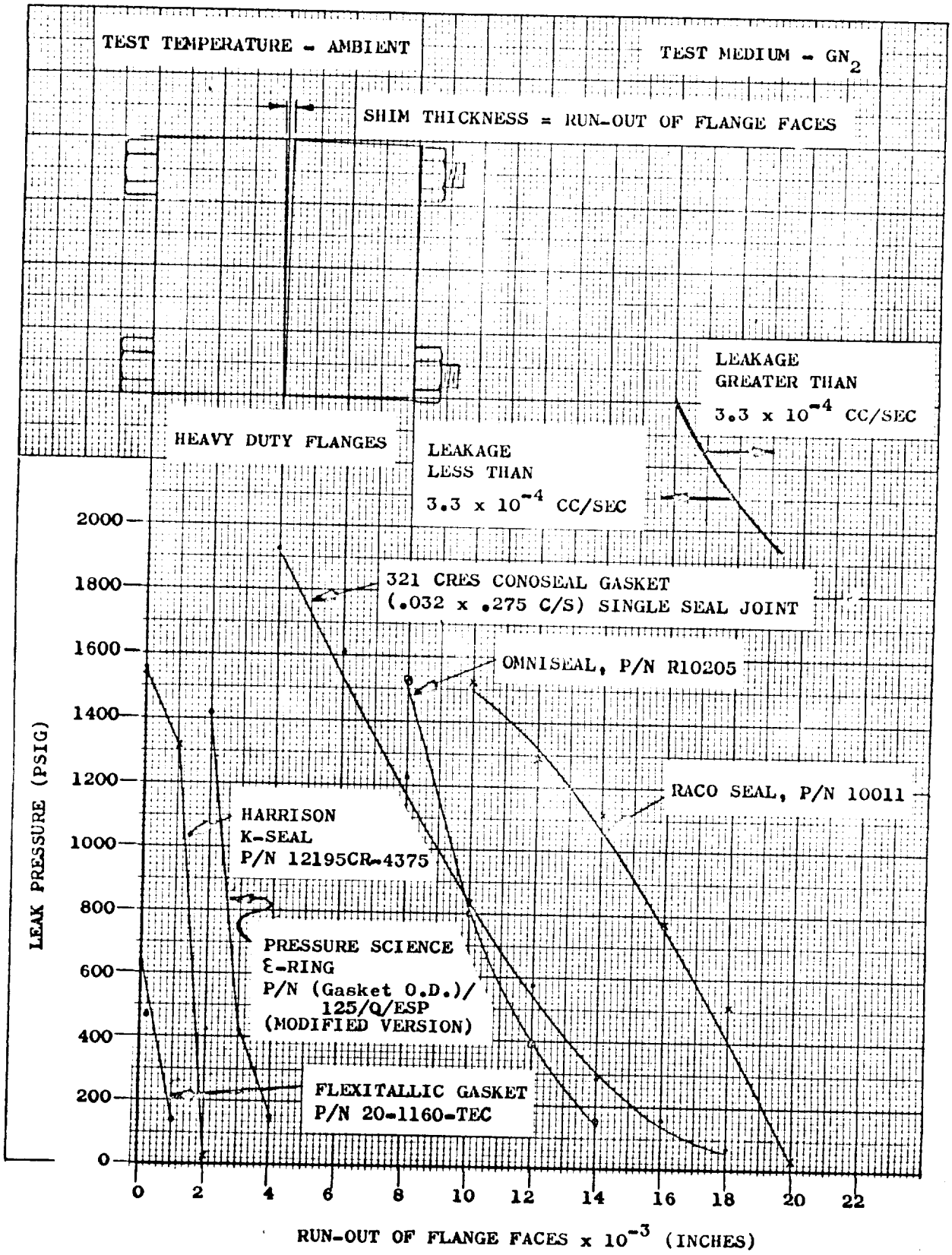

Figure 66
Results of Flange Angular Misalignment Tests

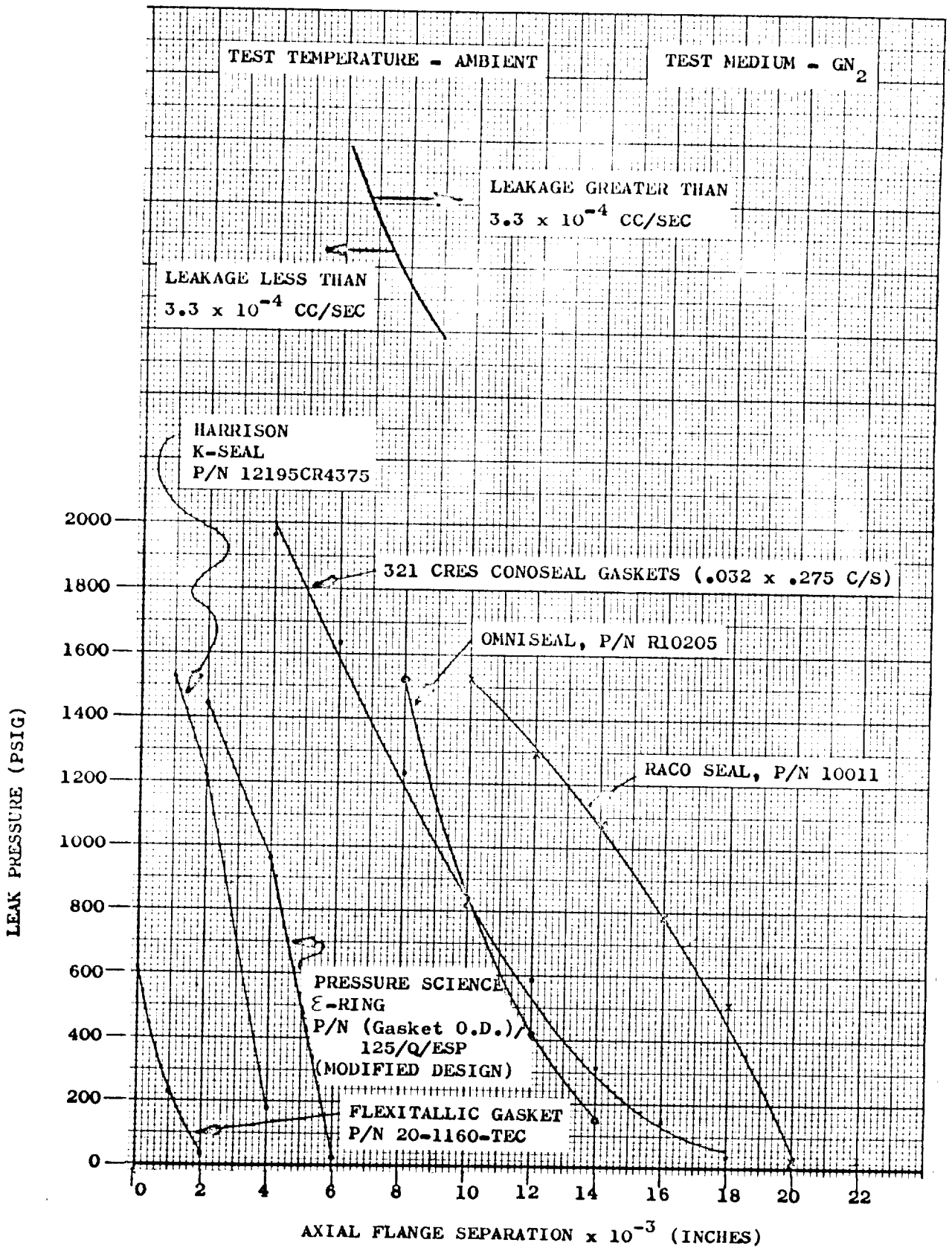

Figure 67
Results of Flange Axial Separation Tests

cross-section. It ranged from .012-in. maximum at 1500 psig for the plastic-jacketed gaskets to 0.001-in. maximum for some of the all-metal gaskets. The affects of larger or smaller diameters for the same cross-section are essentially negligible. Therefore, the flanges and fasteners must be designed so that the maximum axial deflections expected in the joint are within the values for the particular cross-sectional size tested. It is not known what affects that pressures above the 2000 psig (proof) pressures would have upon these capabilities.

The average axial deflection capabilities for the diametral gaskets were approximately 0.005-in. maximum (somewhat better than for the face gaskets). The single Conoseal joints proved approximately equivalent to the plastic-jacketed face gaskets in this respect, at least up to the pressures used for these tests. The dual Conoseal joints were capable of only 0.005-in. maximum axial deflection. The increased radial distance between the bolt circle and the gasket circle for this configuration and the interaction between the two glands apparently causes a drastic reduction in these capabilities.

f. Flange Reusability

The number of times that a joint can be assembled and disassembled without remachining the flanges is dependent upon the gasket and mating surface hardness, how free of particles and contaminants the joint is kept during the assemblies, and the sealing stresses involved. The elastomer and plastic-jacketed gaskets present very little problems in this respect, providing the surfaces are cleaned before each assembly, because the mating surface materials are much harder than the gasket materials. The all-metal, pressure-energized, and diametral gaskets can present some serious problems if the hardness of the actual sealing materials on the gasket and the mating surfaces are ignored.

Tests with 7075-T6 aluminum flanges and the highly-loaded gold-plated pressure-energized gaskets (i.e., Koppers, some of the V-seals, and some of the Astrodynamics seals) indicated that the flange surfaces deformed after the first assembly cycle and an increase in leakage of 100 times was experienced during the second assembly leak test. After resurfacing the flange, leakage rates returned to approximately the original values. This gasket provided the highest sealing stresses (and the lowest leakage rates) of any of the pressure-energized gaskets tested on the first assembly. Very few of the gasket manufacturers specify minimum flange hardness values for their gaskets; therefore, the gasket selected should be thoroughly analyzed to determine the minimum flange hardness required. As a general rule, the groove surface hardness should be a minimum of 1.3 times the gasket sealing material hardness (on the Brinell Scale). However, this value is greatly affected by the sealing material (plating) thicknesses and the hardness of the gasket base material. If plating thicknesses are greater than 0.001-in., the general value applies providing no deformation of the base material of the gasket (i.e., sharp corners) takes place as the joint is assembled. If plating thicknesses are less than 0.001-in. or if deformation of the gasket base material takes place,

the flange surface hardness should be increased as the plating thickness decreases until the hardness of the base material of the gasket becomes the governing factor.

Reusability is also governed by the amount of radial motion between the groove surface and the pressure-energized gasket with thin cross-section. The E-ring (N-155, gold plated) deteriorated the 347 CRES flanges (after four assembly and proof and leak tests) to the extent that the groove surfaces had to be remachined. The best method for eliminating this problem is to increase the plating thickness. Closer fits between the gasket outside diameter and the mating groove diameter can also improve flange reusability because it prevents relative movement of the sealing interface under pressure. However, this introduces manufacturing (tight tolerances) problems as well as possible assembly and gasket removal problems.

It cannot be assumed that the raw flange materials specified will provide sufficient groove hardness, particularly if the flanges are machined from thick sections. Heat treatments applied to the raw stock do not penetrate through thick sections, particularly with aluminum materials. The flanges must be rough-machined close to the final machining dimensions, then heat-treated to assure proper surface hardness. Aluminum 6061-T6 Conoseal gaskets were used in properly processed 6061-T6 flanges for a total of 30 assembly cycles with no deterioration in leakage rates and no noticeable deformation or galling in the groove surfaces. The average Brinell hardness of the gaskets was only two points lower than the hardness of the flange surface. Also, 6061-T4 gaskets galled a 6061-T6 flange fabricated from thick raw stock sections after three assembly cycles. Brinell hardness tests indicated that the average gasket hardness was six to ten points higher than the hardness of the groove area.

Thorough cleaning of the flange sealing areas and/or well-sealed protective closures proved to be extremely important in providing a good joint initially and in preventing deterioration of the groove surfaces, particularly with the all-metal gaskets. A small test program was conducted to determine the effects of assembling flanges both with and without pre-assembly cleaning. The flange surfaces were left exposed in an enclosed (but uncontrolled) area for two to three days and were not cleaned before assembly and leakage testing. The results of these tests were compared with test results where the flange surfaces were protected and thoroughly cleaned before each assembly and leak test. As expected, the plastic-jacketed gaskets have the ability to absorb some fine particles and still provide approximately the same sealing capabilities. However, some deterioration of the flange surfaces did occur under these test conditions. The all-metal gaskets usually did not provide the same sealing characteristics at the first assembly with contaminated flanges.

A summary of these test results is presented as Figure No. 68. It should be noted that the "uncleaned" test results are somewhat random and no attempt was made to determine either the particle size or the type of contamination that accumulated on the flange surfaces. However, the

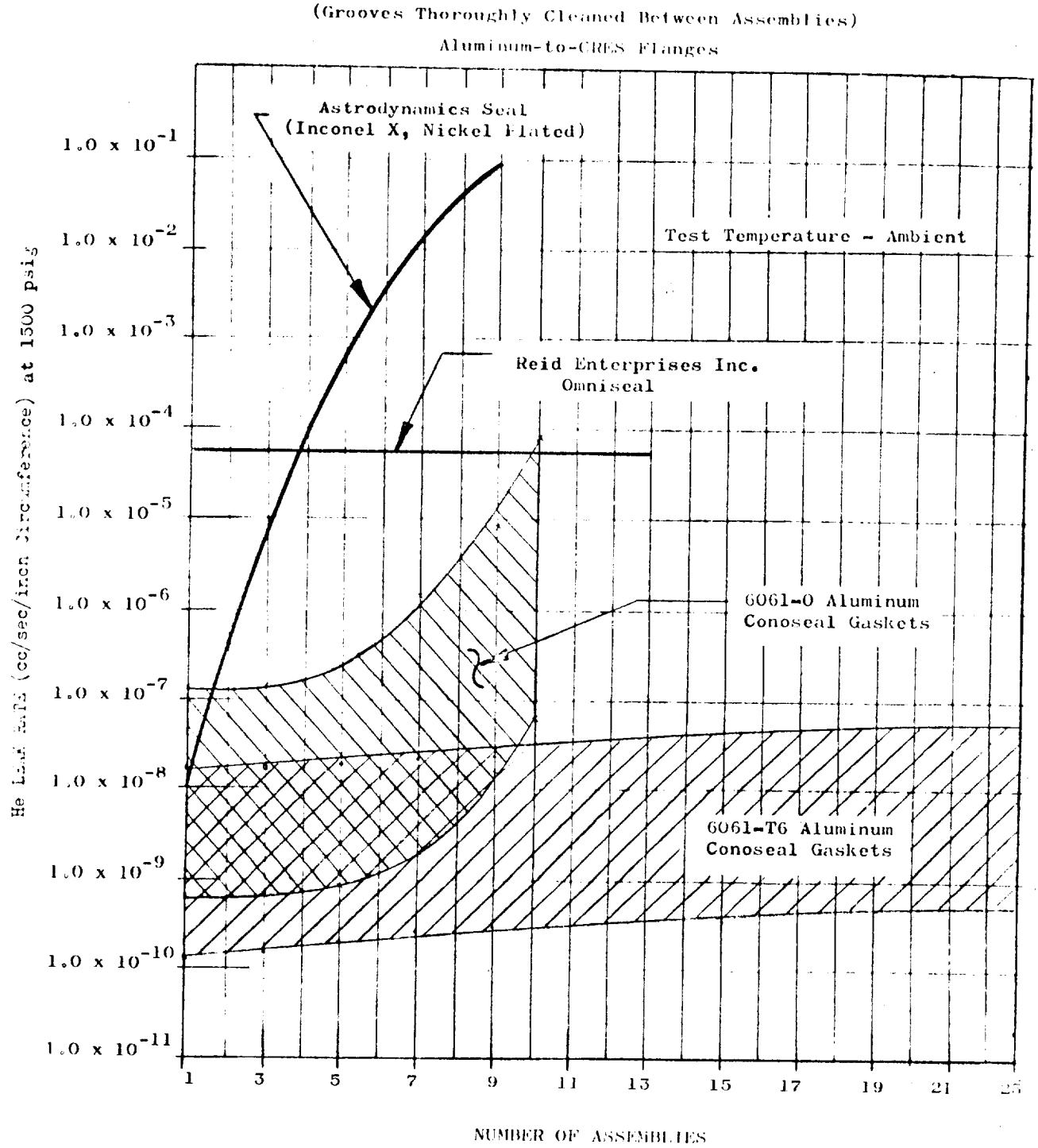

Figure 68
(Sheet 1 of 2)
Leakage vs. Number of Assemblies

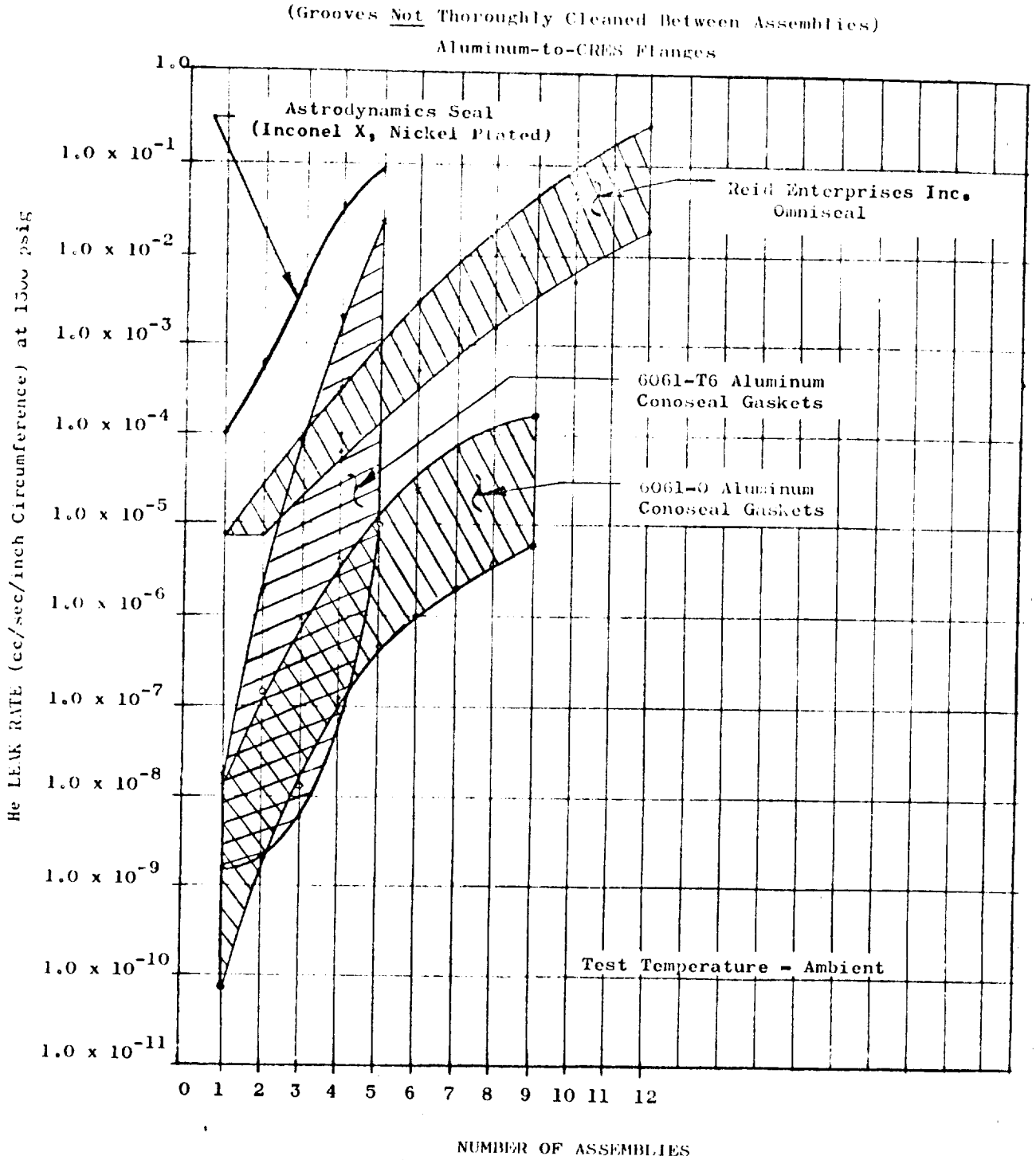

Figure 68
(Sheet 2 of 2)
Leakage vs. Number of Assemblies

results point out the importance of proper cleaning procedures before assembling the joint. They also show the importance of selecting the proper flange surface and gasket sealing surface hardnesses where repeated assembly of the joint may be necessary. The Astrodynamics seal rapidly deteriorated both the aluminum and the CRES flanges when there was no pre-assembly cleaning. When the flanges were thoroughly cleaned before assembly, the aluminum flange surface gradually deteriorated. The 6061-0 aluminum Conoseal gaskets deteriorated both the flanges less rapidly than the 6061-T6 aluminum gaskets when no pre-assembly cleaning was done. It was not definitely determined why the leakage rates increased with the 6061-0 gaskets when the flanges were thoroughly cleaned before each assembly. No apparent deterioration of the flange surfaces occurred. It was assumed that the oxidation of the bare aluminum flange surface was to an extent that precluded the soft aluminum gasket sealing load from compensating for it.

A by-product of the flange reusability tests was some data for gasket reusability. It was found that if the flanges are thoroughly cleaned before each assembly, the plastic-jacketed gaskets could be reused (with little affect upon leakage rate) a number of times providing that the gasket is replaced between the flanges in approximately the same position. Also, great care must be taken in removing the gasket from the groove to prevent damage to both the gasket and the groove because this type of gasket is relatively difficult to remove. Special tools fabricated from Teflon or soft plastics were devised to remove the gasket from the groove and reduce the possibility of damage. However, reuse of gaskets is not recommended for actual hardware applications because the reliability (repeatability of leakage rates) after the first assembly is considerably reduced. It is highly recommended that a new gasket be used for each assembly of actual hardware.

Reuse of some all-metal pressure-energized gaskets proved successful under laboratory conditions. Again, extreme care was taken in removing and replacing the gasket. However, the plating or coating used on the highly stressed areas of the gasket would usually flake off after two or three assembly cycles.

g. High Temperature Tests

Results of the high temperature tests are shown on Figure No. 69. The data were collected from results of tests at various temperatures depending upon the gasket type and materials as well as the application desired. The all-metal, pressure-energized gaskets were tested at temperatures to 1700°F. No leakage rate data were collected during these high temperature tests, but visual observations were made to determine any gross leakage. Leakage tests were made at ambient temperature after the high temperature tests to determine the extent of the leakage and possible erosion of the sealing area during its exposure to high temperature. If gross leakage occurred during this test, the joint was disassembled and the parts examined to determine the cause.

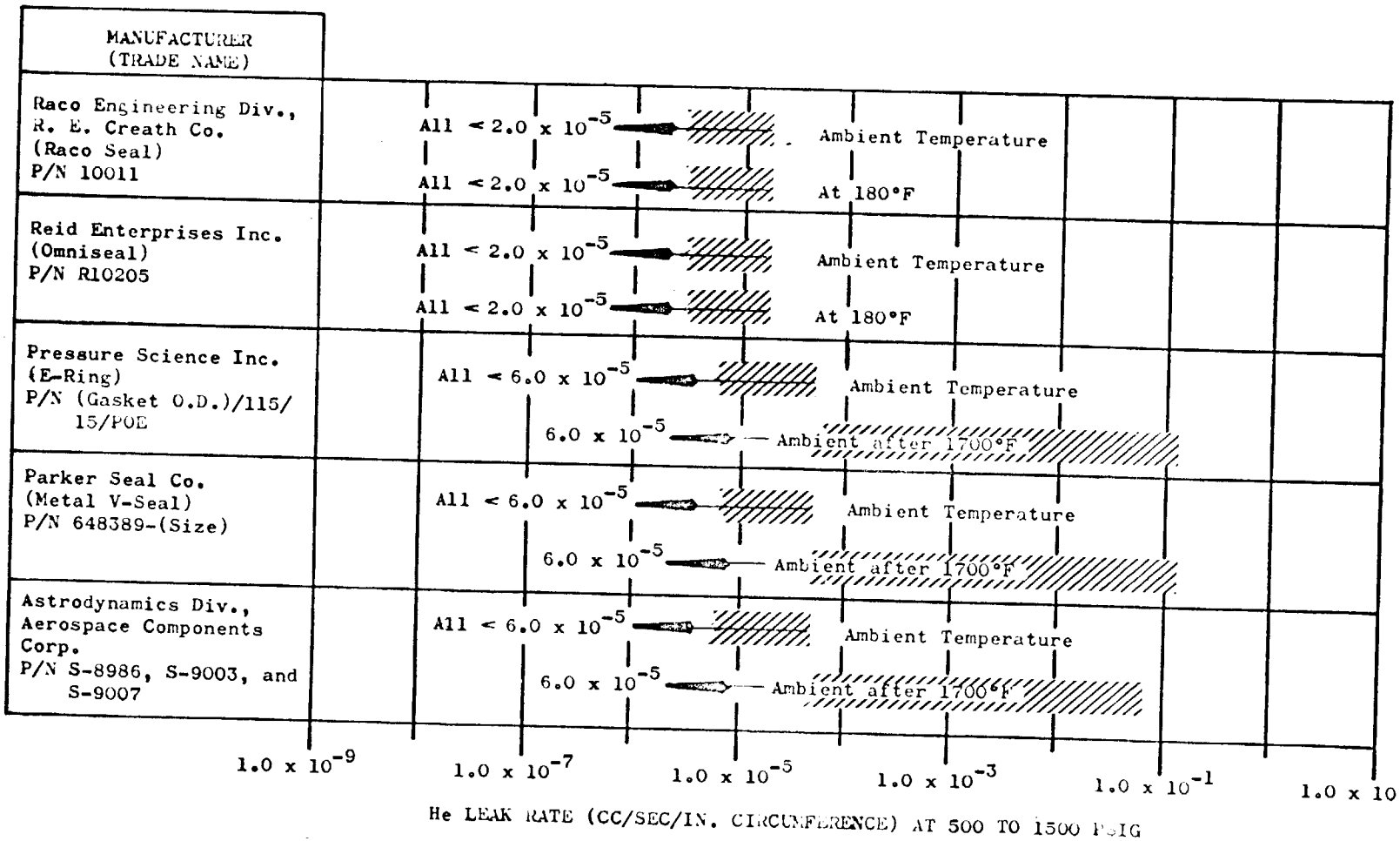

Figure 69
Initial High Temperature Test Results

The plastic-jacketed gaskets were tested for a system having a maximum temperature requirement of 180°F. Therefore, 180°F was the highest temperature used in the evaluation tests. Leakage tests were made at both ambient and 180°F. Although these gaskets had provided satisfactory service in other applications at higher temperatures, no useful leakage rate data was collected.

h. Low Temperature Tests

Results of low temperature tests are shown on Figure No. 70. The data were collected from results of tests at various temperatures depending upon the gasket type and materials as well as the application desired.

In most cases, the leakage rates at low temperatures were greater than the leakage rates at ambient temperature. In some cases, particularly the transition joint (CRES to aluminum), the leakage rate increased after the joint was returned to ambient temperatures.

Low temperature brittleness, elongation characteristics, and contraction characteristics caused hoop tension, bending failures, and/or circumferential crazing and cracking of the jacket on the plastic-jacketed gaskets. Typical hoop tension and bending failures are shown on Figure No. 71 while circumferential cracking is shown on Figure No. 72. The Omniseal is used as the example in Figure No. 72 although similar failures were encountered with the Raco seal. The hoop tension failures occurred only during the "low pressure bleed-in followed by high pressure" leak tests when temperatures were less than -380°F. The hoop tension failure was caused by the jacket shrinking away from the back-up diameter during low pressure bleed-in, after which, pressure forced it to expand out to the back-up groove diameter during the high pressure leakage test. Before the jacket could reach the back-up groove diameter, the jacket failed in tension as shown in Figure No. 71. This problem can be reduced to some extent if the outside diameter of the gasket is a slight press-fit in the groove diameter. If the jacket section has built-in compressive stresses, the contraction will not move the jacket as far away from the back-up groove diameter, and when high pressure is applied, the jacket outside diameter reaches the back-up groove diameter before the ultimate tensile strength is reached. No failures of this type were encountered when pressure was applied to the gasket before the jacket temperature was reduced. Circumferential crazing and cracking occurred at the jacket corner when the joint deflected during pressurization as shown on Figure No. 72. No deflection capabilities were determined for these gaskets at very low temperature, but the tests proved that they are drastically reduced at a temperature of -380°F or less. If deflections of any magnitude occur during operation of the joint, crazing of the plastic jacket would be expected after a few pressure and low temperature cycles.

No serious problems were encountered with the all-metal, pressure-energized gaskets providing the gasket material contraction characteristics were reasonably close to those of the flanges. Where differential contraction between the two flanges, or between the flanges and the gasket, was

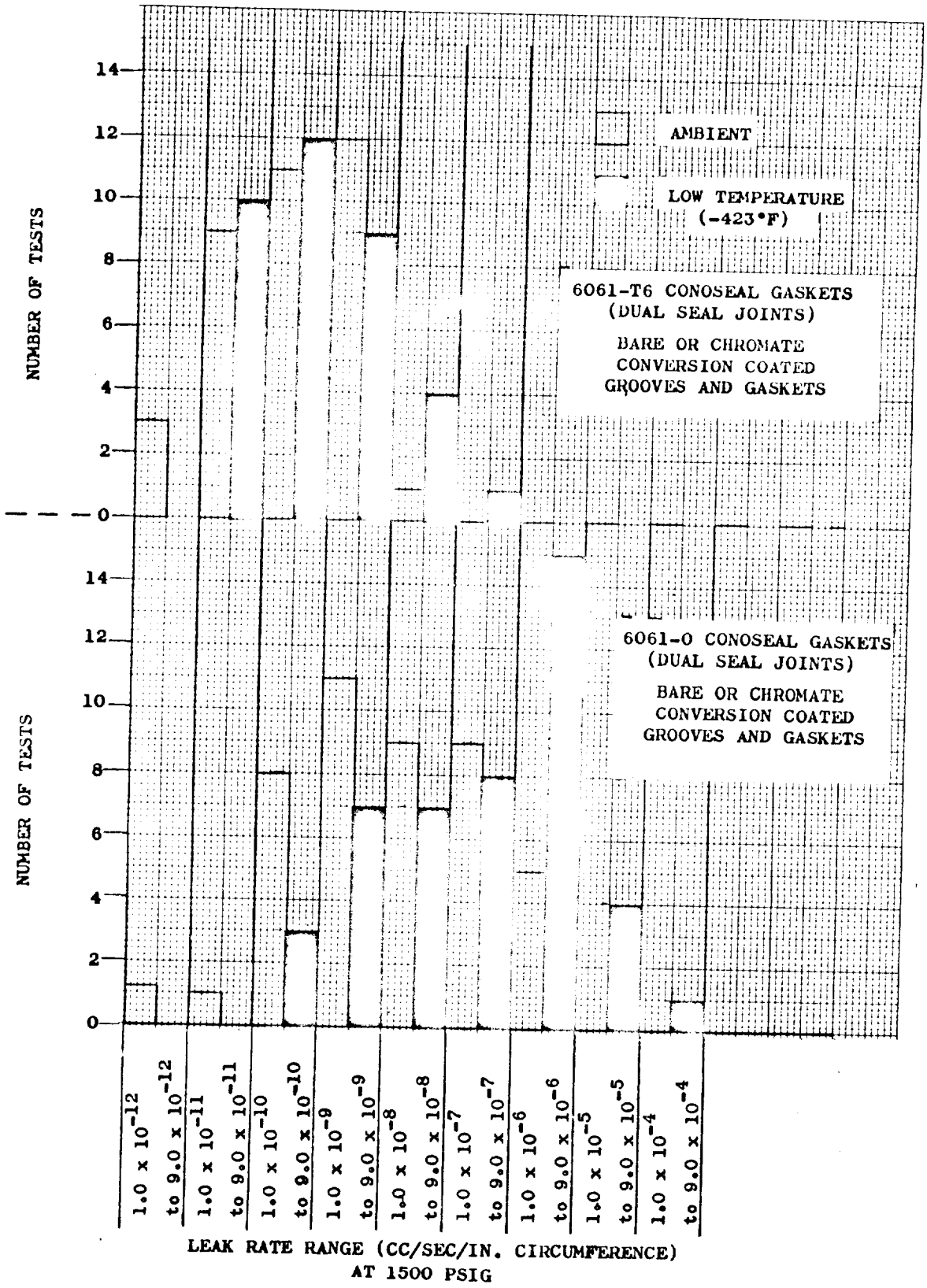

Figure 70
(Sheet 1 of 2)
Low Temperature Test Results (Laboratory Evaluation)

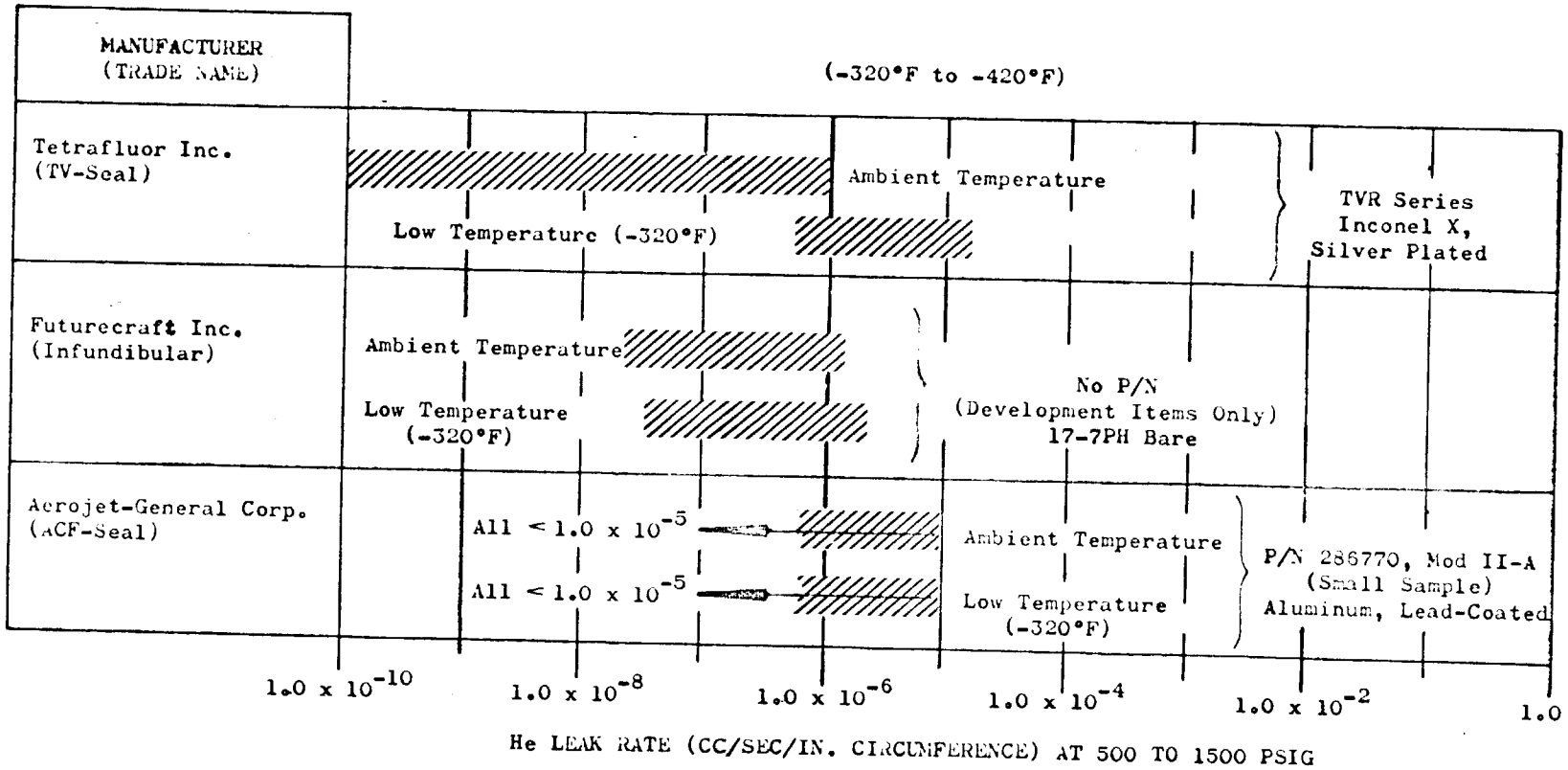

Figure 70
(Sheet 2 of 2)
Low Temperature Test Results (Laboratory Evaluation)

OMNISEAL [illegible]

Failure occurred during low temperature pressurization after low pressure cryogenic bleed-in. No failures occurred when the gasket was pressurized before or at the same time that the low temperature was applied.

Figure 71
Typical Low Temperature Tensile and Bending Failure of Teflon Jacket (Spring-Loaded, Teflon-Jacketed Gaskets)

Figure 72
Typical Low Temperature Circumferential Cracking of the Teflon Jacket (Plastic-Jacketed Gaskets)

Failure usually occurred at inner junction of the gasket outside diameter and the sealing leg in contact with the flat-faced flange.

large, the intimate sealing contact was lost resulting in excessive leakage. When this happened, radial scratches or marks were left in the groove from the wiping action between these surfaces. The gaskets with stiff sections (or stiff legs) were more susceptible to this problem than the gaskets with relatively flexible sections.

The diametral gaskets can present some very serious problems when the differential contraction of the two flanges is large. This is a function of the radial flexibility of the gasket and the hardness of the gasket materials. Stiff sections with soft materials presented more of a problem than the more flexible sections with harder materials. The diametral gasket joints must be designed so that differential contractions tend to increase the pre-load on the gasket and make a tighter seal. The soft metal gaskets with heavier sections can present problems in maintaining the seal after a number of thermal cycles because the gasket takes a permanent set during the thermal cycle.

i. Thermal Cycling Tests

The results of the low temperature thermal cycling tests conducted with the gaskets tested are shown on Figure No. 73.

The plastic-jacketed gaskets would craze and crack during these tests if the lowest temperature was -380°F or less. This problem was discussed previously and a typical example was shown on Figure No. 72. No problems were encountered if the lowest temperature was greater than -320°F.

Low temperature thermal cycling tests were conducted with one of the all-metal pressure-energized gaskets; the small diameter Tetrafluor "TVR" series. The results from these tests are shown on Figure No. 73.

If the initial seal was good, no thermal cycling problems were encountered with the diametral gaskets when the contraction characteristics of the two flanges and the gaskets were approximately equal. Thermal cycling problems were encountered with the diametral gaskets in transition joints (CRES-to-aluminum flanges). The problem was eliminated with the Conoseal gaskets by substituting 6061-T6 or 6061-T4 aluminum gaskets for the 6061-0 gaskets used initially. This can present a problem of deteriorating the groove surfaces in the aluminum flanges, if the groove surface hardness is not greater than that of the 6061-T4 or 6061-T6 gaskets (see paragraph III, C, 1, f).

A by-product of these tests was the discovery of a dimensional instability problem in the forged aluminum valve bodies. Permanent dimensional changes occurred in the forged aluminum flanges during the low temperature thermal cycling tests to the extent that proper fits between the grooves and the gaskets could not be maintained and excessive leakage resulted. The dimensional changes were caused primarily by residual stresses from heat treatment (quenching) and the machining operation. A special fabrication process incorporating a re-heat treatment to relieve machining stresses and a

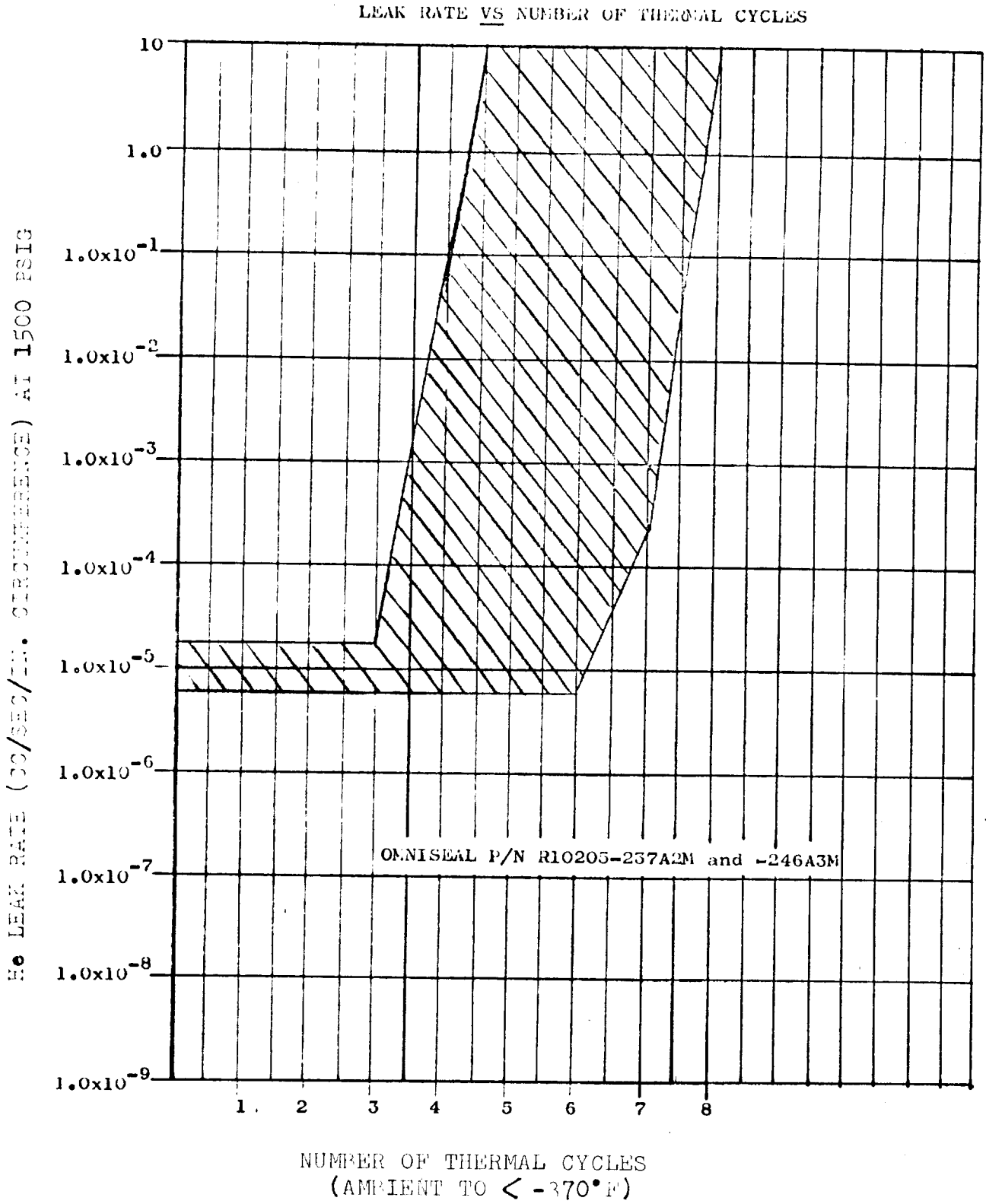

Figure 73
(Sheet 1 of 3)
Low Temperature Thermal Cycling Test Results

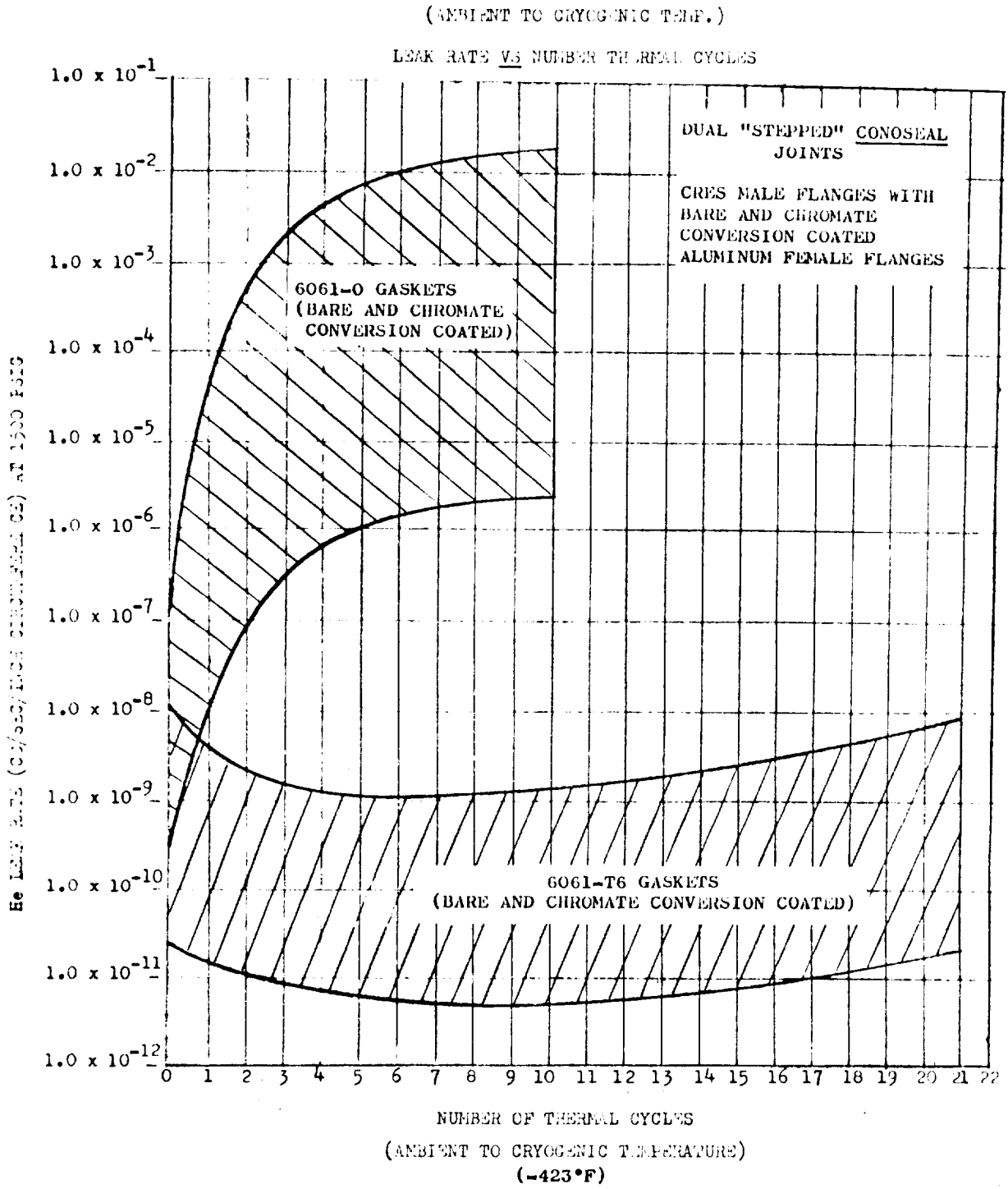

Figure 73
(Sheet 2 of 3)
Low Temperature Thermal Cycling Test Results

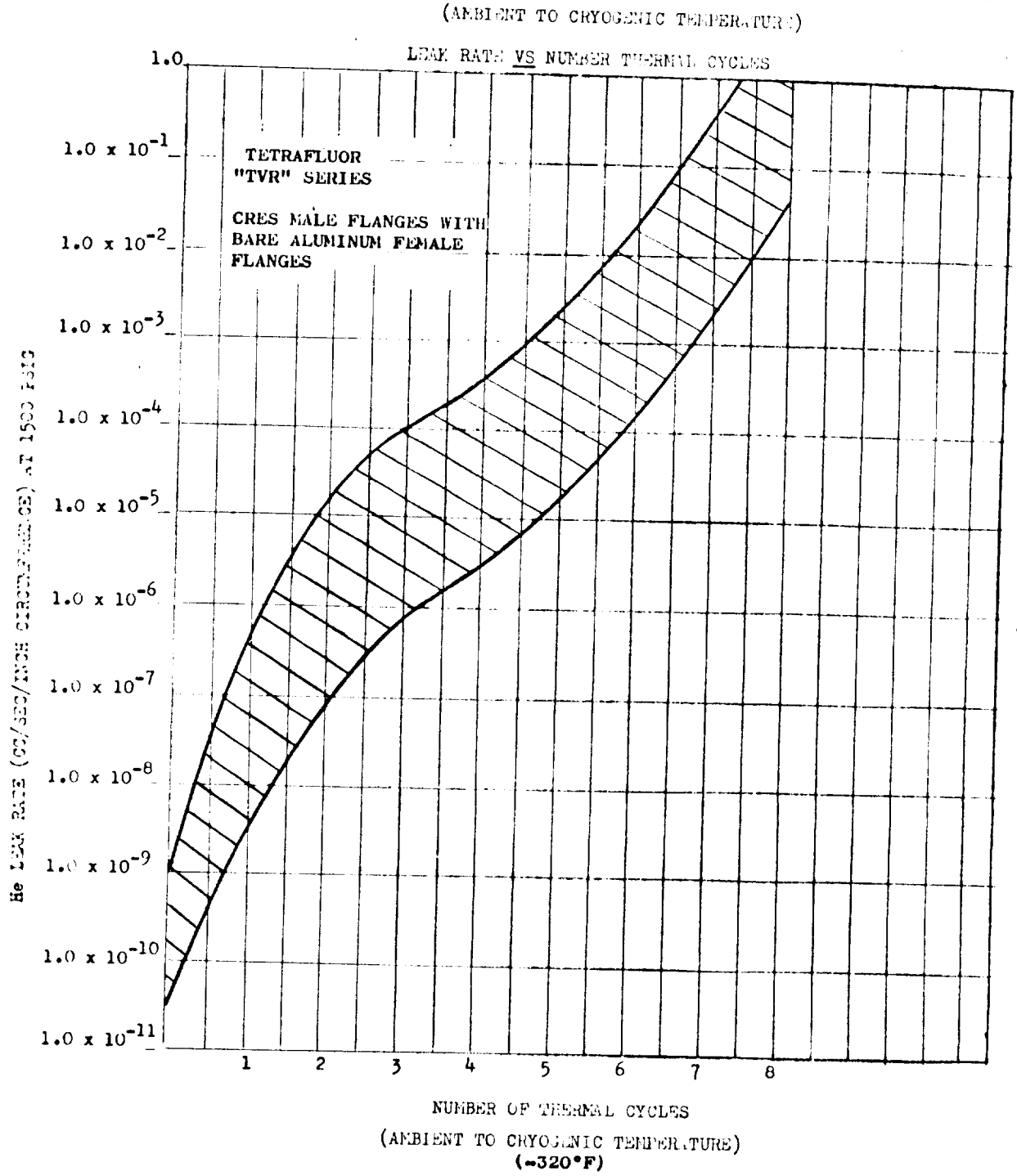

Figure 73
(Sheet 3 of 3)
Low Temperature Thermal Cycling Test Results

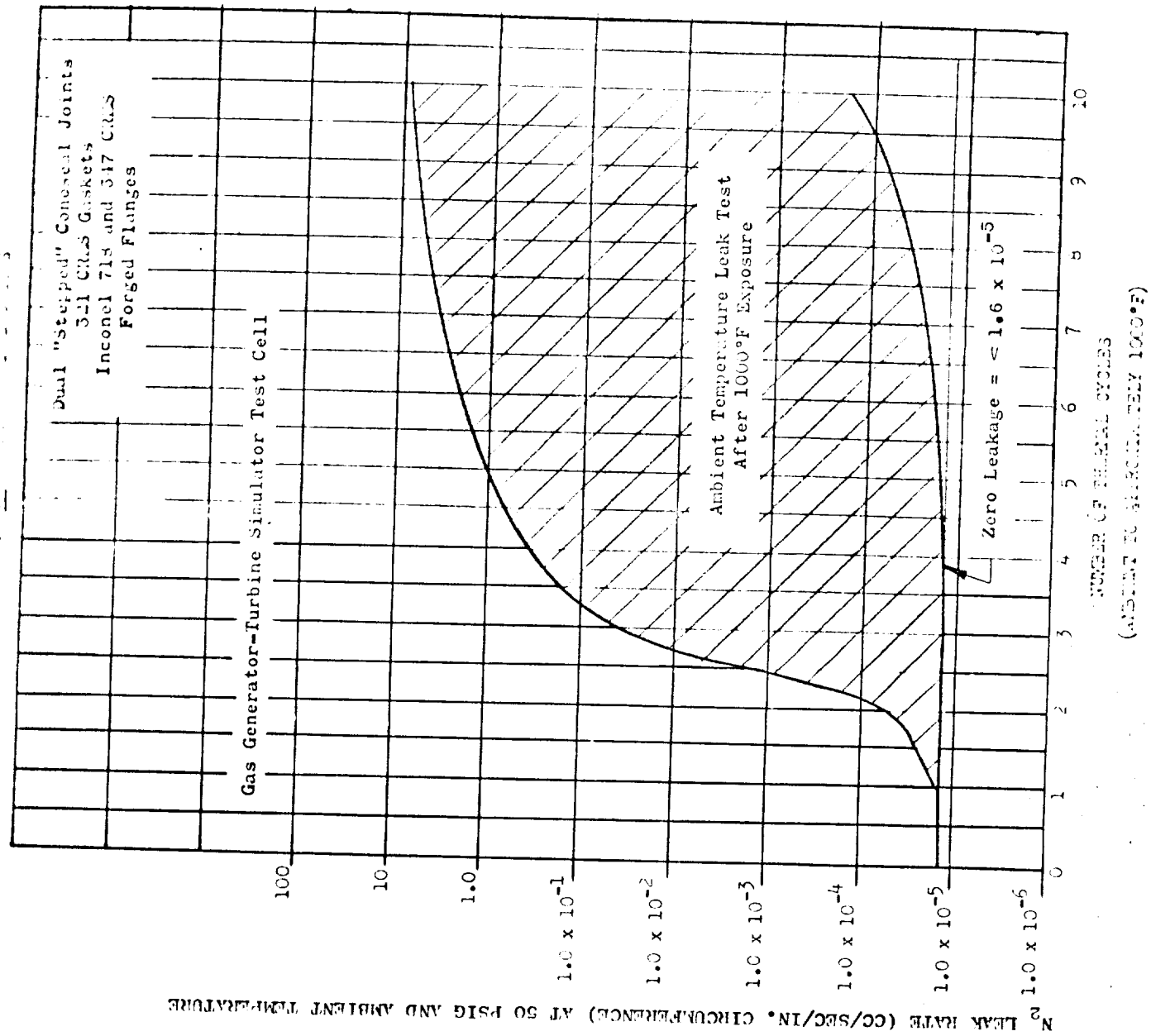

Figure 74
(Sheet 1 of 2)
High Temperature Thermal Cycling Test Results

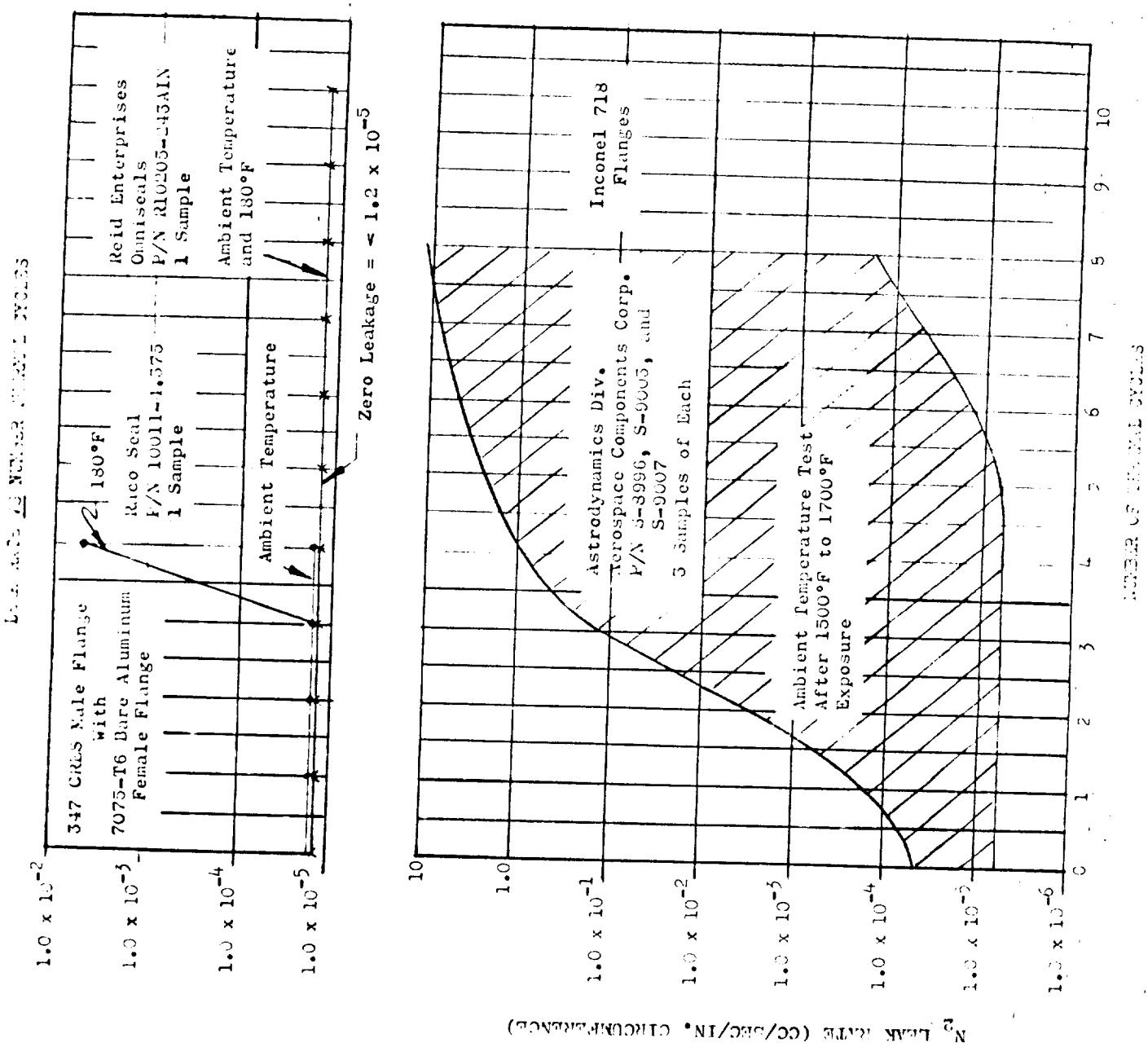

Figure 74
(Sheet 2 of 2)
High Temperature Thermal Cycling Test Results

pressure stabilization operation to reverse the stress pattern resulting from the quenching operation was devised to reduce the dimensional changes caused by low temperature cycling. The details of the resolution of this problem are presented in a separate report[31].

j. High Temperature Thermal Cycling Tests

The results of the high temperature thermal cycling tests with those gaskets tested are shown in Figure No. 74. The all-metal pressure-energized gaskets were cycled from ambient temperature to approximately 1700°F. The test cell hardware used was intentionally designed for absolute minimum weight for this application and the data should not be construed as representing the maximum sealing capabilities of the gasket. Permanent flange distortions as high as 0.012-in. (rotation of the flange section) were noted at the sealing surfaces at the conclusion of the thermal cycling tests. The testing supplied sufficient data to provide a good approximation of the particular gasket capabilities for high temperature service.

The plastic-jacketed gaskets were cycled from ambient temperature to 180°F. No satisfactory explanation could be found for the sudden increase in leakage rate for the particular Raco seal gasket. Test cell designs were such that no distortions of the flanges were encountered during the cycling.

k. Fluid Resistance and Compatibility Tests

Fluid resistance and compatibility tests were conducted with the plastic-jacketed and one all-metal pressure-energized gasket (304 CRES, Teflon coated) for nitrogen tetroxide system service. The gaskets were installed in transition joints (CRES-to-aluminum). After 45 days of storage, both joints passed the 1250 psig leak tests. When the plastic-jacketed gasket joint was disassembled, it was found that the flanges were corroded on the portion of the groove outside the gasket. The corrosion present on that portion of the groove which was inside the gasket was expected. Investigation revealed that sufficient permeation of the nitrogen tetroxide had occurred to mix with atmospheric moisture and cause corrosion of the outer groove surface. The all-metal, pressure-energized gasket also passed the post-test leakage check. The Teflon coating tended to flake off the exposed area of the gasket and this could present serious contamination problems with some systems. The results of these tests are shown on Table VII.

l. Salt Spray and Storage Tests

Salt spray and storage tests were conducted with the bare aluminum Conoseal gasket in a transition joint (CRES-to-aluminum) with bare aluminum grooves only. The rest of the aluminum half of the test cell was

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(31) Henson, F. M. and Inouye, F., Dimensional Instability of Aluminum Alloys for Extreme Low Temperature Cycling Applications, NASA Report No. CR-54829, 31 January 1966

TABLE VII

N_2O_4 FLUID RESISTANCE TEST RESULTS

(Leakage in cc/sec/inch of circumference)

GASKET MANUFACTURER AND TYPE	PART NUMBER	FLANGE MATERIALS	INITIAL LEAK (N_2)	INITIAL LEAK (N_2O_4)	LEAK AFTER 2 DAYS STORAGE	LEAK AFTER 5 DAYS STORAGE	LEAK AFTER 30 DAYS STORAGE	LEAK AFTER 60 DAYS STORAGE	N_2O_4 DRAINED & 4 DAYS STORAGE LEAK TESTED 65th DAY	CONDITION OF HARDWARE
HARRISON K-SEAL	12195CR 3.375	347 CRES to 347 CRES	$2.4x10^{-5}$	$2.4x10^{-5}$			$2.4x10^{-5}$	$2.4x10^{-5}$	$2.4x10^{-5}$	TEFLON FLAKING AT INSIDE OF GASKET
	12195CR 4.375	347 CRES to 347 CRES	$3.1x10^{-5}$	$3.1x10^{-5}$	$3.1x10^{-5}$	$3.1x10^{-5}$				SATISFACTORY
RASO ENG. & A.A.CREATH CO. RASO SEAL	1007-4.050	347 CRES to 7075-T6 ALUM	$2.6x10^{-5}$	$2.6x10^{-5}$	$2.6x10^{-5}$	$2.6x10^{-5}$				SATISFACTORY
	1007-5.750	347 CRES to 7075-T6 ALUM	$1.8x10^{-5}$	$1.8x10^{-5}$			$1.8x10^{-5}$	$1.8x10^{-5}$	$1.8x10^{-5}$	ALUMINUM GROOVE SURFACES CORRODED OUTSIDE OF GASKET INDICAT-ING PERMEATION THRU SEAL JACKET.
	1007-7.250	347 CRES to 7075-T6 ALUM	$1.5x10^{-5}$	$1.5x10^{-5}$			$1.5x10^{-5}$	$1.5x10^{-5}$	$1.5x10^{-5}$	SAME AS ABOVE.

anodized. The joint passed the leakage tests before, during, and after salt spray and storage. However, the gasket and the exposed bare surface of the aluminum flange were found to be corroded at disassembly. The program was terminated before this test could be repeated with the chromate conversion coated gaskets and grooves.

m. Vibration Tests

The results of the vibration tests are presented on Table VIII. The ability of a joint to withstand vibration depends upon many factors (i.e., the stiffness of the flanges and fasteners, the stress levels induced in the joint, the ability of the gasket to follow deflections and distortions in the joint while maintaining sufficient sealing stresses). Stress levels (and deflections) in the joint were not determined during any of these vibration tests; therefore, the deflections induced are not known.

The Raco seal was tested in both the preliminary (heavy duty) and the flight weight joint designs. No change in the leakage rates were noted either during or after completion of the vibration tests. No appreciable loss in bolt torque values was noted after the test. There was no apparent distortion or deterioration of the groove surfaces or gasket after the test.

Two flightweight Conoseal joint designs were tested; a six-inch single seal tube connection (CRES-to-CRES) using V-band couplings and a bolted six-inch dual seal transition (CRES-to-Aluminum) joint. Copper and 321 CRES gaskets were used in the V-band coupling joint and 6061-T6 Aluminum gaskets were used in the bolted dual seal joint. No change in leakage rates were noted either during or after completion of the tests. There was no appreciable loss in fastener torque values noted after the test. No distortion or deterioration of the groove surfaces or gaskets were apparent after the test.

n. Vibration - Low Temperature Tests

The results of the low temperature vibration tests are presented on Table IX. Careful consideration must be given to the effects of very low temperature upon the ability of the gasket to withstand vibration. Most of the elastomeric materials tend to spall the gasket corners at the sealing interface if the temperature is below the brittle point (Tg) of the material. The gasket will eventually fail even if small relative movements occur between the flange and gasket. Some of the elastomers will fracture like glass. However, satisfactory joints have been achieved for this service using the elastomers with high squeeze concepts.[32] A temperature-actuated design[33] was used successfully in the liquid oxygen system of the Titan I Engine (see Figure No. 75).

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(32) Weitzel, D. H., et al, Design of Static Elastometric Seals for Cryogenic Temperatures, Proceedings of the Conference on Design of Leak Tight Separable Connectors, March 1964, NASA, MSFC Huntsville, Alabama

(33) Logan, S. E., Static Seal for Low Temperature Fluids, Journal of the American Rocket Society, Vol. 25, No. 7, July 1955

TABLE VIII

VIBRATION TEST RESULTS

Manufacturer and Flange Size	Nominal Size	Gasket Materials	Flange Type	Flange Material: Face	Flange Material: Groove	Test Cell Pressure psia	Test Method	Test Description	Time to Failure	Leak Rate	Remarks
Hill Engrg. Div. H. K. Breath Co. (Unco Seal) Spec 10011	4-3/8-in.	IPS Jacket, 300 Series CRES Spring	Heavy Duty Bolted (Single Seal)	300 Series CRES	7075-T6 Alum.	2.00 psig.	MIL-H-5272 Procedure XIV	5 to 2,000 cps sinusoidal up to 20 "G" (Longitudinal and Transverse planes.	No Failure after 30 minutes in each plane	$<2.4\times10^{-9}$ (N_2)	No bolt torque loss. No distortion or deterioration of groove surfaces or gasket.
	3-3/8-in.	IPS Jacket, 300 Series CRES Spring	Flight Weight Bolted	300 Series CRES	300 Series CRES	[illegible] psia	MIL-H-5272 Procedure XIV	5 to [illegible] cps sinusoidal up to 20 "G" (Longitudinal and Transverse planes	No failure after 30 minutes in each plane	$<3.1\times10^{-5}$ (N_2)	No bolt torque loss. No distortion or deterioration of groove surface or gasket.
Garman Div. Aerojet Corp. (Unocal)	6-in.	321 CRES & Copper	Flight Weight V-Jouling (Single Seal)	14-4340 CRES	14-4340 CRES	[illegible] psig	MIL-H-5272 Procedure XIV	5 to 2,000 cps sinusoidal up to 20 "G" (Longitudinal and Transverse planes	No failure after 30 minutes in each plane	$<1.4\times10^{-9}$ (He)	No fastener torque loss. No distortion or deterioration of groove surfaces or gasket.
	6-in.	[illegible]-T6 Alum.	Flight Weight Bolted (Dual-Seal)	300 Series CRES	[illegible]-T6 Alum.	155 psig	MIL-STD-810 Method 514	Random [illegible] cps spectral density (Longitudinal & Transverse planes.	No Failure	[illegible] to [illegible] (He)	No bolt torque loss. No distortion or deterioration of groove surfaces or gasket.

TABLE IX

LOW TEMPERATURE VIBRATION TEST RESULTS

Manufacturer and Trade Name	Nominal Size	Gasket Materials	Flange Type	Flange Materials Male	Flange Materials Female	Test Cell Pressure (psig)	Test Cell Temp.	Test Method	Test Description	Time to Failure	Leak Rate (cc/sec/inch)	Remarks
Raco Aerosp. Div. A.E.Creath Co. (Raco Seal) [illegible]	3-3/8 in.	PFE Jacket, 300 Series CRES Spring	Flight Weight, Bolted	300 Series CRES	300 Series CRES	900 psig	<-310°F	MIL-E-5272 Procedure XIV	5 to [illegible] cps sinusoidal up to [illegible] (Longitudinal & Transverse planes)	No failure after [illegible] minutes in each plane	<[illegible] N_2	No appreciable bolt torque loss. Jacket crazed at [illegible] junction ridge at flat flael flange interface.
Garlock Div. Garlock Corp. (Garlock)	6 in.	321 CRES	Flight Weight V-Coupling	14-4DL CRES	14-4DL CRES	[illegible] psig	<-310°F	MIL-E-5272 Procedure XIV	[illegible] sinusoidal up to [illegible] (Longitudinal & Transverse planes)	No failure after [illegible] minutes in each plane	<[illegible] N_2	No flatness torque loss. No flat stain or interpenetration of [illegible] gasket.
Aerojet Standard [illegible] Pure Connection [illegible] Spring	6 in.	Teflonex per MIL-[illegible]	Flight Weight V-Coupling	[illegible] CRES	[illegible] CRES	[illegible] psig	<-310°F	MIL-E-5272 Procedure XIV	[illegible] sinusoidal up to [illegible] (Longitudinal & Transverse planes)	No failure after [illegible] minutes in each plane	<[illegible] N_2	No appreciable flatness torque loss. [illegible] indication of deterioration of the grooves or gaskets.

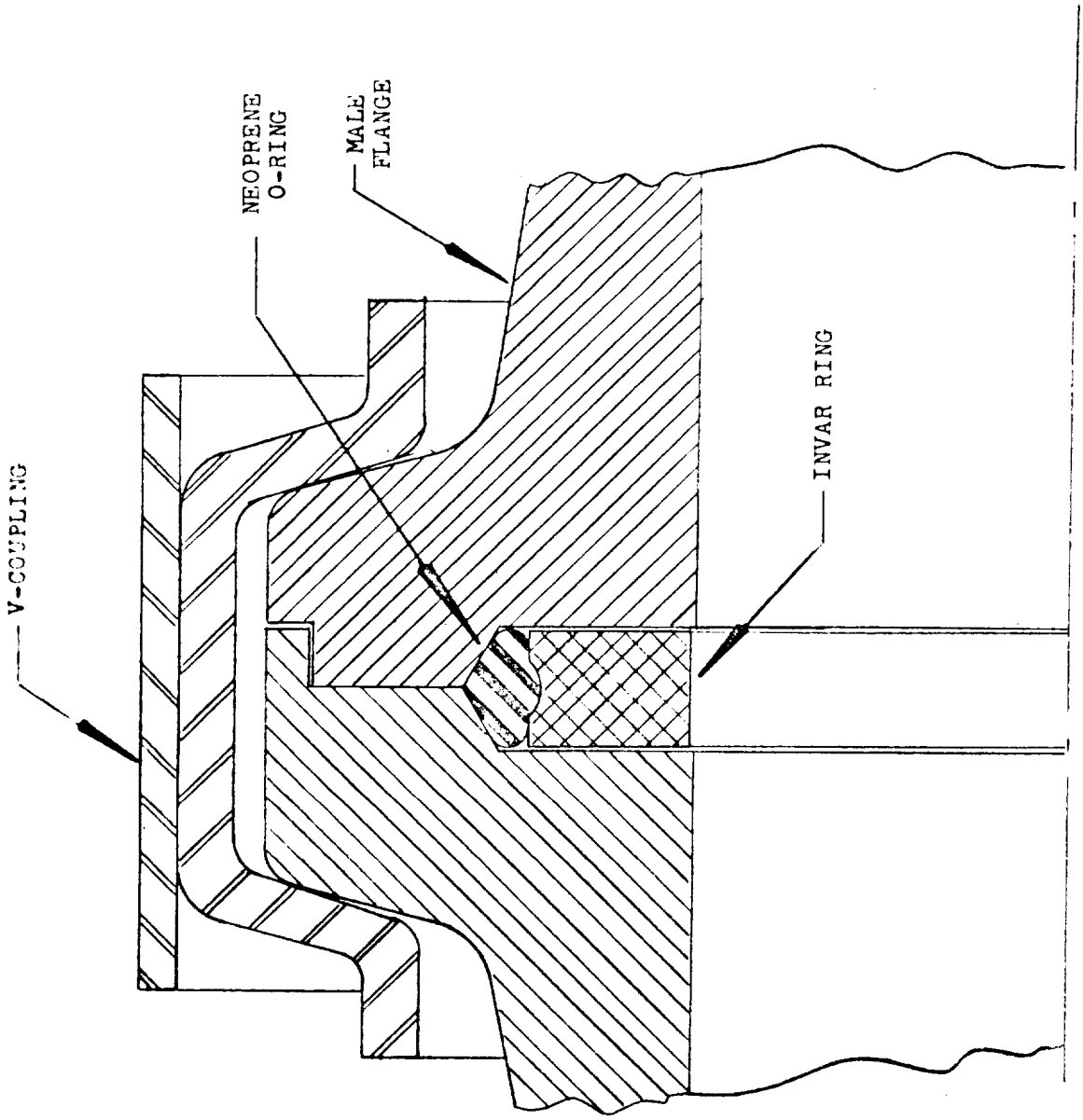

Figure 75
Tube Connection for Liquid Oxygen Service
(Aerojet-General Standard ASD5018)

Minor crazing of the jacket at the outside diameter-to-leg junction (nearest the flat-faced flange) was noted on the Raco seal after completion of the test. No appreciable difference in initial and final leakage rates was noted; therefore, it was assumed that the crazing was not detrimental. No appreciable loss in bolt torque values were noted.

No appreciable change in leakage rate or fastener torque values was noted for the flightweight V-band coupling Conoseal joint after the test. No distortion or deterioration of the groove surfaces or gasket was apparent.

No increase in leakage of the flightweight Aerojet-General standard tube connection (see Figure No. 75) was noted after completion of the test. No appreciable loss in fastener torque values or apparent spalling or crazing of the neoprene O-ring was noted.

o. Burst Tests

The results of the preliminary burst tests with the heavy duty flanges are shown on Figure No. 76. The ability of the gasket to withstand burst pressure is dependent upon the flexibility of the flanges and the strength of the gasket section as well as the gasket materials. The ability of the elastomeric O-rings is usually limited by the extrusion characteristics. If the flange deflection is sufficient to allow extrusion, the gasket will "blow out." The same applies to the plastic-jacketed gaskets. The all-metal, pressure-energized gaskets usually do not present the sudden failure problem encountered with the elastomers and plastic gaskets, but leakage will increase sharply when the deflection capabilities of the gasket are exceeded. The solid-section, diametral gaskets will open up suddenly if differential radial expansion between the flanges becomes too great. The flexible section gaskets will bow and relieve sealing stresses if axial deflection capabilities are exceeded and leakage will also suddenly increase if differential radial deflection capabilities are exceeded.

2. Component Tests

Leakage data were collected during the development of the thrust chamber valves and the gas generator valves designed for use on the M-1 Engine (1.5-million lb thrust, liquid oxygen/liquid hydrogen). These valves are shown schematically on Figures No. 77, No. 78, and No. 79. The dual Conoseal design was used in these valves. The results of tests with these components are presented as Figure No. 80 and No. 81. A dimensional instability of the forged aluminum valve bodies caused by thermal cycling was encountered during development. This problem was discussed in detail in a previous report.[34] All the data were taken from tests with the assemblies having unstabilized bodies and 6061-O aluminum Conoseal gaskets. Only two thermal cycling tests were conducted with stabilized bodies (6061-T6 aluminum

(34) Henson, F. M. and Inouye, F., op. cit.

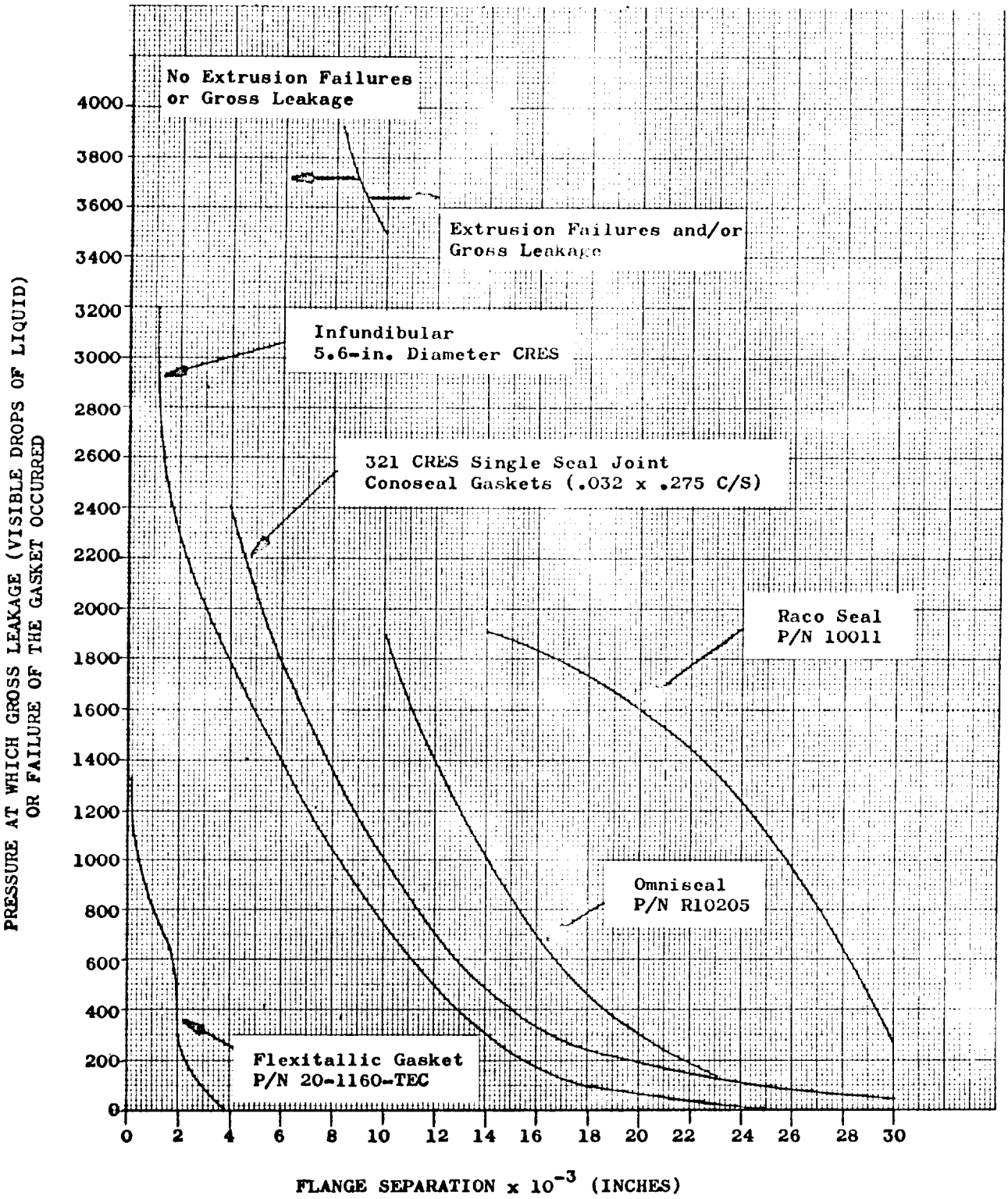

Figure 76
Burst Test Results Ambient Temperature

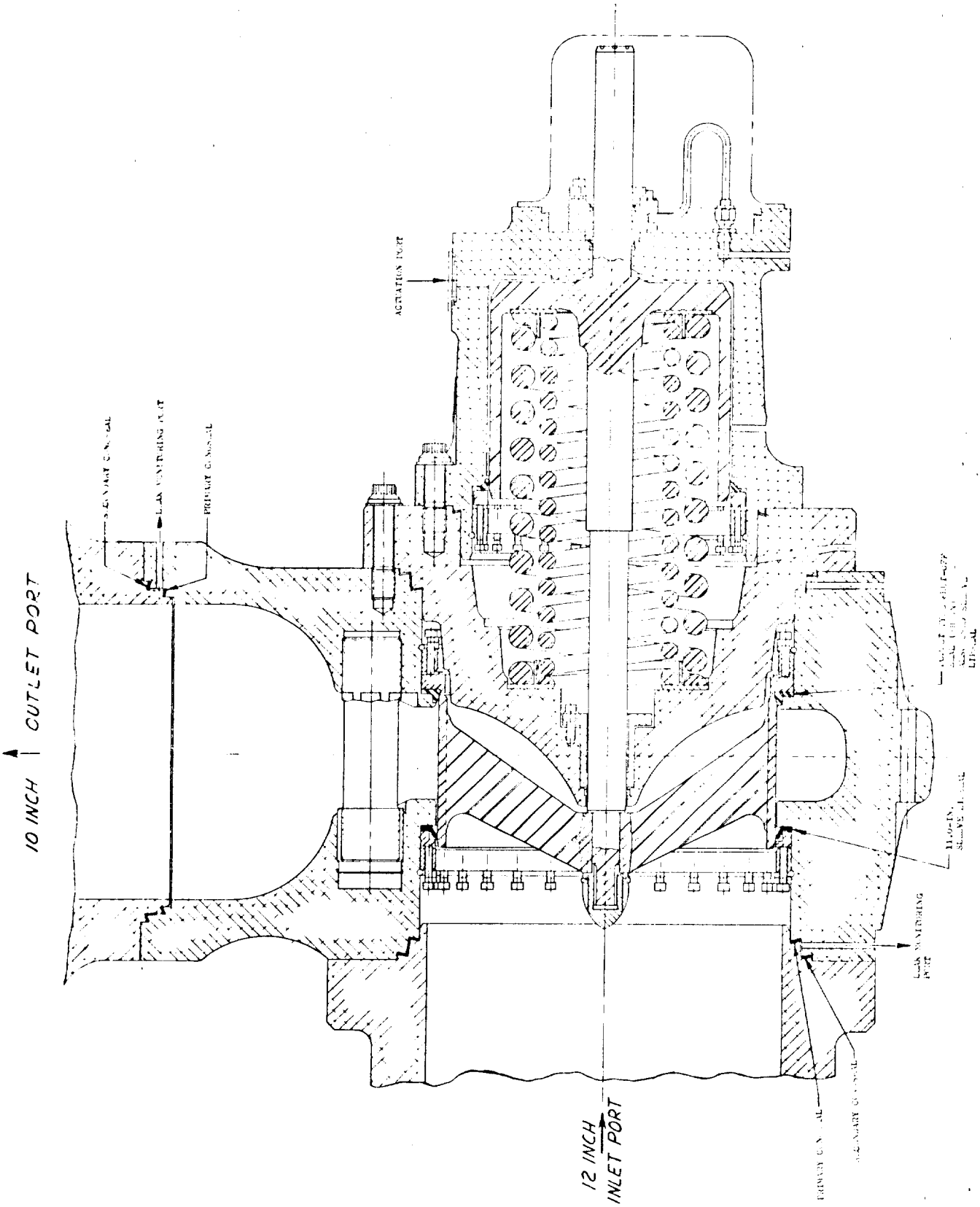

Figure 77
M-1 Thrust Chamber Valves

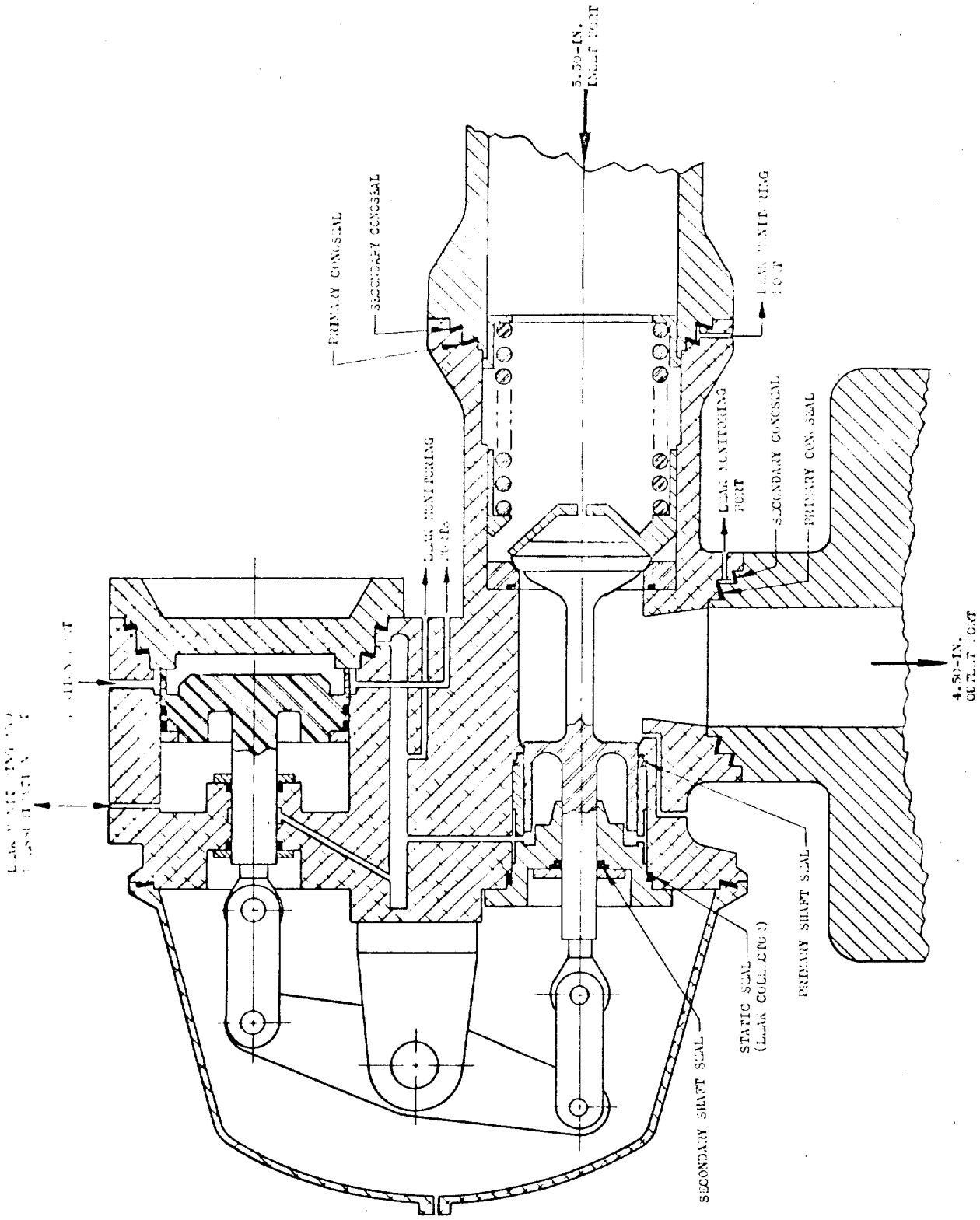

Figure 78
M-1 Gas Generator Fuel Valve

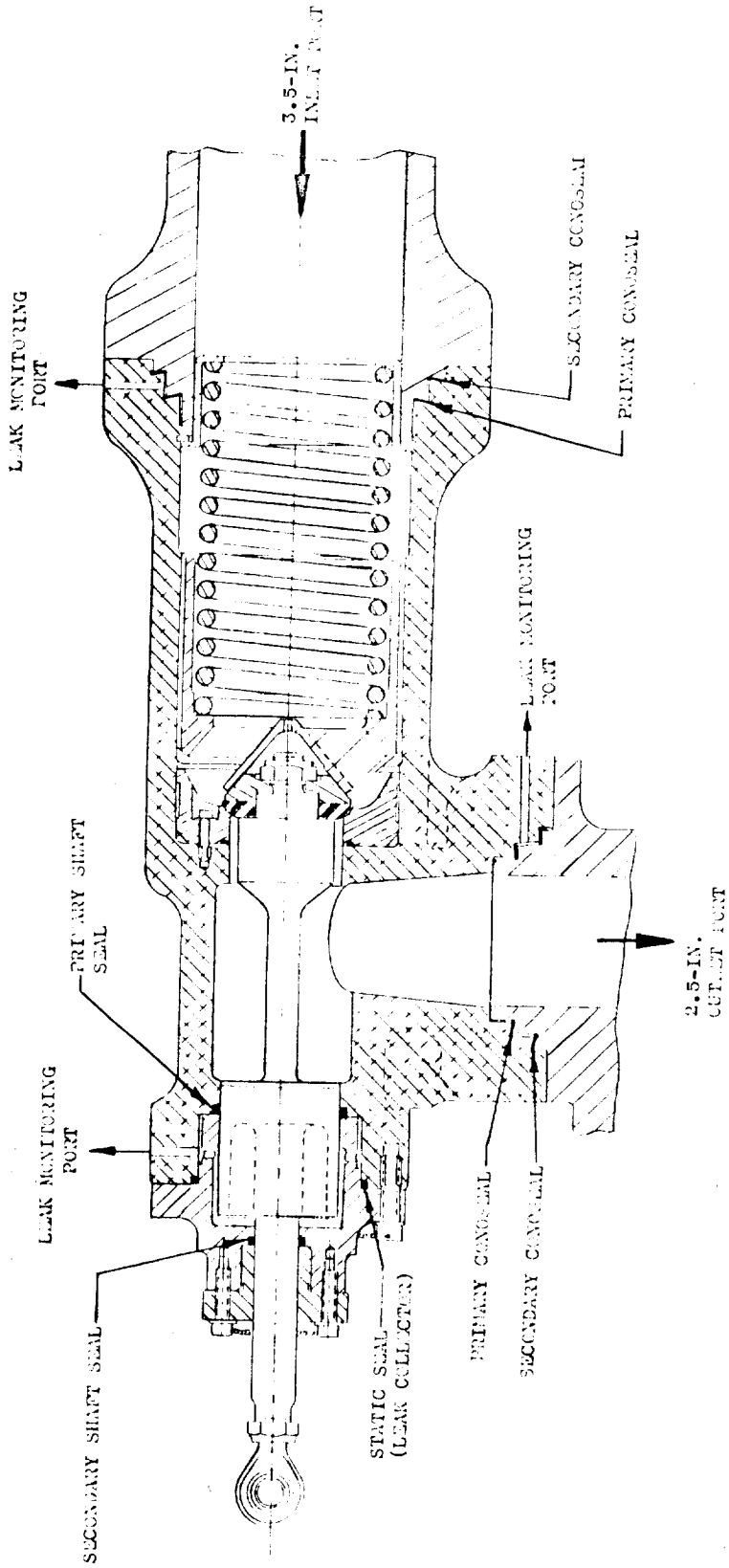

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Figure 79

M-1 Gas Generator Oxidizer Valve

C-3

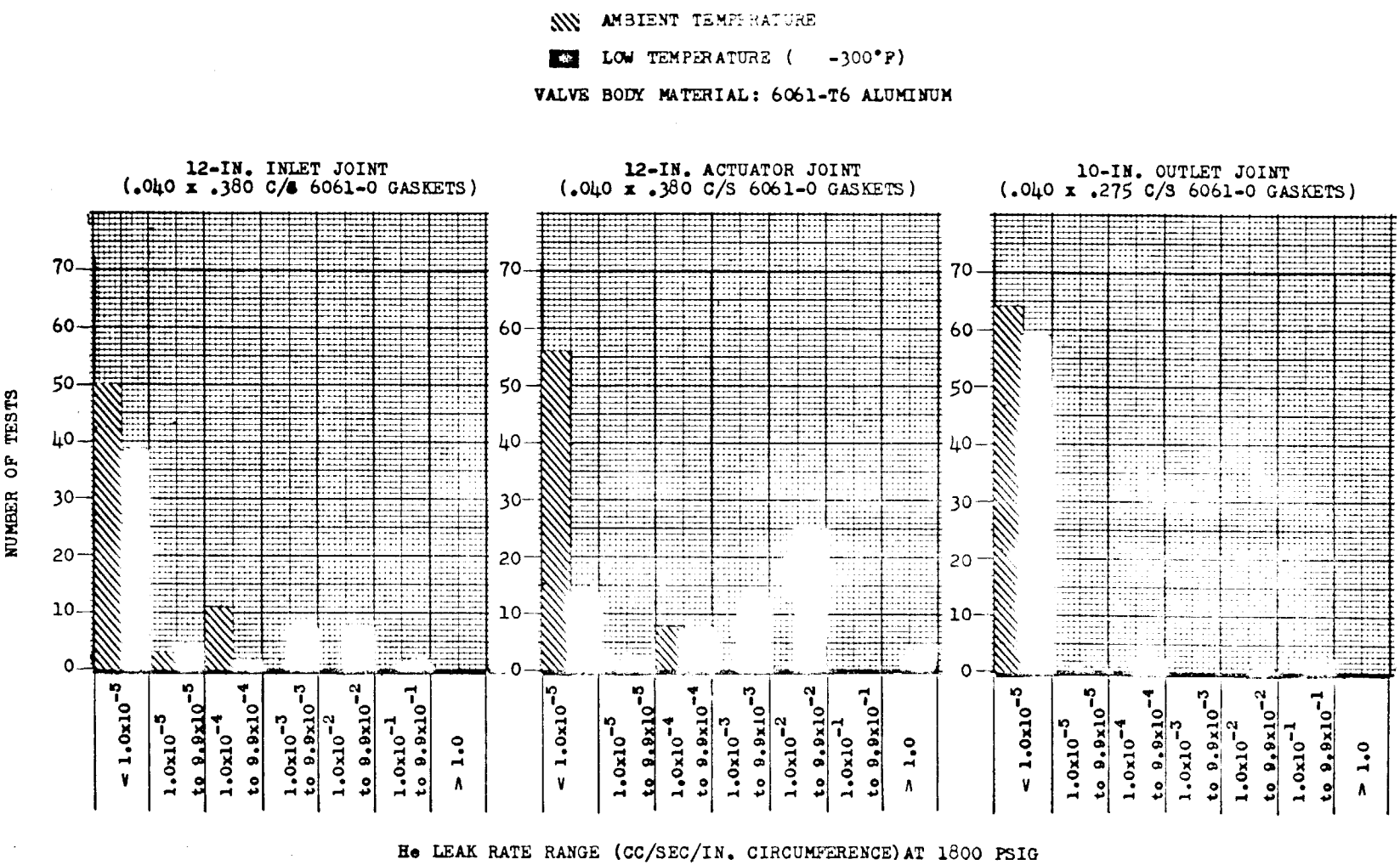

Figure 80
Results of Tests of the M-1 Thrust Chamber
Valve Dual Conoseal Joints

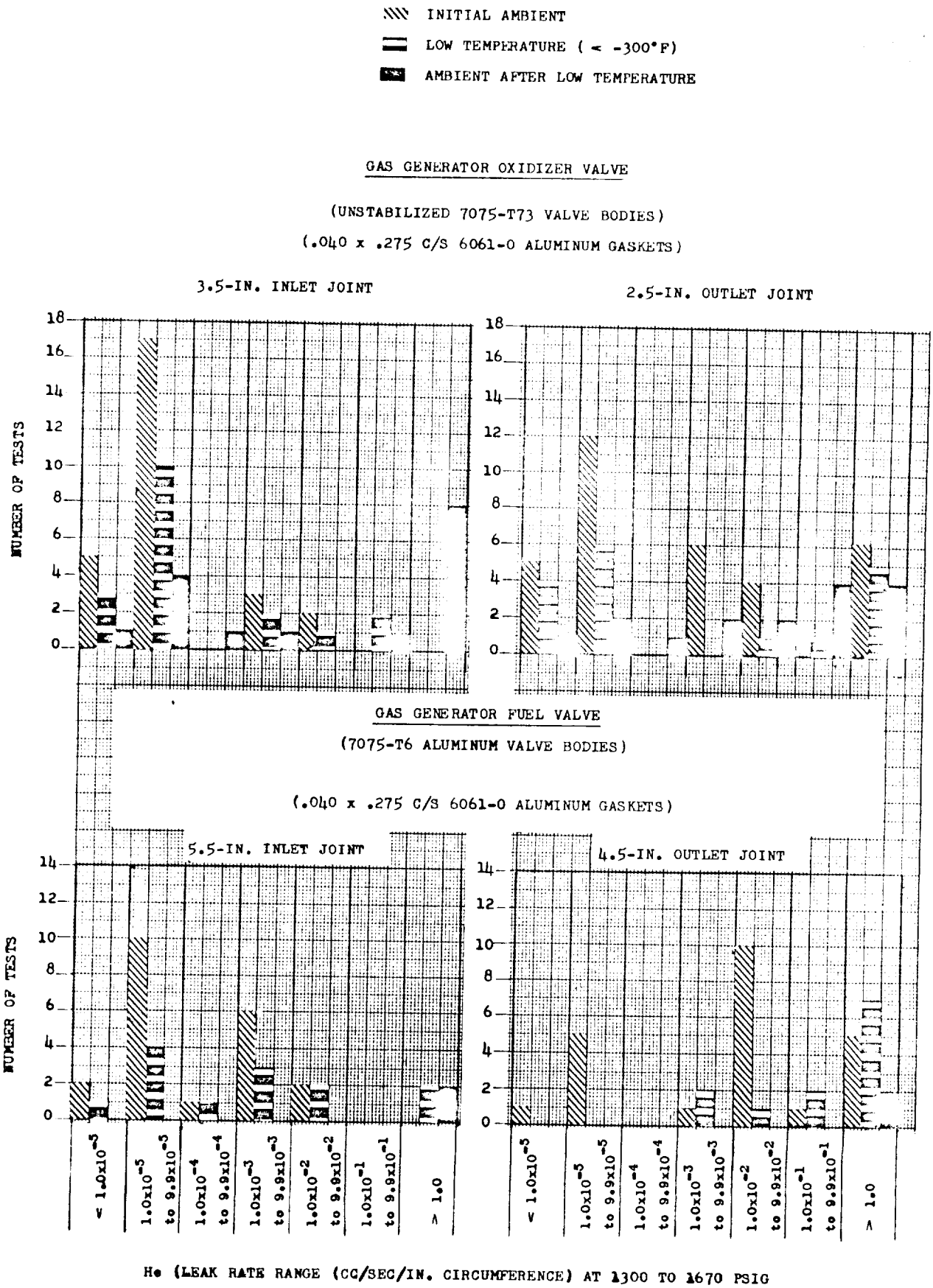

Figure 81
Results of Tests of the M-1 Gas Generator
Valve Dual Conoseal Joints

gas generator oxidizer valve) using 6061-T4 gaskets before the program was terminated. Leakage rates were 4.8×10^{-8} to 5.9×10^{-7} cc/sec/in. during the initial ambient temperature leak tests. It was 1.84×10^{-9} to 6.4×10^{-9} cc/sec/in. at low temperature and 6.2×10^{-8} to 6.0×10^{-7} cc/sec/in. at ambient after the low temperature test.

3. System Tests

Post-firing leak test data were collected from a 10 production engine sample of a storable propellant liquid engine system. Engine firing durations were 20 sec, 60 sec, and 160 sec. The propellants were nitrogen tetroxide (oxidizer) and AeroZINE-50 ® (fuel) and they were pump-fed from a hot-gas-driven turbopump. The results are shown on Table X and Figure No. 82. Though post-fire leakage was detected in some cases (predominantly in the hot gas system), there was no visual leakage noted during any of the runs. In addition there were no instances of a critical malfunction or premature shut-down caused by excessive leakage in the 10 engine sample.

D. CONCLUSIONS

The best method for eliminating leaks in a system is to eliminate mechanically sealed joints. This must be considered from the over-all system design. Sealed joints should be placed in the system only where absolutely necessary for component replacement or the ease of maintaining the system. Methods should be devised for welding or brazing all other joints in the system. Where sealed joints are required, and "zero" leakage is absolutely essential, methods should be devised to weld or braze these joints closed after final acceptance of the system. This necessitates considerable investigation during the initial design phase to provide for disconnecting these joints, if required.

No particular seal design is completely satisfactory for all applications. Some gaskets can provide lower leakage rates than others, but they may not be the best selection for a particular application. The gasket selection must be based upon many factors (i.e., fluid compatibility, materials compatibility with other parts of the joint, leak rate capability, operating temperatures and pressure, gasket deflection capabilities or joint weight). In general, gaskets that provide the highest sealing stress are capable of the lowest leakage rate, but require the highest pre-loading in the flanges. Methods for providing high sealing stresses with low loading by reducing sealing contact area aggravate the problem of damaged sealing surfaces.

The methods used in protecting and handling the finished gaskets or flanges influence whether the joints obtained are good or bad. It is important to evolve proper methods for protecting and handling each type of gasket from the time that the parts are finished to the time that they are installed. Protection and proper handling of mating flanges are also extremely important in connection with the over-all joint capabilities. With some types

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TABLE X
(Sheet 1 of 2)

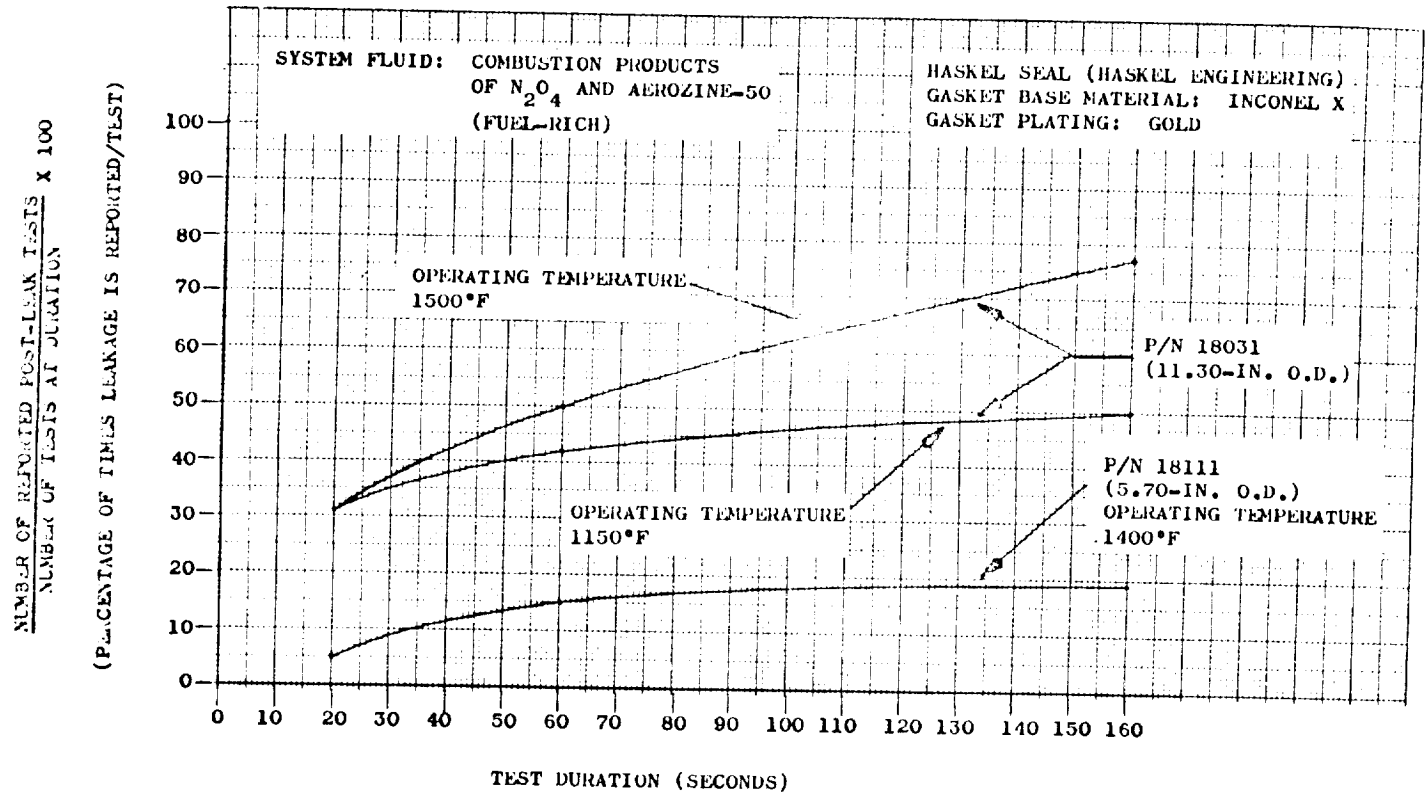
RESULTS OF POST-ACCEPTANCE TEST LEAK CHECKS OF A STORABLE PROPELLANT ENGINE SYSTEM SEALS

ENGINE SYSTEM	SYSTEM FLUID	GASKET MANUFACTURER AND TRADE NAME	GASKET P/N	NOMINAL SIZE	OPERATING TEMP. RANGE	OPERATING PRESS. RANGE	TOTAL NO. REPORTED LEAKS (POST-TEST) / TOTAL NO. OF TESTS
Fuel	Aerozine-50	Various Aerojet Standard (Butyl Rubber O-Rings)	AS4004-010	.070 C/S X .239 I.D.	32°F to 120°F	40 psig to 900 psig	0/92
			-012	X .364 I.D.			
			-014	X .489 I.D.			
			-018	X .739 I.D.			
			-019	X .801 I.D.			
			-020	X .864 I.D.			
			-024	X1.114 I.D.			
			-026	X1.239 I.D.			
			AS4004-028	.070 C/S X1.364 I.D.			
			AS4004-120	.103 C/S X .987 I.D.	32°F to 120°F	40 psig to 900 psig	0/92
			-125	X1.239 I.D.			
			-130	X1.489 I.D.			
			AS4004-144	.103 C/S X2.487 I.D.			
			AS4004-212	.139 C/S X .859 I.D.	32°F to 120°F	40 psig to 900 psig	0/92
			-214	X .984 I.D.			
			-235	X3.109 I.D.			
			-239	X3.609 I.D.			
			-242	X3.984 I.D.			
			-243	X4.109 I.D.			
			-246	X4.484 I.D.			
			-247	X4.609 I.D.			
			-249	X4.859 I.D.			
			-252	X5.234 I.D.			
			-259	X6.234 I.D.			
			AS4004-266	.139 C/S X7.984 I.D.			
		Raco Engineering Div'n. R. E. Creath Co. (Raco Seal)	10005-11.800	3/16 C/S X11.800 O.D.	32°F to 120°F	40 psig to 900 psig	0/92
			10005-9.005A	3/16 C/S X 9.50 I.D.			1/92
			10011-03.620	1/8" C/S X03.620 O.D.	150°F	800 psig	0/92
			10011-21.796 A	1/8 C/S X21.796 I.D.	400°F	800 psig	3/92
Oxidizer	N_2O_4	Raco Engineering Div'n. R. E. Creath Co. (Raco Seal)	10005-10.550	3/16 C/S X10.55 O.D.	32°F to 120°F	40 psig to 800 psig	1/92
			10005-7.755A	3/16 C/S X7.755 I.D.			1/92
			10011-4.375	1/8 C/S X4.375 O.D.		800 psig	0/92
			10011-5.500	1/8 C/S X5.500 O.D.		800 psig	0/92
			10011-7.250	1/8 C/S X7.250 O.D.		800 psig	0/92
			10011-7.442	1/8 C/S X7.442 O.D.		40 psig	0/92
			10011-8.250	1/8 C/S X8.250 O.D.		40 psig	0/92
			10011-21.976	1/8 C/S X21.976 O.D.		700 psig	2/92
		Reid Enterprises and/or Aeroquip Corp. (Omniseal)	R10205-214A1N	1/8 C/S X1.250 O.D.	32°F to 120°F	600 psig	0/92
			R10205-216A1N	1/8 C/S X1.375 O.D.		600 psig	0/92

TABLE X
(Sheet 2 of 2)

RESULTS OF POST-ACCEPTANCE TEST LEAK CHECKS OF A STORABLE PROPELLANT ENGINE SYSTEM SEALS

ENGINE SYSTEM	SYSTEM FLUID	GASKET MANUFACTURER AND TRADE NAME	GASKET P/N	NOMINAL SIZE	OPERATING TEMP. RANGE	OPERATING PRESS. RANGE	TOTAL NO. REPORTED LEAKS (POST-TEST) / TOTAL NO. OF TESTS
Hot Gas	Combustion Products of N_2O_4 and AeroZINE-50	Haskel Engineering (Haskel Seals)	18031	1/8 C/S X11.30 O.D.	1500°F	400 psig	See Figure No. 82
					1150°F	100 psig	
			13111	1/8" C/S X5.70 O.D.	1400°F	400 psig	
		Flexitallic Gasket Co.	9-2155-1PAJ	1/8" C/S X3.81 O.D.	1670°F	490 psig	See Figure No. 82
			5-1565-1PAJ	1/8" C/S X1.18 O.D.	1100°F	100 psig	
					1400°F	400 psig	
			R4-15F	2.50 O.D.	1400°F	350 psig	



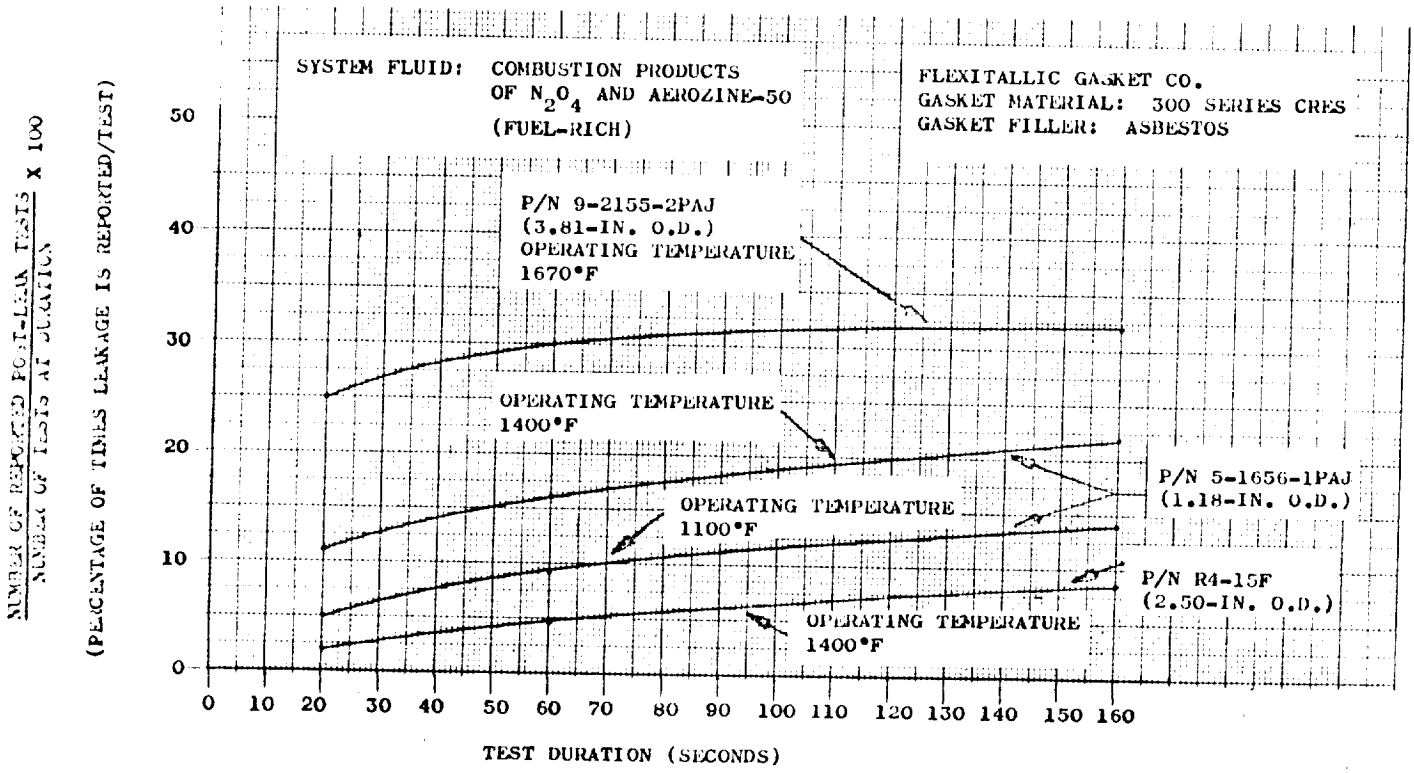


Figure 82
Results of Post-Acceptance Test Leak Checks
of Hot Gas System Seals

of gaskets, even slightly damaged mating or gasket surfaces will prevent the attaining or maintaining of the desired low leakage rate.

The design of the other parts of the joint (flanges and fasteners) is as important a factor as the design of the gasket in obtaining a good joint. The gasket capability must be determined and the design of the joint must be adequate to satisfy all of the requirements for the gasket. Before a particular joint design is made final, thorough development and evaluation testing should be performed to assure that all the requirements for the gasket have been satisfied in the joint design. The capabilities and design requirements of a number of the proprietary gaskets is not readily-available from some suppliers. With other suppliers, the capabilities and requirements for any of their standard configurations have been thoroughly analyzed and developed. It is necessary for the designer to be knowledgeable of these requirements before the preliminary design of the joint can be made.

Double-seal joint designs compound the number of variables in the joint design. Interactions between the two seals and the mating grooves, or between two sealing points, can cause a considerable reduction in the primary seal capability. Also, all dual seal joint designs are larger and heavier than single seal designs, particularly in applications where a metal seal must be used. If the primary seal leakage is limited to very low gaseous leakage rates (less than 1.0×10^{-5} cc/sec), the secondary seal must be nearly as good as the primary seal to prove leakage capabilities. The methods available to detect these low leakage rates require a vacuum between the primary and secondary seals; therefore, if the secondary seal is not good, apparent leakage may be from this source rather than from the primary seal.

Assembly procedures are also extremely important in obtaining a satisfactory joint. This is the point where all the properly designed parts come together and can be misapplied if the assembly procedure and requirements are not fully understood. Each type of gasket or joint design can require considerably different methods of assembly. Proper lubrication, tightening, and the tightening sequence of the fasteners can make the difference in a completely satisfactory or completely unsatisfactory joint.

E. RECOMMENDATIONS

If "zero" leakage is absolutely necessary, sealed joints should be eliminated in favor of welded or brazed connections. At least careful consideration should be given to methods for welding or brazing the joint shut after final acceptance testing of the system. Although dual seal joint designs are a reasonably good method for assuring low external leakage, the additional problems in joint design, weight, and assembly should be carefully considered with respect to the over-all system before attempting to use this concept. Once the requirement for the dual Conoseal design was determined and designed for, this concept performed very satisfactorily and is highly recommended where very low gaseous leakage rates are required. However, the supplier should be contacted before attempting to use this concept to obtain the necessary design information for the joint.

Additional testing and evaluation of the various seal designs is recommended for the new high pressure (3000 to 6000 psig) fluid systems proposed for aerospace service. Many of the gaskets tested were limited in maximum pressure and would not be suitable for high pressure systems. New designs by the various suppliers for this service would be necessary, particularly for hot gases. These should be further investigated and evaluated.

A large number of improvements in test procedures and instrumentation are required to accurately determine the many gasket capabilities. Additional and continuous investigations should be conducted to improve test methods, instrumentation, and test procedures. Possibly, a group of standardized test procedures could be developed (similar to ASTM methods) for evaluating or qualifying various gasket designs. Seals could be classified according to pressures, temperatures, and other capabilities. These seals could then be tested to determine their maximum deflection capabilities as well as other requirements. Standardized procedures, test methods, and instruments would provide comparative data from each evaluator.

A preliminary standardized test procedure for use in static seal and gasket evaluations was developed at Aerojet-General. The basic purpose of the procedure, which is continuously being expanded and improved, is to obtain maximum relative data using a minimum of testing and test equipment.

By standardizing groove machining requirements, many of the variables involved in the evaluation of the various gaskets could be eliminated. It is strongly recommended that such a standardization program be initiated particularly for the "pressure-energized" face seal designs. The groove standard should be based upon the original MIL-P-5514 face seal O-ring grooves so that O-rings could be used at the interfaces of components for ambient temperature production acceptance testing as well as other non-qualification tests. The Aerojet-General standards designed around this concept are presented in Appendix C. The gasket (part) standards are designed by the different manufacturers to fit these grooves. In addition, all pre-qualification and qualification testing is conducted with flanges (joints) containing these grooves.

IV. SMALL TUBING JOINTS

A. TECHNICAL DISCUSSION

1. Types of Small Tubing Joints

There are three basic types of small tubing joints; the welded or brazed, the swaged, and the threaded connections. The same basic considerations apply to small tubing joints as to the larger static seal joints. If the joint is not absolutely necessary, it should be eliminated. If the joint is necessary, it could possibly be welded or brazed. There are a number of new concepts in welded and brazed joints that warrant investigation. There are also new concepts in the swaged joints that present possibilities for good, solid connections. Where none of these possible solutions are feasible and a method for easily disconnecting the joint is necessary, the threaded connection must be used. Each of these concepts of small tubing connections are described in this discussion. The basic types of small tubing connections are shown on Figure No. 83.

There are a number of types as well as methods available for welded and brazed joints for small tubing connections. The simplest, which is a permanent method, is the resistance butt-welded joint. There are two known methods used for disconnectable welded joints. Both methods use relatively thin flange sections that are actually melted at the outer tips of the flange section to make the connection. One design accepts loading by shear in the weld joint and the other design accepts loading in tension with the associated bending in the weld joint. Brazed joints are available where electrical, torch, or pyrotechnic devices are used to heat the joint to the brazing temperature. The tooling required to make the connections for these joints varies from the relatively bulky torch or electrical heating devices to a simple dry cell battery. Disconnection methods also vary from re-heating, where the same tubing is used for connection, to sawing or grinding the fitting from the tubing.

Many swaged methods and fitting designs have been devised for tubing connections. However, most methods are primarily for swaging tubing onto mechanically connectable flanges and require that the tubing be rolled or pressure-formed from the inside of the tube. Only one method was found that could be used to connect two tubing ends together by cold forming from the outside of the tube. This method actually cold forms the fitting onto the tube outside diameter for the connection with one simple operation. It requires a relatively bulky hydraulic pressure device to make the connection and the fitting must be cut out of the line for disconnection.

There are many types of threaded connections available. Some of the threaded connections require the tubing to be welded or brazed to each half of the connector. Others use methods where the tube is swaged to each half of the connector. There are also the military standard flared and flareless tubing connections which are the most widely-used of all the threaded connections. Each flared or flareless tube connection actually contains two

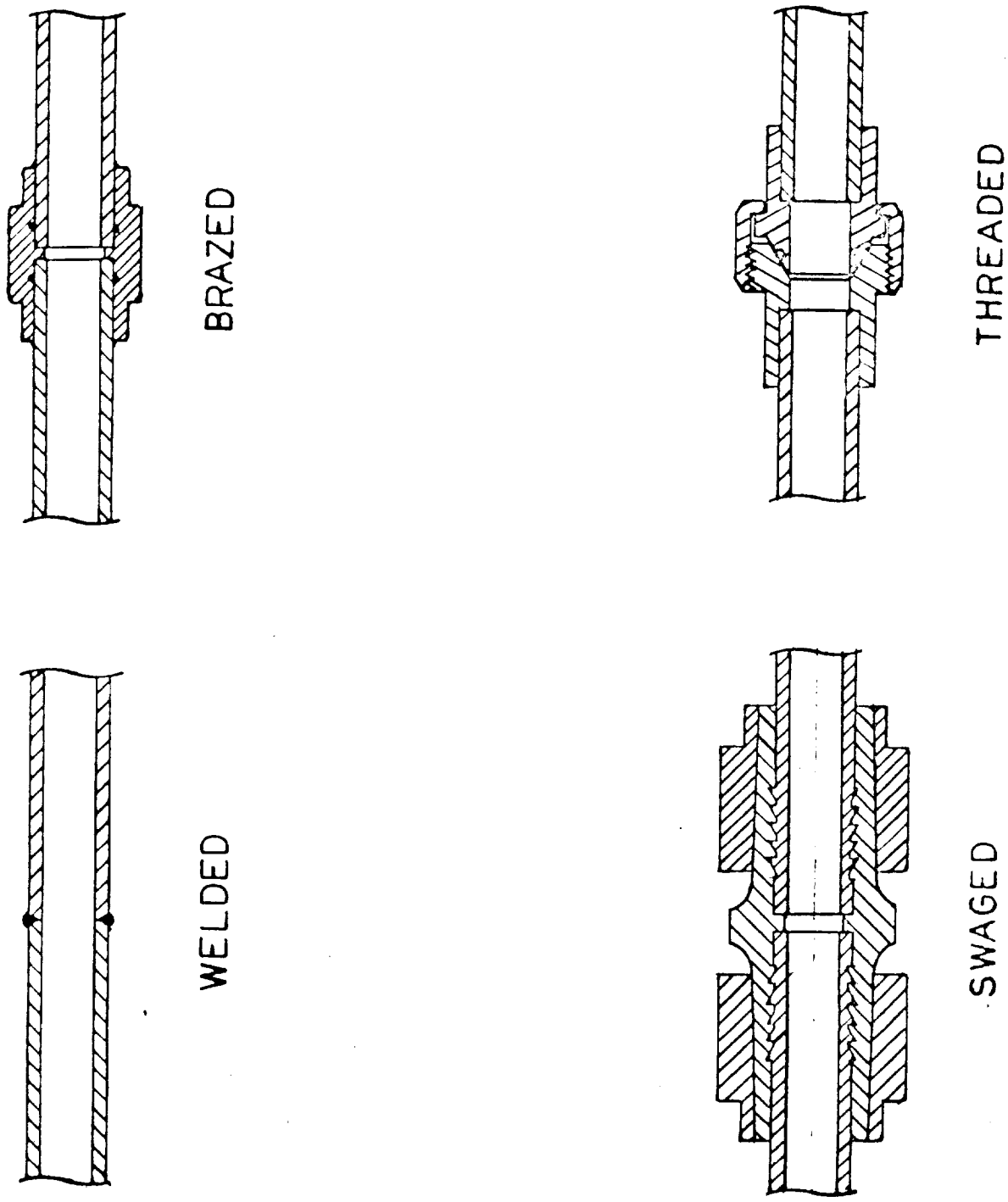


Figure 83
Basic Small Tubing Joint Configurations

joints; one for each end of the fitting. The basic requirements for sealing of the threaded connectors are the same as those specified in the static seals section (paragraph III, A, 1) and the design requirements are also essentially the same. Sufficient plastic (or elastic) deformation must take place at the sealing interface to achieve the initial seal and the joint must be designed so that the necessary minimum seating stresses are maintained throughout the life of the connector.

2. Advantages and Disadvantages of the Small Tubing Joint Types

Until recently, welded and brazed joints were difficult to use in a complete system assembly because methods and tooling were not available for either expeditious or reliable welding and brazing. Recent developments in techniques and tooling have made these joints much more attractive for system assembly.

The advent of new tools for electron-beam welding and improved controls provide a relatively easy method for making the welded connection. Fittings, tooling, and techniques have been devised that make the brazed fitting a relatively simple operation. The tooling for one method (the Aeroquip "SpaceCraft" fitting) can be used for both connection and disconnection. However, most of the welded and brazed joints require that the connection be sawed or cut from the line and can present some serious contamination problems during this process. The resistance butt-welded joints could probably be reassembled if the tubing is carefully cut and, at the same time, redressed for welding with tooling using a thin cutter designed to clamp to the tubing and travel around the tube. The other welded joints are designed to be ground-off and enough material is left to reweld the joint. These connections are not too useful for higher pressures unless the joint loads are taken by a threaded section. Also, a crack or notch is left in the weld joint because full penetration of the interface is not made at the weld. This crack can progress through the weld after a few pressure cycles have caused some movement (deflection) between the flanges. Most of the welded and brazed joints require somewhat cumbersome equipment (primarily because of the associated electrical and fluid lines) and tools to make the connection, except for one brazed joint (the Deutsch "Pyrobraze") which uses pyrotechnics. This unit requires only a six-volt dry cell battery to make the joint.

The swaged fitting connection is a permanent one that must be cut from the lines. The tool used to make this connection is a somewhat bulky (although easy to use), hydraulically-actuated scissor that squeezes the swaging locks over the fitting. The making of a new connection requires that two fittings be installed with a short stub tube section between each fitting.

Connections to components using welded, brazed, or swaged joints require a tubular-shaped boss on the component. If the type of connection used must be cut from the line, the length of the tubular boss must be sufficient to allow a pre-determined number of assembly and disassembly operations, unless the Aeroquip "SpaceCraft" brazing system is to be used. The "SpaceCraft" brazing system provides for "de-brazing" the connection.

Threaded connections are easily assembled and disassembled with standard wrenches and torque wrenches. The purchased complete connection assemblies require each half of the connection to be welded or swaged to the tubing or component. These also require a tubular boss on the component for attaching one half of the connection. Most of the purchased connections require the male threaded half to be installed on the component, which leaves the threaded section exposed to possible damage. These connections have the advantage over the standard flared and flareless connections of requiring only one sealed joint, which axiomatically improves the reliability providing that the welded, brazed, or swaged joint is properly made. Some types require relatively critical alignment between the two tube sections (or the tube and the component) to make a good joint.

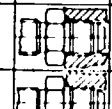
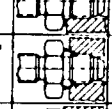
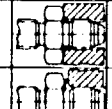
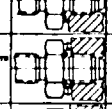
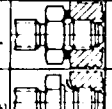
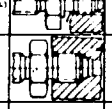
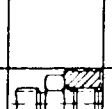
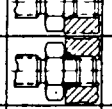
As previously mentioned, the standard flared and flareless connection each contain two sealed joints; one at each end of the fitting. However, tooling has long been established for manufacturing the flared section of the tube for the flared tube connection and no flaring or particularly special preparation is needed for the flareless tube connection. The boss (and unexposed female thread) for the component connection also has established standards and tooling. The male thread is only on the fitting and if these threads are damaged, they can be easily replaced. This also applies to the sealing surfaces of the fitting. If sealing surfaces on the tubing are damaged, the tubing must be replaced. The same applies to the somewhat exposed sealing surfaces on the component bosses.

3. Engineering Evaluation

a. Capabilities

Before selecting a connector design for a particular application, it is necessary to evaluate each of the various types with respect to the application. First, it should be determined what type of connection is required or desirable. The welded, brazed, or swaged connections should be investigated in relationship to the problems expected with components as well as the possible number of assemblies that may be required for each particular joint in the system. If it is decided that the threaded connections are required, the best configuration for the application should be selected from the numerous possibilities. However, to examine each possible connection becomes a difficult task. Probably, the first basic rule is that in ambient liquid systems, the standard flared or flareless tube connections are used while for gaseous systems (particularly the liquified gaseous and high temperature gas systems), the selection becomes more difficult. A reasonably complete list of the various types of connections and the basic requirements for each is presented on Figure No. 84 (sheets 1 through 7). The information presented was gained from standards, drawings, and specifications, available literature, catalogue information, discussions with manufacturers, and to a lesser extent, actual laboratory testing. Figure No. 84 was used as a basis for selecting complete joints (in the case of the proprietary items), parts of joints (in

FITTING-TO-BOSS GASKET

MANU-FACTURER	CONFIG-URATION	APPROX. COST		AVAILABLE MATERIALS	P/N	MACH REQUIR.			FITTING REQUIREMENTS				BOSS REQUIREMENTS			INSTALL RED PROCEDURES	OPERATING TEMP RANGE	REMARKS	TEST EXPERIENCE				
		3/8"	3/4"			SEAL SURF FINISH	FITTING OD RUNOUT	THREAD CLASS	STD.	SURFACE FINISH	ROUNDNESS	CONCEN OD RUNOUT	STD	SURFACE FINISH	CONCEN. OD RUNOUT				TEST PRESS	TEST MEDIA	TEST TEMP	LEAK RATE	REMARKS
[illegible]				[illegible]	[illegible]	[illegible]			[illegible]	[illegible]			[illegible]	[illegible]		[illegible]	[illegible]	[illegible]					
[illegible]				[illegible]	[illegible]		[illegible]		[illegible]	[illegible]			[illegible]	[illegible]		[illegible]	[illegible]	[illegible]					
[illegible]				[illegible]	[illegible]				[illegible]				[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]				[illegible]	[illegible]				[illegible]				[illegible]			[illegible]	[illegible]	[illegible]					
[illegible]				[illegible]	[illegible]				[illegible]				[illegible]			[illegible]	[illegible]	[illegible]					
[illegible]				[illegible]	[illegible]				[illegible]				[illegible]			[illegible]	[illegible]	[illegible]					
[illegible]				[illegible]	[illegible]				[illegible]				[illegible]			[illegible]	[illegible]	[illegible]					
[illegible]				[illegible]	[illegible]				[illegible]				[illegible]			[illegible]	[illegible]	[illegible]					
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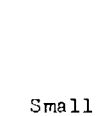
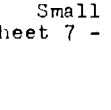
[illegible]					[illegible]				[illegible]				[illegible]					[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]					[illegible]				[illegible]				[illegible]					[illegible]					

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Fig 84-13

190-13

Figure 84 - A
Small Tubing Joint Review
Sheet 7 - Fitting-to-Boss Gasket

100-A

FOLDOUT FRAME 1.

FOLDOUT FRAME 2.

TUBING FLARE-TO-FITTING GASKETS

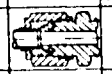
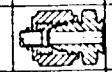
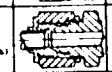
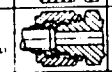
MANU-FACTURER	CONFIG-URATION	APPROX. COST 3/8"	APPROX. COST 3/4"	AVAILABLE MATERIALS	P/N	MACH REQUIR. SEAL SURF FINISH	MACH REQUIR. CONCEN OR RUNOUT	MACH REQUIR. THREAD CLASS	TUBING REQUIREMENTS STD.	TUBING REQUIREMENTS SURFACE FINISH	TUBING REQUIREMENTS ROUNDNESS	TUBING REQUIREMENTS CONCEN OR RUNOUT	BOSS REQUIREMENTS STD.	BOSS REQUIREMENTS SURFACE FINISH	BOSS REQUIREMENTS CONCEN. OR RUNOUT	TORQUE REQ'D INCH-LBS	OPERATING TEMP. RANGE	REMARKS	TEST EXPERIENCE TEST PRESS	TEST EXPERIENCE TEST MEDIA	TEST EXPERIENCE TEST TEMP	TEST EXPERIENCE LEAK RATE	TEST EXPERIENCE REMARKS
[illegible]				[illegible]	[illegible]				[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]					NO ASC TEST EXPERIENCE.
[illegible]				[illegible]	[illegible]				[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]					NO ASC TEST EXPERIENCE.
[illegible]				[illegible]	[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1500	O_2 Gas	-320°F to +200°F	BUBBLE TIGHT TO 10 CC/MIN	NAS TESTS OF NICKEL GASKETS.
[illegible]				[illegible]	[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1500	O_2	-320°F to +200°F	BUBBLE TIGHT	NAS TESTS OF NICKEL AND COPPER GASKETS.
[illegible]				[illegible]	[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]					NO ASC TEST EXPERIENCE.

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Fig. 84-A

189-A

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Figure 84 B
Small Tubing Joint Review
Sheet 6 - Tubing Flare-to-Fitting Gasket - 9B

FOLDOUT FRAME 1

FOLDOUT FRAME 2

REPLACEMENT FLARED TUBE FITTINGS & "B" NUTS (COUPLINGS)

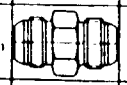
MANU-FACTURER	CONFIG-URATION	APPROX. COST 3/8"	APPROX. COST 3/4"	AVAILABLE MATERIALS	STD.	MACH. REQUIR. SEAL SURF. FINISH	MACH. REQUIR. CONCEN. OR RUNOUT	MACH. REQUIR. THREAD CLASS	TUBING REQUIREMENTS STD.	TUBING REQUIREMENTS SURFACE FINISH	TUBING REQUIREMENTS ROUNDNESS	TUBING REQUIREMENTS CONCEN. OR RUNOUT	BOSS REQUIREMENTS STD	BOSS REQUIREMENTS SURFACE FINISH	BOSS REQUIREMENTS CONCEN. OR RUNOUT	TORQUE REQD INCH-LBS	OPERATING TEMP. RANGE	REMARKS	TEST EXPERIENCE TEST PRESS	TEST EXPERIENCE TEST MEDIA	TEST EXPERIENCE TEST TEMP.	TEST EXPERIENCE LEAK RATE	TEST EXPERIENCE REMARKS
[illegible]									[illegible]				[illegible]					[illegible]	0 TO 1000	[illegible]	60°F	ZERO BUBBLES (5 MIN)	[illegible]
[illegible]				SAME MATERIALS AS "AN" FITTINGS					[illegible]				[illegible]					STANDARD FLARED FITTINGS WITH SERRATED SURFACES IN BODY AND TAPLOK OR BOSS SEALS.	1500	GN_2, GHe	-80°F TO +150°F	BUBBLE TIGHT TO 1 CC/MIN	
[illegible]									[illegible]				[illegible]					TWO-PIECE FITTING WITH SEPARATE METAL "BELLOWS" SEAL.					NO SOC TESTS.
[illegible]									[illegible]				[illegible]			3/8" - 405 TO 450 3/4" - 1350 TO 1500	-425°F TO +1200°F	REPLACES AN818 BUT USES [illegible] SLEEVE. INCREASED STRENGTH AT SLEEVE LOADING POINT. ALLOWS HIGHER TORQUE WITH OF LOCK STRADDING BUT AND SLEEVE. MORE DIRECT LOADING ON TUBING.	1500	GN_2, GHe	-320°F TO +200°F	BUBBLE TIGHT TO 1 CC/MIN	
[illegible]				C-1137 STL AL. ALLOY					[illegible]				[illegible]				MORE ALT SAME AS STD. B-NUT	REPLACES AN818 B-NUT. TWO-PIECE NUT WITH LOCKING FEATURE SIMILAR TO "PELL-NUT". INFINITELY ROTATE FLANGE LOSS PROBLEM.	1500	GN_2	AMB.	BUBBLE TIGHT TO TO ONLY TO [illegible]	HIGH TIGHTENING TORQUE REQUIRED TO SEAL JOINT.
[illegible]									[illegible]				[illegible]				UP TO 800°F	REPLACES AN818 B-NUT. NUT INCORPORATES OUT-OF-ROUND TO PROVIDE LOCKING FEATURE IN THREADS. SILVER PLATED. SAFETY WIRE HOLES CAN BE PROVIDED.	—	—	—	—	INCONSISTENT NUT PRE-SET TORQUE. NO LEAK TESTS.
[illegible]				CRES TITANIUM AL. ALLOY					[illegible]				[illegible]				-425°F TO 1200°F	REPLACES AN818 B-NUT AND MS20819 SLEEVE. NUT AND SLEEVE MADE OF SAME MATERIAL.					NO SOC TESTS.

Figure 84 A
Small Tubing Joint Review
Sheet 5 - Replacement Flared Tube Fittings
and "B" Nuts (Couplings)

188 D

Fig. 84 B

188 B

FOLDOUT FRAME 1

FOLDOUT FRAME 2

FLARELESS TUBE CONNECTIONS

MANU-FACTURER	CONFIG-URATION	APPROX. COST 3/8"	APPROX. COST 3/4"	AVAILABLE MATERIALS	STD.	MACH. REQUIR. SEAL SURF. FINISH	MACH. REQUIR. CONCEN. OR RUNOUT	MACH. REQUIR. THREAD CLASS	TUBING REQUIREMENTS STD.	TUBING REQUIREMENTS SURFACE FINISH	TUBING REQUIREMENTS ROUNDNESS	TUBING REQUIREMENTS CONCEN OR RUNOUT	BOSS REQUIREMENTS STD.	BOSS REQUIREMENTS SURFACE FINISH	BOSS REQUIREMENTS CONC. EN. OR RUNOUT	TORQUE REQD INCH-LBS	OPERATING TEMP. RANGE	REMARKS	TEST EXPERIENCE TEST PRESS	TEST EXPERIENCE TEST MEDIA	TEST EXPERIENCE TEST TEMP	TEST EXPERIENCE LEAK RATE	TEST EXPERIENCE REMARKS
[illegible]				[illegible]		[illegible]	[illegible]	[illegible]		[illegible]			[illegible]	[illegible]		[illegible]		[illegible]	0 TO 2700 PSIG	N_2	[illegible] TO [illegible]°F	BUBBLE TIGHT TO LEAKAGE	[illegible]
[illegible] FITTING CO. "[illegible]"				ALUMINUM 316 CRES [illegible] CARBON STEEL MONEL PTFE					NO SPECIAL REQUIREMENTS EXCEPT CRES TUBING [illegible] AVAILABLE [illegible] CRES IN [illegible] WALL [illegible]					AVAILABLE WITH LARGE HEX SIZE AND STRAIGHT THREAD [illegible] WITH [illegible] PIT ARRANGEMENTS.		NONE 1 1/4 TURNS PLUT BACK TIGHT POSITION						NO ASC TESTS TO DATE.
[illegible] CO. "[illegible]"				ALUMINUM 316 CRES [illegible] CARBON STEEL		[illegible]			NONE --- CRES TUBING [illegible] ALSO FOR [illegible] TUBE END [illegible] FLARING WITH [illegible]					AVAILABLE [illegible] ONLY FOR PIPE END. [illegible]		NONE 1 1/4 TURNS PLUT BACK TIGHT POSITION						NO ASC TESTS TO DATE.
[illegible] "[illegible]"				304 CRES 321 CRES 17-4PH [illegible] STEEL PTFE, ETC.		[illegible]			NONE --- FITTINGS AVAILABLE FOR ALL TYPES OF TUBE					[illegible]		NOT SPECIFIED		CLAIMED ON "GRIPS" INTO BOTH TUBE I.D. AND O.D. AT TUBE END (SELF-FLARING). CLAIMED BETTER SELF-LOCKING ACTION OF THE NUT.					NO ASC TESTS TO DATE.
[illegible] "[illegible]"				[illegible] 316 CRES		[illegible]			TUBE END [illegible] 1/2".	[illegible]				AVAILABLE FOR [illegible] PIT PATTERN [illegible]		NONE 1/60 TO 1/40 TURN PLUT BEFORE [illegible]						NO ASC TESTS.
[illegible] CO. "[illegible]"				ALUMINUM CRES TITANIUM		[illegible]		5						AVAILABLE WITH [illegible] PIT ARRANGEMENTS.		NONE 1/60 TO 1/20 TURN PLUT BEFORE [illegible]						NO ASC TESTS.
[illegible] CORP. "SELF-FLARE"				321 CRES CARBON STEEL 304 L3 ALUMINUM PLASTIC		[illegible]			TUBE END [illegible]					AVAILABLE WITH [illegible] PIT ARRANGEMENTS.		[illegible] "SELF-SETTING" [illegible]		SAME AS SELF-FLO TOC.					NO ASC TESTS.
																							NO ASC TESTS.

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Fig 84-11
18)-A

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Figure 84-13
Small Tubing Joint Review
Sheet 4 - Flareless Tube Connections

1 / 3

FOLDOUT FRAME 1

FOLDOUT FRAME 2

FLARED TUBE CONNECTIONS

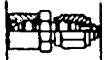
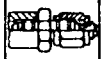
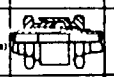
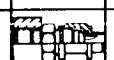
MANUFACTURER	CONFIGURATION	APPROX. COST 3/8"	APPROX. COST 3/4"	AVAILABLE MATERIALS	STD.	MACH. REQUIR. SEAL SURF. FINISH	MACH. REQUIR. CONCEN. OR RUNOUT	MACH. REQUIR. THREAD CLASS	TUBING REQUIREMENTS STD.	TUBING REQUIREMENTS SURFACE FINISH	TUBING REQUIREMENTS ROUNDNESS	TUBING REQUIREMENTS CONCEN. OR RUNOUT	BOSS REQUIREMENTS STD.	BOSS REQUIREMENTS SURFACE FINISH	BOSS REQUIREMENTS CONCEN. OR RUNOUT	TYPICAL OR OD PER LBS	OPERATING TEMP RANGE	REMARKS	TEST EXPERIENCE: TEST PRESS	TEST EXPERIENCE: TEST MEDIA	TEST EXPERIENCE: TEST TEMP.	TEST EXPERIENCE: LEAK RATE	TEST EXPERIENCE: REMARKS
[illegible]				300 SERIES CRES., 2024-T3 & -T4 ALUM.	[illegible]	100	.005	-3A	FLARE PER MS33561, MS33564, OR AN5220	BORE 32	BORE .001	.005 .005	AN816049 AN816050	100 32	.005	1/8" - 3/4" ± 1° 1/4" - 3/4" ± 1° INSTALLATION PER [illegible]	-423°F TO +1200°F	TWO SEALED JOINTS WITH BACK CONNECTION. INSUFFICIENT SEALING; STEMS MAY ELIMINATE STRUCTURAL STRESS LEVELS FOR SEALING; PROPERLY STRUT. INSUFFICIENT PROBLEMS BETWEEN [illegible] FOR PROP. TO PROPER INSULATION CAUSED BY STEMS FUNCTIONALITY AND CREEP.	0 TO 3000 PSIG	[illegible]	[illegible] TO -120°F	[illegible]	TITAN I AND TITAN II ENGINE EXP PROGRAMS. TUBING DEGRADATION, LEAKAGE.
[illegible]				300 SERIES CRES., 2024-T3 & -T4 ALUM.	[illegible]	100	.005	-3A	FLARE PER MS33561, MS33564, OR AN5220	BORE 32	BORE .001	.005 .005	AN816050	32	.005	1/8" - 3/4" ± 1° 1/4" - 3/4" ± 1° INSTALLATION PER [illegible]	-423°F TO +1200°F	TWO SEALED JOINTS PER CONNECTION. ESSENTIALLY SAME AS "AN" TYPE EXCEPT IMPROVED SEALING SURFACE FINISHES OR PHYSICAL MODIFIED SEALING STRUCTURE. REDUCED FLAR SPECIFICITY TOLERANCES AND PROBLEMS. INSTALLED STRUCTURE IS FREE AND SLEEVE ALLOWS BEARD TORQUE VITERRY FROM STRESSING PARTS. DEPENDS BOND TORQUE RELAXATION PER SLEEVE.	0 TO 1500 PSIG	[illegible]	[illegible] TO -120°F	[illegible] TO 0 CC/MIN.	TITAN II ENG. EXPERIENCE. REDUCED PROBLEMS. STILL SOME BORE LEAKAGE PROBLEMS.
[illegible]				AN955 or 7075-T73 SLEEVES 316 CRES 7075-T73 FITTINGS ETC.	MC172 OR MC175 FIT-TINGS ETC.	22-26 FOR CRES 16-14 FOR ALUM	0.001 AN 0.0001	-3A	FLARE PER MC146	16 FOR CRES 8 FOR ALUM	.0005	.005	MC240	32	.005	[illegible]	-423°F TO +1200°F	TWO SEALED JOINTS PER CONNECTION. MACH IMPROVED SURFACE FINISHES REDUCED SEALING STRESSES. REDUCED FLAR SPECIFICITY TOLERANCES AND PROBLEMS. INSTALLED STRUCTURE IS FREE AND SLEEVE ALLOWS BEARD TORQUE VITERRY FROM STRESSING PARTS. DEPENDS BOND TORQUE RELAXATION PER SLEEVE.					NO SAC TEST EXPERIENCE.
[illegible]				AN955 OR 7075-T73 SLEEVES 316 CRES 7075-T73 ETC.	MC172 & MC175 FIT-TINGS ETC.	22-26 FOR CRES 16-14 FOR ALUM	0.001 AN 0.0001	-3A	FLARE PER MC146	16 FOR CRES 8 FOR ALUM	.0005	.005	MC240	32	.005	[illegible]	-423°F TO +1200°F	SAME CONCEPTS AS "MC" CONNECTORS EXCEPT SELF-LOCKING FIT PROVIDED TO RELIEVE PROBLEM OF BACK-OFF DURING VIBRATION.					NO SAC TEST EXPERIENCE.
[illegible] (ACORN CONNECTOR)				300 SERIES CRES AL. ALLOY. TITANIUM ALLOY.	MANUFACTURER'S STD.				FLARE PER MS33561, MS33564 OR AN5220	BORE 32	.001	.005	FITTINGS AVAILABLE TO FIT STANDARD AN10000 BOSSES.			BORE SPECIFIED 1/4" TO 1/2" THRU PLAT PILOT OF VALVE PORT, ETC.		TWO SEALED JOINTS PER CONNECTION. ALLOWS HIGHER DATIE. FIS OPEN FTS CONNECTIONS.					NO SAC TEST EXPERIENCE.
L.A.A. MFG. CO. (MS-2)					MANUFACTURER'S STD.				FLARE PER MS33561, MS33564 OR AN5220	BORE 32	BORE .001	.005 .005	MS16142	100	.005	1/8" - 1/4" - NO STD. INSTALLATION SPEC. FURNISH SPECS.		TWO SEALED JOINTS WITH BACK CONNECTION. ALSO AVAILABLE IN ELBOWS, TEES, ETC. THAT REQUIRE A SPECIAL (CAPTIVE NUT) PORT.					WORKS WELL ON TITAN II INSTRUMENTATION LINES. NO LAB TESTS.
PARKER AEROSPACE CO. (PAC)				300 SERIES CRES. AL. ALLOY	MANUFACTURER'S STD.				FLARE PER MS33561, MS33564 OR AN5220	BORE 32	BORE .001	.005 .005	MS16142	100	.005	1/8" - 1/4" - NO STD. INSTALLATION SPEC. FURNISH SPEC.		TWO SEALED JOINTS PER CONNECTION. ALSO AVAILABLE IN ELBOWS, TEES, ETC.					NO SAC TEST EXPERIENCE.

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Figure 84 A
Small Tubing Joint Review
Sheet 3 - Flared Tube Connections

Fig. X4B

186 B

FOLDOUT FRAME 1

FOLDOUT FRAME 2

(WELDED, BRAZED, OR SWAGED CONNECTIONS)

MANU-FACTURER	CONFIG-URATION	APPROX. COST 3/8"	APPROX. COST 3/4"	AVAILABLE MATERIALS	MACH. REQUIR. STD.	MACH. REQUIR. SEAL SURF. FINISH	MACH. REQUIR. CONCEN. OR RUNOUT	MACH. REQUIR. THREAD CLASS	TUBING REQUIREMENTS STD.	TUBING REQUIREMENTS SURFACE FINISH	TUBING REQUIREMENTS ROUNDNESS	TUBING REQUIREMENTS CONCEN. OR RUNOUT	BOSS REQUIREMENTS STD	BOSS REQUIREMENTS SURFACE FINISH	BOSS REQUIREMENTS CONCEN. OR RUNOUT	TORQUE REQD INCH-LBS	OPERATING TEMP. RANGE	REMARKS	TEST EXPERIENCE: TEST PRESS	TEST MEDIA	TEST TEMP.	LEAK RATE	REMARKS
	[illegible] (INSERTABLE)			[illegible]		[illegible] FITTING			[illegible]				[illegible]				[illegible]	[illegible]					NO ASC TESTS.
				COULD BE SAME AS TUBING		[illegible] FITTING			[illegible]				[illegible]				[illegible]	[illegible]					NO ASC TESTS.
				COULD BE SAME AS TUBING.		[illegible] FITTING			[illegible]				[illegible]				[illegible]	[illegible]					NO ASC TEST
	ELECTRICAL BRAZE			SAME AS TUBING		FITTING COULD BE LATHING UP MACHINED TO FIT THE TUBE WITH ATTACH. SAME THICKNESS AS TUBE.			TUBE BOSS COULD PROBABLY PASS THRU, BE SLIED FOR CLOSE FIT TO FITTING [illegible]				SAME AS FOR TUBING.				[illegible]	[illegible]					
SURFACE STABALOC CORP. DIV. "STABALOC"				[illegible] CRES		[illegible]			[illegible]		DIA = [illegible] BORED WITH [illegible] DIA. TOL.	TUBE OR BOSS BORE MUST BE MACHINE WITH [illegible] & [illegible] BORES.	TUBULAR BORE	[illegible]	DIA = [illegible] BORED WITHIN DIA. [illegible]		[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	
AEROQUIP CORP. "SPACE-CRAFT" JOINTS				300 SERIES CRES		[illegible]			[illegible]		DIA = [illegible] BORED WITHIN DIA. TOL.	TUBE OR BOSS BORE SURFACE FITS [illegible] BORES. TOOLS AVAILABLE FOR TUBE OPERA-	TUBULAR BORE	[illegible]	DIA [illegible] BORE WITHIN [illegible]		[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	NO ASC TESTS
PARKER AIRCRAFT CO.				321 CRES 6061T6 CRES 1-CORRA I 2075-T6 ALUM ALLOY		[illegible]			[illegible]				BELLOWS TUBULAR BORE REQ'D.				[illegible]	[illegible]					NO ASC TESTS
VOI-SHAN PRC CO.				CRES PER AMS5765 321 CRES TITANIUM		125 MIN			MORE SPECIFIED				MORE SPECIFIED				[illegible]	[illegible]					
PARKER AIRCRAFT CO.	"AF" FITTING			INCONEL "X" AMS5542, AMS5750 300 SERIES CRES 2075-T6 AL. ALLOY		[illegible]	[illegible]		[illegible]		TUBE BOSS SURFACE WITH [illegible] & WITHIN [illegible] AND FLAT WITHIN [illegible] DIA. [illegible] OF NORMAL		TUBULAR BORES REQUIRED OF COMPONENT; SAME AS TUBE				[illegible]	[illegible]	[illegible]	AND TO -300°F	LESS THAN 1.0×10^{-4}		FIBRATION, LOW TEMP., PRESSURE CYCLES TESTS.

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Fig. 84 B

1848

Figure 84 A
Small Tubing Joint Review
Sheet 1 - Welded, Brazed, or Swaged Connections

184 - A

FOLDOUT FRAME 1

FOLDOUT FRAME 2

COMPLETE THREADED CONNECTION ASSEMBLIES

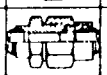
MANU-FACTURER	CONFIG-URATION	APPROX. COST 1/8"	APPROX. COST 1/4"	AVAILABLE MATERIALS	MACH REQUIR. STD.	MACH REQUIR. SEAL SURFACE FINISH	MACH REQUIR. CONCEN. OR RUNOUT	MACH REQUIR. THREAD CLASS	TUBING REQUIREMENTS STD.	TUBING REQUIREMENTS SURFACE FINISH	TUBING REQUIREMENTS ROUNDNESS	TUBING REQUIREMENTS CONCEN. OR RUNOUT	ROSS REQUIREMENTS STD	ROSS REQUIREMENTS SURFACE FINISH	ROSS REQUIREMENTS CONCEN. OR RUNOUT	TORQUE REQD INCH-LBS	OPERATING TEMP. RANGE	REMARKS	TEST EXPERIENCE TEST PRESS	TEST EXPERIENCE TEST MEDIA	TEST EXPERIENCE TEST TEMP	TEST EXPERIENCE LEAK RATE	TEST EXPERIENCE REMARKS
[illegible]				[illegible]	[illegible]	✓			[illegible]				[illegible]				[illegible]	[illegible]					NO ASC TEST TO DATE.
[illegible]				[illegible]	[illegible]	✓			[illegible]				[illegible]			1/8" - [illegible] 1/4" - [illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]				[illegible]	[illegible]	✓			[illegible]				[illegible]			1/8" - 1/4" -	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	
[illegible]				[illegible]	[illegible]				[illegible]				[illegible]			1/8" - 1/4" -	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	
[illegible]				[illegible]	[illegible]	✓ ✓			[illegible]				[illegible]			1/8" - [illegible] 1/4" - [illegible]	[illegible]	[illegible]					NO ASC TEST TO DATE.
[illegible]				[illegible]	[illegible]	✓			[illegible]				[illegible]			1/8" - 1/4" -	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	
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Fig 84 P
185-11

Figure 84 P
Small Tubing Joint Review
Sheet 2 - Complete Threaded Connection Assemblies

185 B

the case of supposed improvements over the standard connectors), and gaskets for laboratory evaluation for a broad range of services. The connections, fittings, and seals have been separated by types: welded, brazed, and swaged; threaded connections that can be purchased complete; and the standard flared and flareless tube connections. The interchangeable standard flared tube fittings are also separated for easy reference, as are the various types of couplings (i.e., nuts and fittings) that are interchangeable with the standards. The gaskets are designed for either the coupling or boss ends of the flared tube connections and are classified accordingly.

The materials available in the parts or complete connections as actual standards or as suppliers standards are included. The machining requirements column applies only to the critical dimensions for the sealing interface and any applicable standards. In the case of the welded, brazed, or swaged connections, this applies to the actual fitting (if one is required). The tubing requirements section presents the standard for tubing and/or the flaring operation and specifies the necessary surface finishes, roundness of the tube or flare, and concentricity or run-out of the tube inside diameter to outside diameter or flare (angle-to-tube diameter). The required boss that must be machined on the component is also specified along with any critical surfaces or runout of surfaces to the threads.

The required torque values are specified for the 3/8-in. and 3/4-in. sizes for comparison purposes only. Any applicable standards for torque values are also specified in this column.

The operating temperature range specifies the range of temperature at which the connection, fitting, and seal can be used considering the available materials for all the parts. This is not intended to mean that completely satisfactory leakage rates will be attained with a particular connection, fitting, or seal over the range of temperature specified. This is a function of the designed stresses for the particular connection at the specified torque, the possibility of stress relaxation, and many other factors. The temperature range specified considers only that the materials available for the connection can be used over this temperature range. However, in some cases, the supplier has limited the temperature service range based upon his knowledge of the stresses in the connection. In these instances, the values are specified as such.

The remarks column provides a brief summary of the favorable and unfavorable points in the connection, fitting, or seal. Possible problems are also indicated, but this is not intended to be a detraction from the connection because other factors can be more important for a particular application. It is intended only to alert the designer for further studies regarding the particular application.

The test experience section is a summary of actual laboratory evaluation of particular connections, fittings, and seals with the fluids, temperatures, pressures, and resultant leakage rates shown.

b. Design Practices

The forces tending to separate or cause deflections in the small tubing joints are the same as those for the static sealed joints. The same design proof (or burst) pressures to prove the joint are also applicable. The joint must be capable of withstanding the sum of proof (or burst) pressures, axial tension, bending moment, and shear loads, while maintaining the required minimum sealing stress in the gasket (for the threaded connection) or it must prevent separation of the joint (for the welded, brazed, or swaged connections) under all operating conditions and environments during the life of the joint.

The type of small tubing joints selected for a particular application must be based upon many factors. First consideration should be given to the more permanent welded, brazed or swaged connection. Each of these connections present their own unique set of problems, which will be subsequently discussed; however, if the essential precautions are taken and proper tooling is used, they are the most reliably leak-tight and trouble-free joints available. Where the connection is to a component in the system (which is usually the case), the experience and confidence in the reliability of the component must be thoroughly considered. If it is desirable to provide a capability for removing the component a number of times, the cylindrical boss or weld tip on the component must be considered. The length of the tubing required to permit cutting the connection for a specified number of times while maintaining sufficient boss or tubing length to make the new connection must also be considered. This depends upon the type of joint and/or the length as well as the number of fittings needed to accomplish the required number of assemblies. Good design for welded, brazed, or swaged joints are no less an engineering problem than the threaded connections. They are probably as much a fabrication and field (if a component must be replaced) problem as the standardized threaded connections, but the gain in reliability and leak-tight joints make them very attractive. The manufacturer of the purchased brazed connections should be thoroughly informed of system fluids, pressures, temperatures, and operating environment so that the connector and brazing materials will be compatible with the system. If the threaded connections are selected, the same basic principles for the joint design are applied as for the larger static seal joints, except that bowing between fasteners is no longer a problem. The preload is applied to the gasket (or joint) by one, large, threaded section and loading can be more uniform than for the bolted joints. Probably, the first rule in selecting threaded connections would be to use the available proven connections wherever possible for the particular application. The design pressures, temperature range, and other parameters should be available from vendors (in the case of the purchased complete joints) or are controlled by specifications and standards for the standardized connection for most liquid systems. If none of these connections are suitable for a particular application, methods for design and analyses are presented in the available literature[35].

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(35) Rathbun, F. O., op cit.

Threaded connection designs utilizing any of the "pressure-energized" face gaskets, flat gaskets and crush washers, or diametral gaskets require the same design considerations as previously delineated in the discussion of the Static Seal Joints. Angular and radial misalignment between the sealing surfaces are the same as for the static seal joints; however, they can be somewhat more critical and present more serious assembly problems because the gasket cross-section is usually much smaller. This smaller gasket cross-section provides a much smaller deflection and distortion capability.

c. Problem

The brazed and welded connections present some problems of alignment and may require holding fixtures during the brazing or welding operation. The welded connections that do not provide full penetration at the joint interface can also present some serious problems of cracking after a number of pressure cycles because of the inherent notch (or crack) at the inside of the joint which will eventually progress through the weld. This is particularly a problem where bending can occur at the weld joint. Tooling must be available to make and hold the proper alignment during assembly. Tooling is also required for cutting the connections from the line when necessary to replace components. An associated problem with this cutting of a connection from the line is contamination of the system by metal chips or particles from the cutting operation. Careful selection of the brazing materials must be made for compatibility with the system fluids. In addition, the associated brazing temperatures can cause more problems. Heat-sinks or similar devices may be required in many instances to prevent over-heating component parts in both the welded and brazed connections. The brazed connections may also require dressing the tube for short lengths to specified diameter and roundness tolerances to provide proper fits between the tube outside diameter and the fitting inside diameter. The swaged connections present essentially the same problems of alignment and holding fixtures during assembly as well as cutting tools for removing the connection from the line.

The most difficult problem to solve with the welded, brazed, and swaged joints are with the transition (dissimilar metal) joints. Welding methods are available for transition materials but tooling is not yet available for proper connection of dissimilar tubing joints. Also, the transition joint weld is not particularly strong. These joints usually require a threaded connection for load carrying members as well as a seal weld. Brazing of the transition joint is made more difficult because the brazing materials are usually a compromise between the best selection for each of the tubing materials and the fitting. Swaging of the transition joint can be a problem because the swaging pressures required for the two materials may be different. Also, electrolytic corrosion and fluid compatibility of the completed brazed or swaged joints must be thoroughly investigated.

The proprietary threaded connections require that the fitting be welded, brazed, or swaged to the tubing and, in the cases where one half of the fitting cannot be machined on the component, will also require welding, brazing, or swaging to the component part. Most of these fittings also necessitate that exposed male threads be provided on the components which can present some serious handling problems. Extremely good alignment of the two halves of the connection is also required during assembly for most of these fittings. Replacement of a damaged or worn fitting necessitates that the fitting be cut from the tubing or component. The big advantage of these connections over the standard flared or flareless fittings is that each connection contains only one sealed joint and no rotation occurs between the two flanges as the connection is tightened.

The standard flared and flareless threaded connections are basically less reliable than any of the other joints because each connection contains two sealed joints. The tube-to-fitting connection of the flared tubing joint presents some serious concentricity problems between the fitting, the sleeve, and the coupling. This causes unequal loading at the sealing interface (tube flare-to-fitting). To gain sufficient sealing stresses (plastic deformation) in the tubing flare, excessive stresses (and associated deformation) is experienced in the sleeve and the B-nut. These excessive stresses cause hoop deformation in the sleeve and bending at the loading point between the B-nut and the sleeve. The thread lubricants selected can also create wide variations in the sealing stresses caused by variations in the friction coefficients. Equalization of the stresses and creep causes loss of torque in the connection. If the flare of the tube is not properly fabricated, bending cracks (crazing) can occur at the radius to the flare and result in complete failure of the joint. The premium grade "MC" standard flared tube connections are an attempt to reduce these problems by increasing the strength in the sleeve and the nut while providing a known (sprayed-on) lubricant. They also provide refined finishes on the flare and the fitting (sealing interface) to reduce the required sealing stresses. Reduced concentricity and runout tolerances help to lessen the required loading to obtain sufficient sealing stresses. Also, these connections need an improved (low tolerance) seamless tubing. These fittings can still present some loading problem because of eccentricities in the connection and the basic mechanics of transcribing torque-to-gasket seating stresses at the B-nut-to-fitting joint.

Tube-to-fitting connections with the flareless fittings contain a ferrule that "digs-into" the tube and can be a fairly good joint for some applications. However, this can cause problems regarding the reusability of the connection if proper handling and assembly procedures are not followed. The flareless connection contains the same tolerance and concentricity problems as the flared tube connections along with some of the same creep and stress equalization problems. The flareless connectors are somewhat more sensitive to combination temperature and vibration environments than the flared tube connections.

In the component (boss-to-fitting) flared and flareless connection, rotation between sealing surfaces during the installation of the fitting causes problems. The wiping action can cause galling between the seal and fitting (in the case of the metal gaskets) or some shaving in the case of the elastomeric and Teflon O-ring. Elastomeric O-ring gaskets present the same gas permeability and fluid compatibility problems as do the static sealed joints. Teflon O-rings also present cold flow problems because the gland volume is greater than the O-ring volume which means that the Teflon O-ring is not completely constrained. It is possible to use O-rings with oversized cross-sections to alleviate this problem, but then installation of the O-ring over the fitting threads without damaging the gasket is difficult. Replacing the AND10050 boss with the MS16142 boss will reduce the volume of the groove area and somewhat better service is then attained with O-ring boss gaskets of standard cross-section. Metal gaskets can cause deterioration of the fitting and the boss sealing surfaces after a number of assemblies. Sealing surface finishes on the military standard fitting bosses and associated parts are usually not fine enough for the metal gaskets. Improved surface finishes, concentricities, and tolerances of the "MC" standards reduce, but do not eliminate, the problem. Some metal gaskets also require over-stressing the military standard fittings to gain sufficient sealing stresses. Sealing of bulkhead fittings with metal gaskets is very difficult because the gasket must seal around the neck of the fitting. Extreme pressure lubricants must be used sparingly at the gasket-fitting-boss interfaces to prevent galling between the parts.

4. Joint Design

The best welded joints require full penetration of the weld at the joint interface. If this is not possible, the joint should be designed so that the loads are carried by shear in the weld. The welded joints that carry the line loads in bending can be useful at the lower pressures and limited pressure cycles. The current technological tooling available for these joints should be thoroughly investigated. It is strongly recommended that joint design concepts using the welded connections be thoroughly tested and evaluated with respect to the design loads, temperature, fluids, all environments, assembly and disassembly techniques before making the design final for actual hardware.

The requirement for the brazed joints should also be thoroughly investigated for compatibility with the system fluid, brazing temperatures and its affects upon structures and components, as well as the required machining of tubes and bosses. Pressure and load carrying capability is a function of the clearance between the fitting inside diameter and the boss or tubing outside diameter, brazing material strengths, and the length of the braze joint. Sufficient tubing, boss lengths, and space must be provided for the brazed connections that must be cut out of the system to allow for the desired number of disconnections and connections. The required tooling for making the connections and cutting or de-brazing the connections should be thoroughly investigated so that sufficient space can be provided around the

connection for tool accessibility. De-brazing temperatures are somewhat higher than the temperatures required during the brazing operation. Selection of cutting tools and procedures for preventing contaminants from entering the lines can be extremely important. The required additional boss length and space to make the desired number of connections can be reduced if the cutting tools cut close to the fitting and properly prepare the boss end for the new connection at the same time. The same requirements apply for the swaged connection as for the welded or brazed connections insofar as installation and disconnection tooling accessibility and space for the desired number of new connections is concerned. Line contamination during disconnection must also be considered. Pressure and load-carrying capability is a function of swaging pressure, length of the swage, as well as fitting and tubing or boss material strength.

Most of the proprietary connections are designed to withstand at least the burst pressures of the tubing to be used. However, all design loads and conditions that could be encountered cannot be considered in a standard connection. The design loads, temperature, environments, as well as other factors should be accurately determined and supplied to the manufacturer of the selected connection before the joint design is made final. The manufacturer should be capable of determining whether the selected connector is suitable for the application and make the necessary recommendations to assure good design. If the connection halves must be welded or brazed to the tubing or component during system assembly, tooling accessibility must also be considered. Usually, this is done as prefabricated assemblies so that welding, brazing or swaging to the mating parts is less difficult and more reliable. The preparation requirements for the mating tube or boss is presented on sheet 2 of Figure No. 84 for those connections where the information was readily-available. These requirements must be requested from the manufacturer to assure a good welded, brazed, or swaged joint to the connection.

If the standard flared or flareless tube connections are selected, all of the requirements for the fittings, tubing, and bosses and the associated specifications must be studied and analyzed for the particular application. Most of these are included on sheet 3 of Figure No. 84 for ready reference. It should be understood that the military standard connections were specifically designed for liquid systems. The more refined MS 24000 series flared fittings should be used for most non-critical gaseous systems and the NASA "MC" standard fittings are recommended for the critical gaseous or liquified gaseous systems. The standard flareless fittings can also be useful for some applications. However, it is strongly recommended that any of the standard fittings be thoroughly tested and evaluated for the particular pressure, line loads, temperature, vibration and other environments before the final selection is made. It may be necessary or desirable to use some of the interchangeable coupling nuts, gaskets, and fittings to assure a reliable joint at all of the expected conditions. Also, the desired leakage rates and load-carrying capabilities at all operating conditions may not be attainable with the standard connections.

Considerable thought must be given to the assembly tightening torques, particularly at the component bosses, during the initial design when the threaded connections are used. Sufficient torsional strength must be provided in the boss to resist assembly tightening loads, or wrench flats must be provided so that excessive twisting loads are not transmitted to the boss.

If it is desirable to design the complete threaded connection for a particular application, the design considerations are essentially the same as for the static sealed joint except that the fastener (threaded section) is somewhat different. A very complete method of design analysis for new designs (or for selected designs) of threaded connections is available in the existing literature[36].

5. Handling Procedures and Problems

After the connectors have been selected and all of the necessary design and machining requirements are specified, the next problem is one of properly handling the component parts of the connection up to the time that they are assembled. Handling of component parts after fabrication presents many possibilities for damage to critical fits, sealing surfaces and threads, particularly if these surfaces are exposed. It is necessary to inspect, clean, store, and assemble parts after fabrication; therefore, handling cannot be completely eliminated. The susceptibility of the connection to damage depends upon the type connection, the materials involved, and how critical the required fits and surfaces are. The welded, brazed, and swaged connections are probably least affected by damage of the component parts. However, even these cannot provide a good joint if the tubing or the fitting becomes out-of-round, tube ends are excessively damaged, or excessively deep scratches are inflicted upon the tube. Threaded connections are subject to the same problems concerning damage to the gaskets or sealing surfaces as the static sealed joints but have the additional problem of damage to the integrated, threaded male flange.

Preventing damage to the prepared tubing for the welded, brazed, or swaged connections is usually expeditiously resolved by providing appropriate plugs and covers for the tube ends during all handling and cleaning operations. The fittings can be cleaned and properly packaged at the supplier's plant if source inspection is provided. The gaskets and fittings for the threaded connections can also be properly cleaned and packaged at the supplier's plant when there is provision for source inspection.

6. Assembly Procedures and Problems

The final problem in obtaining a good small tubing connection is the assembly of the component parts. This involves not only proper handling, but good techniques of assembly to assure proper mating at the connection interfaces.

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(36) ibid.

If cleaning and appropriate protective packaging is provided, the interface surfaces will be free of any foreign particles that could prevent proper sealing or imperfect welding, brazing, or swaging of the connection. If cleaning is not previously required, it will be necessary to assure that all interface surfaces are free of any foreign particles. Any solvents used for this operation must be compatible with the system fluids and only lint-free tissues or cloths can be used. The hand-cleaning process may be much more difficult to apply to the small threaded connectors than with the larger, static sealed joints.

The threaded connections require proper lubrication of the threads. Insufficient lubrication will reduce the applied sealing stresses on the gasket. Excessive lubrication can present serious system contamination problems, particularly after a number of assemblies. This is expecially a problem with the flared and flareless fitting-to-boss joint. The lubricant begins to "stack-up" at the bottom of the boss as more and more of it is applied after each assembly. The best method of eliminating this condition is to allow the lubricant to be applied only to the male thread of the connection.

Lubricants should not be used on the gaskets or interface sealing area of the proprietary threaded connections. These are treated the same as any static sealed joint. The design should provide sufficient gasket deformation at the sealing interface to preclude the necessity of using lubricants and greases. However, there is one type of joint where lubricant must be used on the gasket; this is the flared and flareless boss-to-fitting joint. The relative rotation of the two mating surfaces, as the connection is tightened thereby loading the gasket, requires that lubricants be used for all but the Teflon O-rings or the Teflon-coated metallic gaskets. The elastomeric O-rings require a very light lubrication to prevent tearing or damaging the gasket. The uncoated metallic gaskets need a lubricant to prevent galling at the sealing interfaces. This lubricant must be capable of withstanding high compression stresses without breaking down. The lubricant must also be compatible with the system fluids because this area is directly exposed to the system fluids. Evaluation tests should be conducted with the selected lubricant to assure that it will prevent galling between the gasket and mating surfaces.

As indicated, alignment of the component parts of the connectors can be critical in obtaining a good joint. If this has the potentiality for becoming a serious problem, it must be considered during the initial design stages of the system. The threaded connections are too small to use alignment pins, but the use of alignment shoulders and pilot diameters are possible. These alignment devices should be designed so that the necessary alignment is accomplished before contact is made with the gaskets of the threaded connections.

Methods used for tightening the threaded connections must also assure that sufficient load is applied to the gasket without over-stressing the components parts of the connection. Two wrenches are always necessary in

assembling the threaded connection to prevent excessive twisting of the weaker sections of the flanges and the tube as well as to prevent any relative rotation between the gasket and mating surfaces. Testing must be conducted with the selected threaded connection and lubricants to assure that sufficient minimum sealing stresses are applied to the gasket using the minimum specified torque value. The maximum allowable torque value must not over-stress any of the components parts.

B. TEST PROCEDURES

Except for some preliminary proof and leak tests to determine the best assembly procedures and requirements, the laboratory evaluation test procedures were based upon a predetermined statistical testing plan. The details of each of the planned tests and the over-all statistical plan is presented in the following descussion.

1. Critical Dimension Inspection

A complete first article inspection of all the components of the connection was performed before starting the tests. All of the actual critical dimensions, surface finishes, and other pertinent information were recorded. The equipment and/or methods used to perform these inspections was devised so that no detrimental marks, scratches, or other imperfections resulted from the inspection.

2. Assembly Procedures

The assembly procedures and requirements were those recommended by the supplier of the connection (or in accordance with the specifications of the standard connections) if these proved satisfactory for the preliminary proof and leak tests. Where no particular assembly procedure was recommended, the procedure was initially evolved in conjunction with some preliminary proof and leakage tests. Also, improvements in the recommended procedures were gained from the preliminary tests. These preliminary proof and leakage tests were essentially the same as those specified in the ensuing discussion.

After the necessary alignment, torque values, lubrication, and other requirements were determined for the particular connection, they were specified as part of the standard assembly procedure for the connection. In general, the assembly procedures includes: thorough cleaning of all the components of the connection before assembly; visually examining and recording any nicks, dents, scratches, or tool marks in the critical surface areas; recording the lubricant used; run-up and final torque values, or (in the case of brazed or swaged connections) tooling used for brazing or swaging. The connection part number and serial number, as well as other information pertinent to the particular connection was also recorded.

3. Statistical Testing Sequence

The actual testing plan sequence for each size of each connection for the complete program was as shown on Table XI.

4. Proof Test

Hydrostatic pressure was slowly applied up to the specified proof pressure (usually 3,000 psig) and held for 10 min. Any leakage was determined by visual observation (for the leakage location) and decay check for the leakage rate). The test specimen was then disassembled and inspected for any detrimental distortion of the component parts. The presence of any detrimental distortion was considered to be a joint failure. All of the acceptable joints were thoroughly cleaned, dehydrated, and reassembled for the next test.

5. Leak Test

a. Water Displacement

Leakage tests were conducted by submerging the test specimen in water and slowly pressurizing with the test gas (usually helium) to 75% of the proof pressure, which was held for three minutes. Leakage checks were made at 10%, 25% and 50%, 75% and 100% of the leakage test pressure. Any leakage was collected in an inverted graduated beaker and the amount recorded in cubic-centimeters-per minute. If no bubbles were observed, the joint was considered satisfactory for the mass spectrometer tests. If bubbles were observed, the threaded connections were re-tightened in accordance with the procedures previously determined and again leak tested. The starting and final tightening torque value was recorded. If leakage persisted, the connection was disassembled (or sectioned, in the case of the swage and brazed joints) to determine the cause.

b. Mass Spectrometer Leak Test

If the connector successfully passed the water displacement leakage test, the joint was encapsulated or installed in a vacuum chamber so that the actual leakage rate could be determined with the mass spectrometer. Leakage checks were again made at 10%, 25%, 50%, 75% and 100% of the leakage test pressure with gaseous helium. The leakage rate value was recorded in standard cubic-centimeters-per-second.

6. Pressure Cycling Test

Pressure cycling tests were conducted with the test specimen in accordance with the procedure shown on Table XII. The test medium was usually gaseous nitrogen at ambient temperature. A total of 30,000 pressure cycles were applied with leakage tests conducted at the 50th, 100th, 200th, 500th, 1,000th, and every 1,000th cycle thereafter, up to and including the 30,000th

TABLE XI

TESTING PLAN SEQUENCE

TEST NO.	JOINT SPECIMEN NUMBERS						
	1, 2, 3	4, 5, 6	7, 8, 9	10, 11, 12	13, 14, 15	16, 17, 18	19, 20, 21
1	PL	PL	PL	PL	PL	PL	PL
2	B	PC	V	RA	LT	HT	S
3		V	LT	PC	S	RA	HT
4		RA	PC	HT	V	S	LT
5		LT	S	V	HT	PC	RA
6		HT	RA	S	PC	LT	V
7		S	HT	LT	RA	V	PC
8		B	B	B	B	B	B

SYMBOL	TESTS
B	Burst
HT	High Thermal Shock (-260°F to + 1200°F)
LT	Low Thermal Shock (-423°F to + 160°F)
PC	Pressure Cycling
PL	Proof and Leak
RA	Repeated Assembly
S	Storage
V	Vibration

TABLE XII

PRESSURE CYCLE TEST PROCEDURE

TEST NO.	SPECIMEN NO.	TEMPERATURE, °F	PRESSURE
1	4, 5, 6	Ambient	S. L.
2			High
3			S. L.
4			Low
5	4, 5, 6		S. L.
6	10, 11, 12		S. L.
7			Low
8			S. L.
9			High
10	10, 11, 12		S. L.
11	7, 8, 9		S. L.
12			High
13			S. L.
14			Low
15	7, 8, 9		S. L.
16	16, 17, 18		S. L.
17			Low
18			S. L.
19			High
20	16, 17, 18		S. L.
21	13, 14, 15		S. L.
22			High
23			S. L.
24			Low
25	13, 14, 15		S. L.
26	19, 20, 21		S. L.
27			Low
28			S. L.
29			High
30	19, 20, 21		S. L.

S. L. = Sea-Level Pressure

cycle. The maximum time to pressurize the specimen was one second. If gross leakage developed in the connection during this test, the specimen was disassembled (or sectioned) to determine its cause.

7. Thermal Shock Tests

a. Low Thermal Shock Test

The connectors tested were either submerged in or flowed with liquid nitrogen or liquid hydrogen until the joint temperatures stabilized. They were then submerged in hot water or flowed with hot gaseous nitrogen (+160°F) until the joint temperature stabilized. The joint was then re-tested with gaseous helium in accordance with the leak test procedure described in paragraph IV, B, 5. The low thermal shock testing sequence was in accordance with the procedure shown on Table XIII.

b. High Thermal Shock Test

No high thermal shock tests were conducted prior to program termination. The planned testing consisted of soaking the connectors at -160°F (in liquid nitrogen vapor or in an appropriately controlled environmental chamber) until the temperatures became stabilized. Hot gases (1200°F) would then be flowed over the connector until the connector temperature became stabilized. Leakage tests were then to be conducted at both low and high temperatures using gaseous helium in accordance with the procedure described in paragraph IV, B, 5. The planned high thermal shock testing sequence is shown on Table XIV.

8. Repeated Assembly Tests (Threaded Connection)

Repeated assembly tests of 20 cycles maximum or until failure occurred were conducted with each test specimen. The connector was visually examined for nicks, dents, scratches, or distortion after each assembly cycle. Leakage tests in accordance with the procedure described in paragraph IV, B, 5 were conducted after the first, fifth, tenth, and twentieth assembly.

9. Vibration

Specimens were vibrated in each of two mutually-perpendicular planes as follows:

a. Sweep frequency checks were first conducted to determine the resonant frequency points.

b. Test specimens were vibrated at each resonant frequency for one hour at 20 g's.

c. Leakage tests were conducted after each resonant frequency vibration test.

TABLE XIII

LOW THERMAL SHOCK TEST PROCEDURE

TEST NO.	SPECIMEN NO.	TEMPERATURE, °F	PRESSURE	DATA TO BE RECORDED
1	13, 14, 15	Ambient	0-3000	Lubricant, torque, temperature, pressure, and leakage
2		High		Temperature, pressure, and leakage
3		Ambient		
4		Low		
5	13, 14, 15	Ambient		
6	7, 8, 9	Ambient		
7		Low		
8		Ambient		
9		High		
10	7, 8, 9	Ambient		
11	19, 20, 21	Ambient		
12		High		
13		Ambient		
14		Low		
15	19, 20, 21	Ambient		
16	4, 5, 6	Ambient		
17		Low		
18		Ambient		
19		High		
20	4, 5, 6	Ambient		
21	16, 17, 18	Ambient		
22		High		
23		Ambient		
24		Low		
25	16, 17, 18	Ambient		
26	10, 11, 12	Ambient		
27		Low		
28		Ambient		
29		High		Temperature, pressure, and leakage
30	10, 11, 12	Ambient		Torque, temperature, pressure, and leakage

* -423°F to +160°F, For One Size Only - 25 Cycles per Specimen

TABLE XIV

PROPOSED HIGH THERMAL SHOCK TEST PROCEDURE

TEST NO.	SPECIMEN NO.	TEMPERATURE, °F	PRESSURE	DATA TO BE RECORDED
1	16, 17, 18	Ambient	0-3000	Lubricant, Torque, Temperature, Pressure, and Leakage
2		High	0-1500	Temperature, Pressure, and Leakage
3		Ambient	0-3000	
4		Low		
5	16, 17, 18	Ambient		
6	19, 20, 21	Ambient		
7		Low		
8		Ambient	0-3000	
9		High	0-1500	
10	19, 20, 21	Ambient	0-3000	
11	10, 11, 12	Ambient	0-3000	
12		High	0-1500	
13		Ambient	0-3000	
14		Low		
15	10, 11, 12	Ambient		
16	13, 14, 15	Ambient		
17		Low		
18		Ambient	0-3000	
19		High	0-1500	
20	13, 14, 15	Ambient	0-3000	
21	4, 5, 6	Ambient	0-3000	
22		High	0-1500	
23		Ambient	0-3000	
24		Low		
25	4, 5, 6	Ambient		
26	7, 8, 9	Ambient		
27		Low		
28		Ambient	0-3000	
29		High	0-1500	Temperature, Pressure, and Leakage
30	7, 8, 9	Ambient	0-3000	Torque, Temperature, Pressure, and Leakage

* +1200°F to -160°F, For One Size Only - 25 Cycles per Specimen

10. Salt Spray Test

Salt spray tests were conducted with the specimens in accordance with MIL-E-5272C, Paragraph 4.6, except that the spray was a 20% salt solution and the time period was at least five hours.

11. Burst Test

Hydrostatic burst tests were conducted with water or hydraulic fluid. Pressure was slowly applied to the specimen until rupture occurred. The pressure at which the failure or rupture occurred as well as the location of the rupture were recorded.

C. TEST RESULTS

Testing in accordance with the statistical testing plan was not completed with all of the connections tested prior to the termination of the program. However, the testing that was completed with various connections is presented herein.

1. Critical Dimensional Inspection Results

Most of the proprietary connections inspected were within the supplier's blueprint and specification requirements. Packaging of these connections was well-planned for good protection of the connection component critical dimensions and surfaces during shipping and handling. However, this was not the case with the standard flared tubing components. Some lots of the flared tubing hardware had extremely high rejection rates because of dimensional discrepancies as well as surface and thread defects. One lot of the MS precision fittings had apparently been reworked by the supplier from AN fittings by resurfacing the nose of the fitting. In the process, the resulting intersection of the nose surface and the fitting bore was a sharp corner; additionally, none of the other surfaces and dimensions were within the MS precision fitting tolerances. Many of the standard fittings and other components were received without protective packaging with groups of ten to fifty parts placed in a single brown paper bag. Only some of the standard parts were received properly protected and individually packaged as would be expected with critical parts. Gaskets for the various connections presented the same packaging and handling problems as were described in the static seals discussion. Some manufacturers consistently packaged the gaskets in a proper manner, while the packaging of other suppliers was consistently poor. Some manufacturers were inconsistent in that they provided good packaging for one lot of parts and poor packaging for the next one. Excluding shipment damage, the rejection rate for dimensional and other defects was very small for the gaskets, which were consistently within the supplier's blueprint and specification requirements. The acceptable parts were properly packaged and stored for future use in the evaluation program.

2. Assembly Procedures and Requirements

Assembly procedures and requirements ranged from reasonably simple and foolproof (particularly for one brazed connection and the swaged

connection) to extremely difficult and meticulous for some fitting-to-boss gaskets. The tightening torque values for the standard and proprietary threaded connections and gaskets are shown on Table XV. The assembly procedures and requirements are detailed in the following discussion.

The exothermic brazing system (Pyrobraze) was the simplest and easiest system of all the connections to assemble. The vendor suggests that the tubing (or tubular bosses) be sized to a diameter and roundness tolerance of $\pm$ 0.001 in. The sized diameters are at the maximum allowable standard diameters for the particular tube size. The assembly tooling required consists of a six-volt dry cell battery. The brazing system requires a flux (which is provided with the fitting) and will present contamination problems for some systems because the flux "runs-out" on the inside as well as the outside of the tube. Also, the fitting must be cut from the system if leakage is encountered or if component replacement is necessary. The exothermic brazing system is currently limited to CRES tubing (or boss) connections. This connection should be very useful in solving low level leakage problems in fuel or gas systems that can be easily flushed (or where the flux contamination will not be a problem) after the system is assembled.

The "SpaceCraft" connection is a fluxless induction brazing system that uses an argon gas purge on both the inside and outside of the connection. This connection process is very clean and produces a lightweight joint. Assembly procedures and requirements are somewhat difficult, but tools and equipment are available for all of the steps in the procedure. The tubing must be sized to a diameter and roundness tolerance of $^{+.003}_{-.000}$ in. The sized diameters are near the minimum allowable standard diameters for the particular sized tubing. A tool, which contains a vacuum system to carry away any chips or other particles, is provided for dressing the tube end. The induction brazing tool is a hand-tool which is attached to relatively long flexible electrical and gas purge lines. Hand-tools are also provided to cut the fittings and to de-braze for disconnection. The tube (or tube and boss) ends can then be redressed for the new connection. The system is currently limited to CRES tubing (or boss) connections.

The swaged "H" fitting is also a relatively simple connection to make. Tube sizing tolerances are less stringent ($^{+.004}_{-.002}$ in.) than the brazed connection. The hand swaging tool is somewhat bulky but easy to handle. Fittings are available for use with both CRES and aluminum tubing. This system may also have possibilities for making dissimilar metal (CRES-to-aluminum) joints, although this was not discussed with the supplier. Cleanliness of the system is the responsibility of the user. Also, alignment of the two ends during assembly is somewhat critical.

Two methods were used to attach the two halves of the proprietary threaded connections to the tubing; the <u>socket weld</u> method and the <u>roll swaging</u> method. The socket weld method is standard for the Conoseal and Roylyn tube connections and is available for the Gamah connections on special order. In the Gamah connections tested, the tubing was roll swaged into the

TABLE XV

TIGHTENING TORQUE VALUES (IN.-LB) FOR THREADED CONNECTIONS

STD. FLARED TUBE COUPLING			
SIZE (In.)	TUBE MAT'L	MIN. TORQUE	MAX. TORQUE
1/4	ALUM.	40	65
	CRES	135	150
3/8	ALUM.	75	125
	CRES	200	300
1/2	ALUM.	150	250
	CRES	450	500
5/8	ALUM.	200	350
	CRES	650	700
3/4	ALUM.	300	500
	CRES	900	1000
1	ALUM.	500	700
	CRES	1200	1400

STD. FITTING TO BOSS			
SIZE	FITTING MAT'L	MIN. TORQUE	MAX. TORQUE
1/4"	ALUM.	50	
	CRES	75	
3/8"	ALUM.	70	
	CRES	105	
1/2"	ALUM.	100	
	CRES	150	
5/8"	ALUM.	135	
	CRES	200	
3/4"	ALUM.	165	
	CRES	245	
1"	ALUM.	270	
	CRES	405	

CONOSEAL TUBE UNIONS			
SIZE	GASKET MAT'L	MIN. TORQUE	MAX. TORQUE
1/4"	ALUM.		
	CRES	160	240
3/8"	ALUM.		
	CRES	270	405
1/2"	ALUM.		
	CRES	400	600
5/8"	ALUM.		
	CRES	550	825
3/4"	ALUM.		
	CRES	700	1050
1"	ALUM.		
	CRES	1100	1650

ROYLYN TUBE UNIONS			
SIZE (In.)	GASKET MAT'L	MIN. TORQUE	MAX. TORQUE
1/4			
3/8	17-4PH	100	140
1/2			
5/8			
3/4	17-4PH	400	500
1			

NATORQ BOSS GASKETS			
SIZE	GASKET MAT'L	MIN. TORQUE	MAX. TORQUE
1/4"	ALUM.	240	300
	CRES*	300	360
3/8"	ALUM.	300	360
	CRES*	360	420
1/2"	ALUM.	480	600
	CRES*	600	720
5/8"	ALUM.	720	840
	CRES*	960	1080
3/4"	ALUM.	1200	1320
	CRES*	1380	1500
1"	ALUM.	1320	1440
	CRES*	1440	1560

DEL SHELL FLARED TUBE GASKETS			
SIZE	GASKET MAT'L	MIN. TORQUE	MAX. TORQUE
1/4"	COPPER	40	65
	NICKEL	135	150
3/8"	COPPER	75	125
	NICKEL	270	300
1/2"	COPPER	150	250
	NICKEL	450	500
5/8"	COPPER	200	350
	NICKEL	650	700
3/4"	COPPER	300	500
	NICKEL	900	1000
1"	COPPER	500	700
	NICKEL	1200	1400

* INCLUDES INCONEL X.

VOISHAN "SCOTT" BOSS SEALS (FOR FITTING-TO-BOSS JOINTS)												
TUBE SIZE (In.)	ALL ALUM. PARTS EXCEPT 6061-T6 ALUM. SEAL TORQUE		ALL ALUM. 6061-T6 PARTS ALUM. SEAL TORQUE		ALL CRES & STEEL PARTS COPPER SEAL TORQUE		ALL CRES & STEEL PARTS NICKEL SEAL TORQUE		ALUM. COUPLING NUTS OR BOSS WITH CRES FITTING TIN PLATED ALUM. SEAL TORQUE		CRES COUPLING NUTS OR BOSS WITH ALUM. FITTING TIN PLATED COPPER SEAL TORQUE	
	MIN.	MAX.	MIN.	MAX.	MIN.	MAX.	MIN.	MAX.	MIN.	MAX.	MIN.	MAX.
1/4	50	80	90	150	190	210	210	240	87	150	170	190
3/8	95	160	160	225	375	410	425	475	165	225	325	375
1/2	190	310	375	500	625	690	725	800	375	500	550	625
5/8	250	480	550	700	875	975	1000	1100	550	690	800	875
3/4	375	625	810	1000	1250	1375	1400	1575	800	1000	1100	1250
1	625	875	1125	1375	1625	1950	1875	2200	1125	1375	1500	1750
VOISHAN "CONICAL" SEALS (FOR TUBING-TO-FITTING AND FITTING-TO-BOSS JOINTS)												
1/4	40	65	70	120	150	165	165	190	70	120	135	150
3/8	75	125	150	180	300	330	335	375	130	180	270	300
1/2	150	250	300	400	500	550	575	625	300	400	450	500
5/8	200	350	450	550	710	770	810	875	450	550	650	700
3/4	300	500	650	800	990	1100	1125	1250	650	800	900	1000
1	500	700	900	1100	1300	1550	1500	1750	900	1100	1200	1400

connection halves which is the standard configuration for the Gamah. The socket weld attachment is not the most desirable weld joint because of the inherent notch in the weld. This is particularly true where the connection is to be used in areas of extreme vibration or high pressure impulse cycling. However, no particular problems were encountered with these attachments at the conditions imposed during the laboratory tests. The Conoseal and Roylyn standard connections are available in CRES materials only; therefore, only these connections were tested. The Gamah connection is available in both the aluminum alloy and CRES materials; both types were tested. The Conoseal is also available on special order for aluminum or aluminum-to-CRES connections, but none were tested. It was necessary to use heat-sinks during welding to prevent excessive warpage at the critical sealing surfaces in both of the socket welded connections. Good alignment between the connection halves was essential during assembly and tightening of all three of the proprietary connections tested. It was also important to thoroughly clean the critical sealing areas and the threads before assembly. Only the male threads were lightly lubricated before the assembly. Tightening torque values were those specified by the supplier (see Table XV) unless these were found to be inadequate during the initial testing.

Assembly procedures and requirements for the standard flared and flareless tube connections are specified in the appropriate military assembly procedures and are presented on Table XV. The procedures and requirements were followed for all assemblies of the standard threaded connections, except where replacement (and supposedly improved) coupling, fitting, or seals were used and over-tightening of the connection was either permissible or required by the supplier to effect a seal. For example, the Wiggins "DL" coupling assembly allowed the joint to be tightened as much as 150% of the standard torque values without overstressing the coupling nut or sleeve. Some compressive distortion of the nose (conical surface) of flared tube fittings was possible at these values; therefore, the maximum permissible torque values were lowered for these fittings depending upon the size of the connection. It was found during the initial stages of the program that is was necessary to lightly lubricate the male threads (only) on the flared fitting and also the back side of the sleeve to prevent galling as well as to provide sufficient axial loading to effect a seal.

A potentially serious assembly problem was encountered with the flared tube-to-fitting (Del shell and Voishan conical) gaskets. In some cases, the gasket would remain in the B-nut half of the connection at disassembly and would be inadvertently left in that position when a new assembly (with a new gasket) was made. In some of the cases, the old gasket would cock as the new assembly was made and partially, or completely block the port. However, the major problem was the reduction in clamping force to hold the tube in place. In one instance, the tubing flare pulled out of the coupling when pressure was applied. The same problem would apply to the conical seal boss connections, although this never happended during testing of the joint. A precautionary note was included in the assembly procedure to assure that the technician checked for this condition before reassembling the joint.

Almost all of the proprietary fitting-to-boss gaskets required special procedures and/or tightening torques to effect a seal. The Natorq seals required a light coating of high compression lubricant to prevent galling between the gasket and the mating surfaces. The Natorq union and bulkhead seals tested required such high tightening torque values that the stresses in the fitting exceeded the material yield strength. Two sizes of the MS24392 unions were used to obtain data concerning permanent set (elongation) of the union at various torque values from 50% to 200% of the maximum specified by the supplier. The results of these tests are shown on Figure No. 85. Both the straight union and the bulkhead gasket, as tested, are permanently attached to the fitting after the initial assembly. No method could be devised to remove (cut) the gasket from the fitting without damaging the mating surfaces; therefore, the fitting and gasket were discarded and new assemblies were made when leakage occurred. The bulkhead union gasket was extremely difficult to assemble and to effect a seal with. The procedures developed for this gasket require that the fitting be held stationary with respect to the boss as the nut is tightened to the specified torque valve. The bulkhead nut must be backed-off and retightened to gain initial sealing. In many instances, the nut had to be retightened as much as three or four times before a good joint could be made. Usually, the straight union seal could be sealed satisfactorily during the first assembly, but occasionally these gaskets would also require reseating.

The Omniseal boss gaskets and the oversized Teflon O-ring gaskets both required a special assembly tool to fit the gaskets over the thread on the fitting. This tool consisted of a thimble-shaped Teflon device with a thin tubular section that fits over the thread on the fitting. The gasket would then slide up the tapered section of the tool and over the thread without damage. However, the gaskets were still very difficult to assembly onto the fitting unless they were heated (in warm water above 100°F) before sliding them over the thimble. The purpose of the oversized cross-section O-ring gasket was to sufficiently increase the volume to be completely constrained in the gland provided by the AND10050 boss with the standard straight union and bulkhead union assemblies. In this way, the cold flow problem with the Teflon would be reduced. However, the problem with this approach was that the face of the fitting would not always bottom on the face of the boss at standard torque values during the initial assembly. The Teflon would cold-flow into the gland after a few minutes to the extent that the joint could be retightened and brought face-to-face, but retightening of some joints was again necessary to assure that the standard torque values were maintained.

3. Proof Test

No failures were encountered with any of the connections at proof test pressures. Leakage did occur in some cases and usually, these joints would not pass the lakage test. Leakage of the joints tested at proof pressure is shown on Figure No. 86.

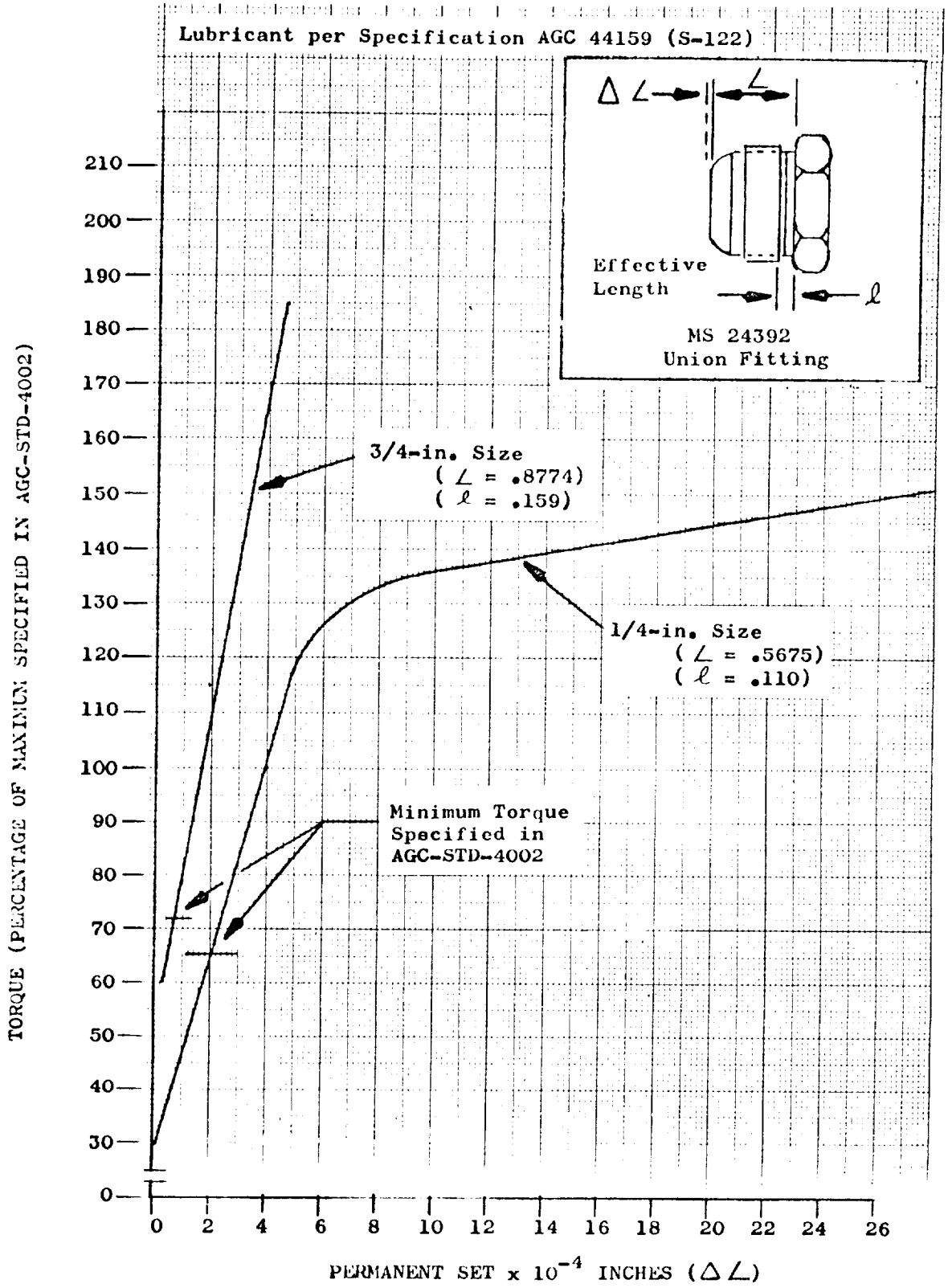


Figure 85
Torque vs. Permanent Set, MS24392 (CRES) Unions in AND10050 (CRES) Bosses with Natorq (CRES) Gaskets

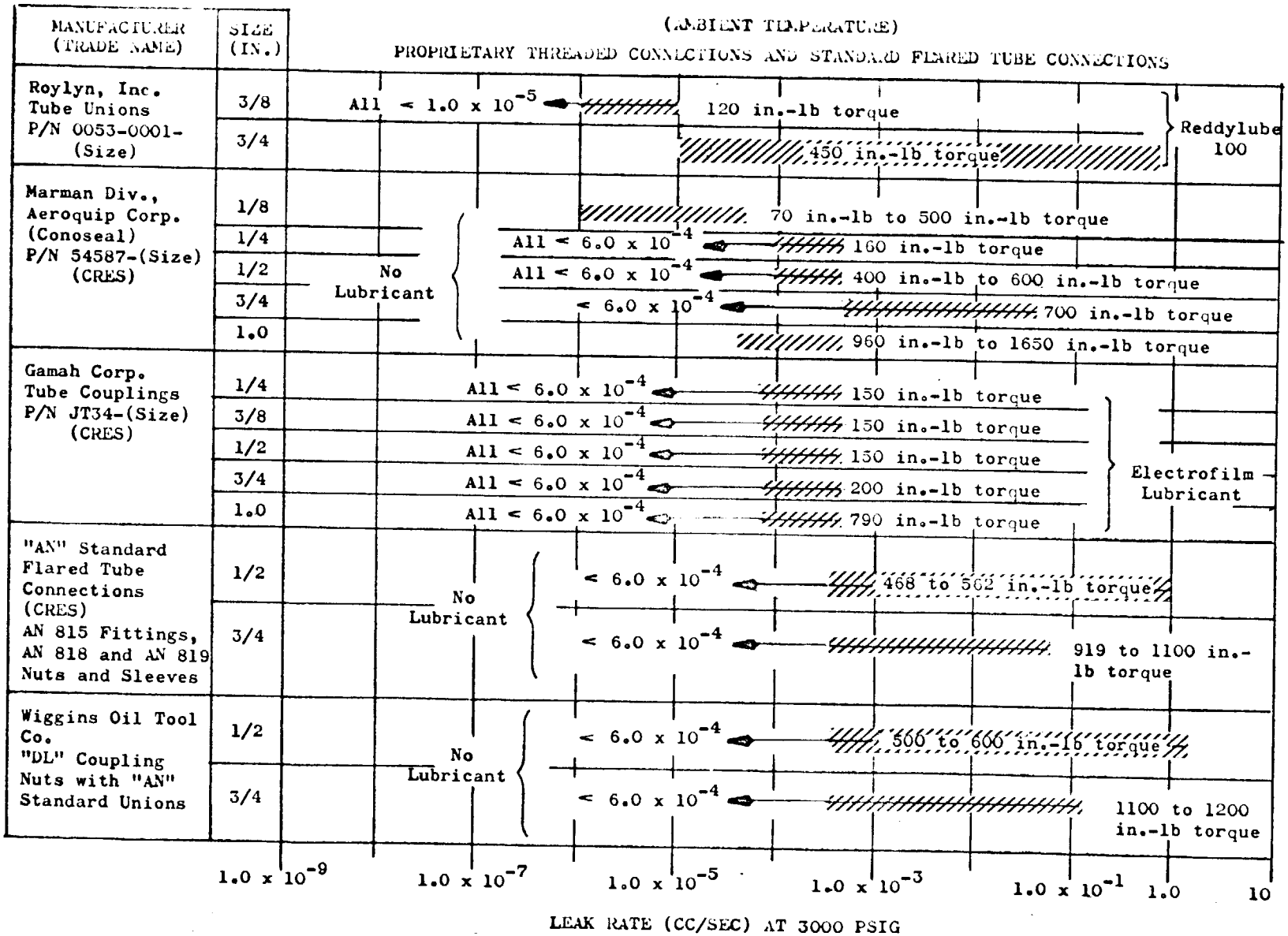


Figure 86
(Sheet 1 of 2)
Proof Test Results

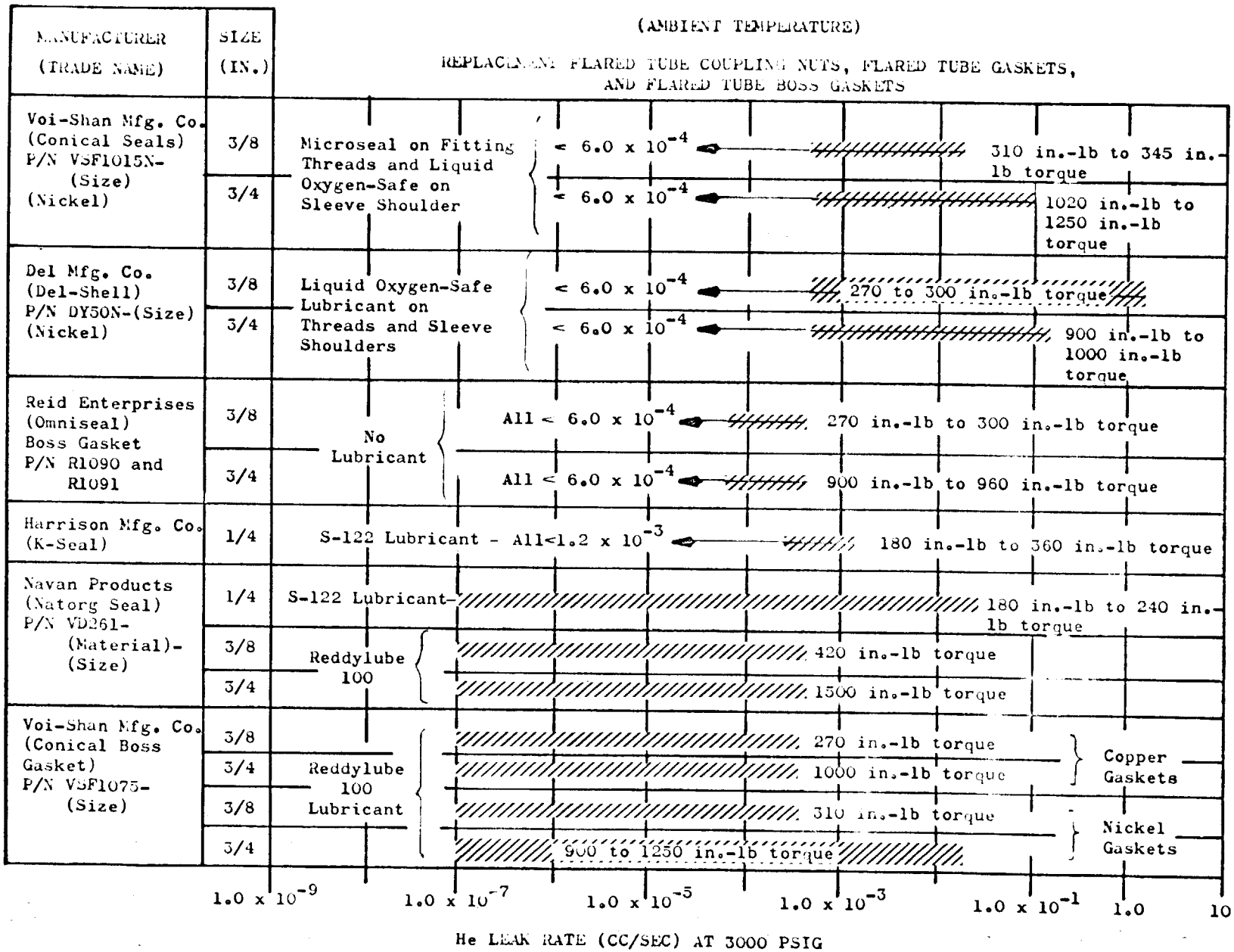


Figure 86
(Sheet 2 of 2)
Proof Test Results

No detrimental permanent distortions were noted on any of the brazed or swaged connections tested at proof pressures. A lessening in the torque of the threaded connections was noted. This was a particular problem with the standard flared tube connections. The torque loss in the threaded connections after the proof test is shown on Figure No. 87. It was not determined what part of the torque loss was caused by the creep characteristics of the connection materials. It was assumed that the greater part of the lessening in torque was caused by the equalization or relaxation of stresses plus the stresses induced by proof pressures in the connection. The Wiggins "DL" replacement coupling nut did not completely resolve the torque relaxation problem in the standard flared tubing connections, at least not at the standard torque values. Torque values above the standard values reduced leakage (subsequently discussed), but some torque relaxation remained evident.

4. Leakage Tests

The results of the initial leakage tests for all of the connections tested are shown on Figure No. 88. The leakage capabilities of the two brazed connections were estimated based upon some liquid leakage tests and sectioning of the connections through voids in the braze detected by radiographic inspection of the connections. All voids were extremely small, closed cells with no continuity between the voids or across the braze section of the joint. A more detailed discussion of the work done with these connections is available in the exisiting literature(37). Leakage rates of the swaged fittings were determined by helium leakage tests. A sealed chamber was constructed around the joint so that the mass spectrometer could be used to determine the leakage from the joint.

The leakage rates shown for the balance of the connections are usually calculated values based upon the length of time of the water submersion leakage test. In these cases, the lower limit and actual range of leakage rate values are not shown because they were not determined with the mass spectrometer. Where mass spectrometer tests were conducted, the lower limit and range is shown on Figure No. 88.

Comparisons were made with the standard flared tube connections and some of the various replacement parts (i.e., fittings and couplings) as well as the gaskets that can be used with these connections. In these instances, comparisons are made using essentially the same hardware as the standard connection, except for using the replacement parts. These comparisons are shown on Figures No. 89 through No. 93. The prime purpose of these comparisons was to determine whether the replacement parts or gaskets actually resolved the problem for which they were designed. Also, data were collected regarding some of these connections concerning leakage at various torque values, including overtorque (torque above the standard values). These data are shown on Figures No. 94 and No. 95.

(37) Hunt, V., Induction Processed Separable Tubular Brazed Connectors, NASA Report No. CR-515, July 1966

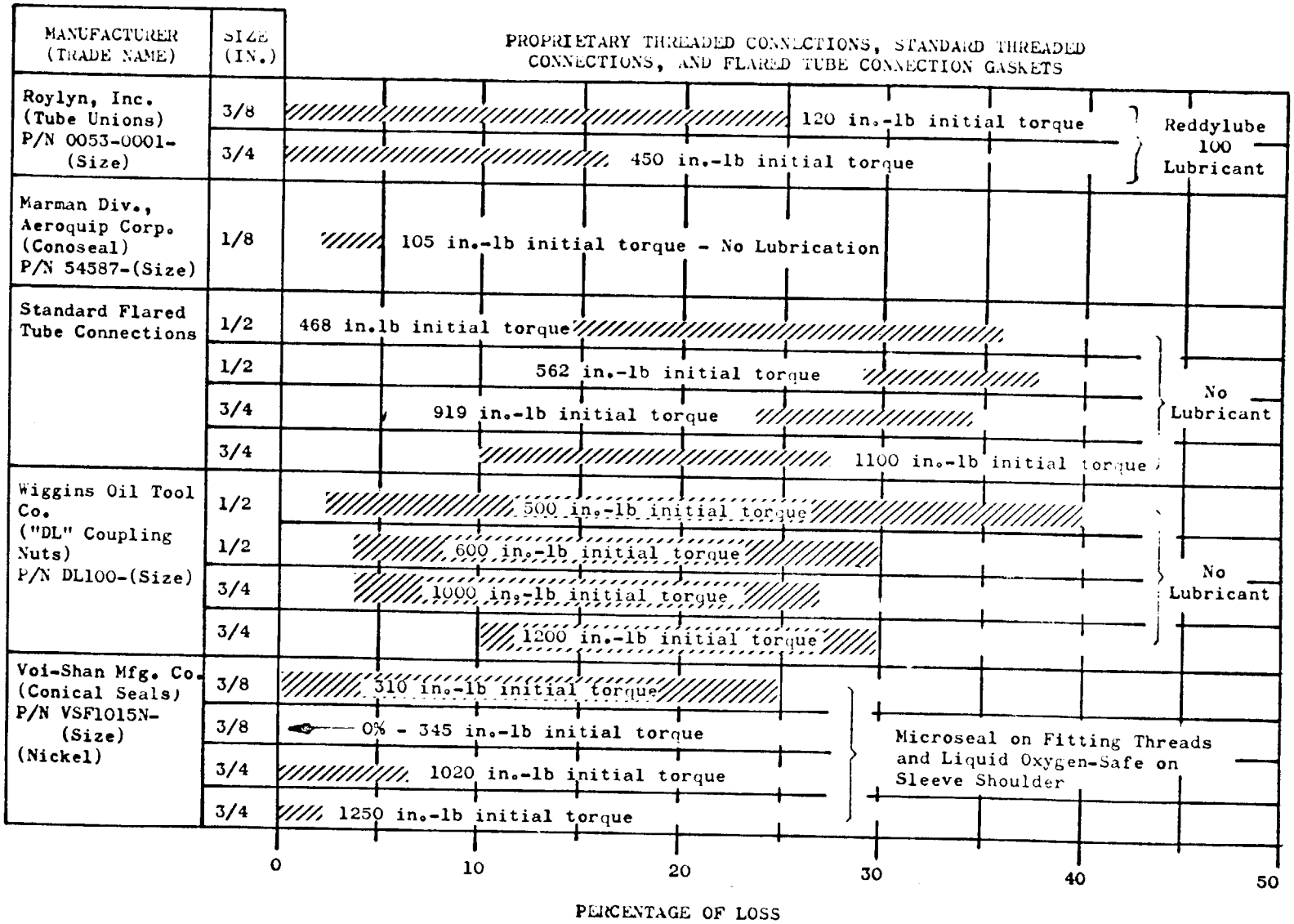


Figure 87
(Sheet 1 of 2)
Torque Loss in Threaded Connections at Proof Pressure

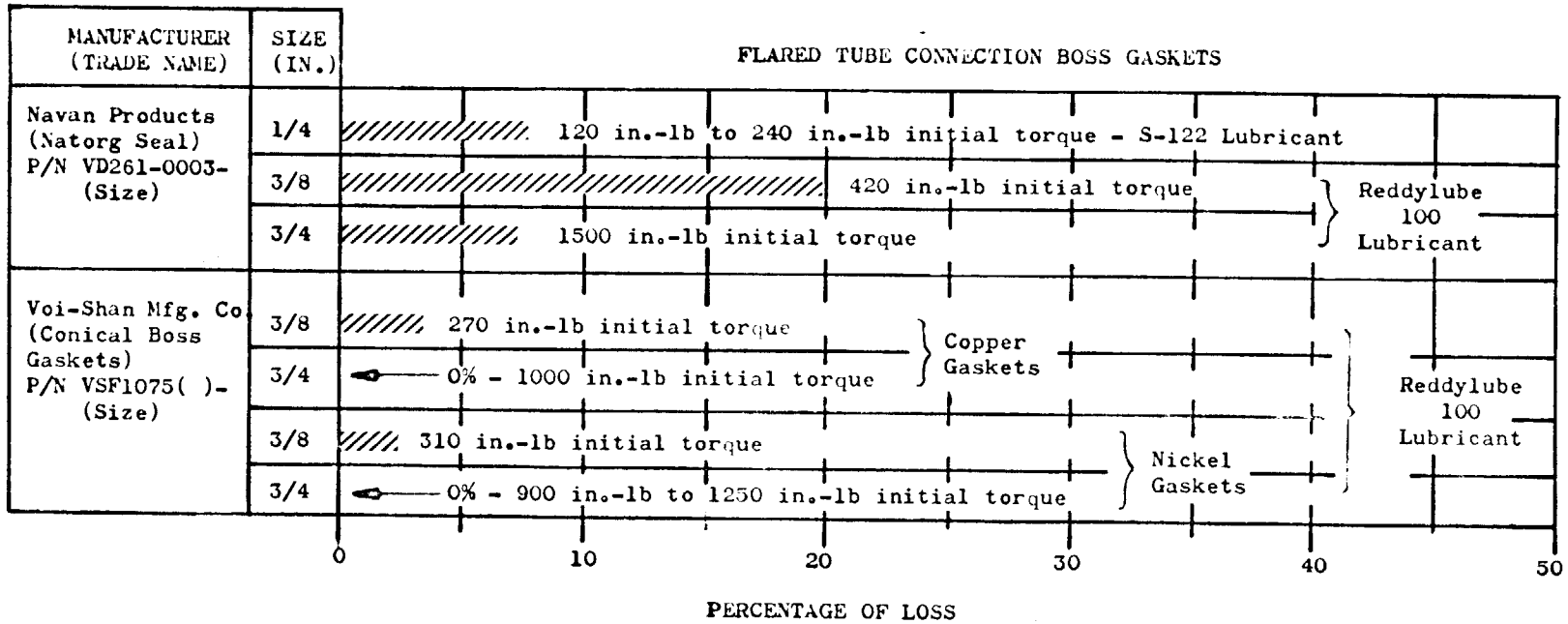


Figure 87
(Sheet 2 of 2)

Torque Loss in Threaded Connections at Proof Pressure

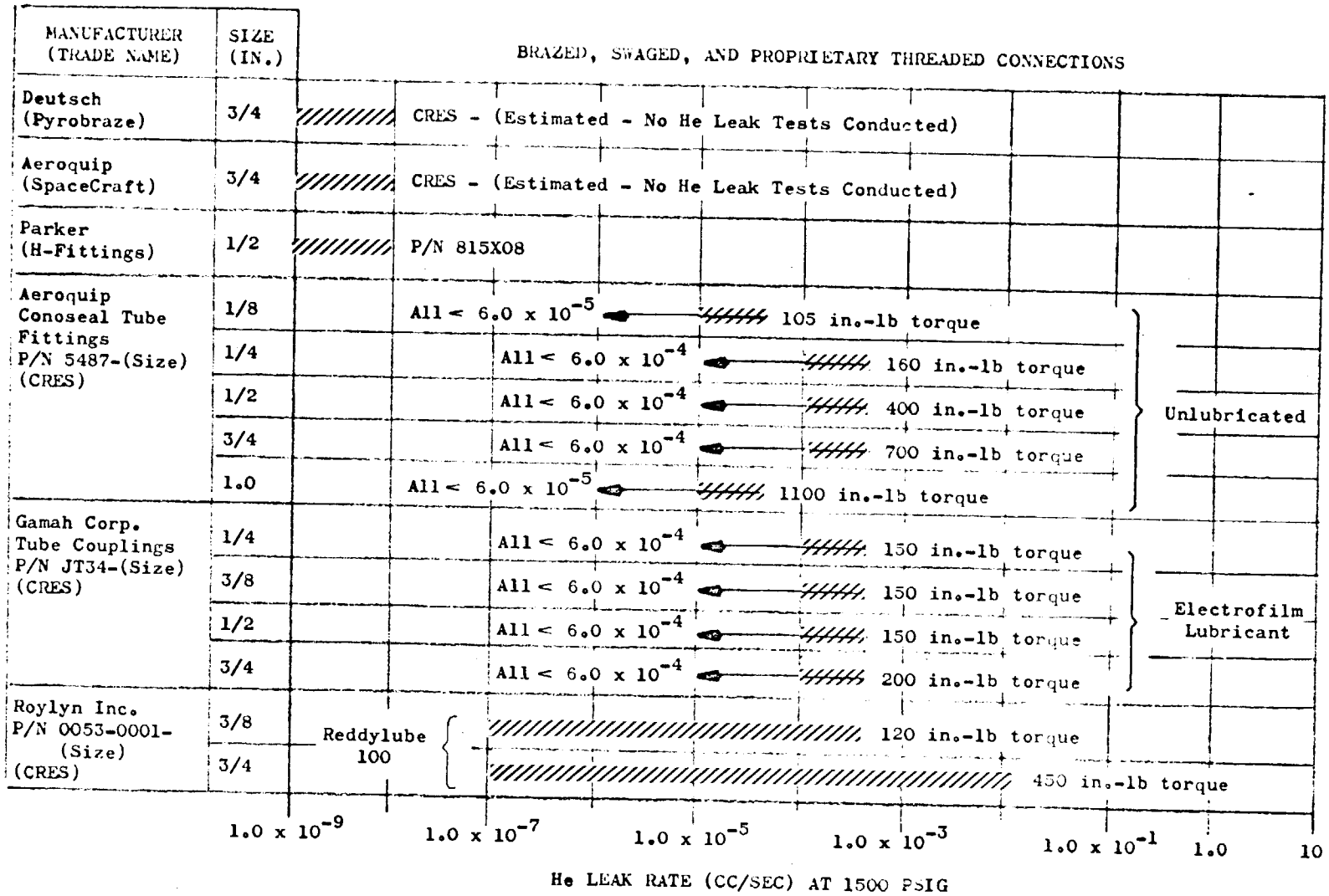


Figure 88
Initial Leak Test Results (Ambient Temperatures)
Sheet 1 - Brazed, Swaged, and Proprietary Threaded Connections

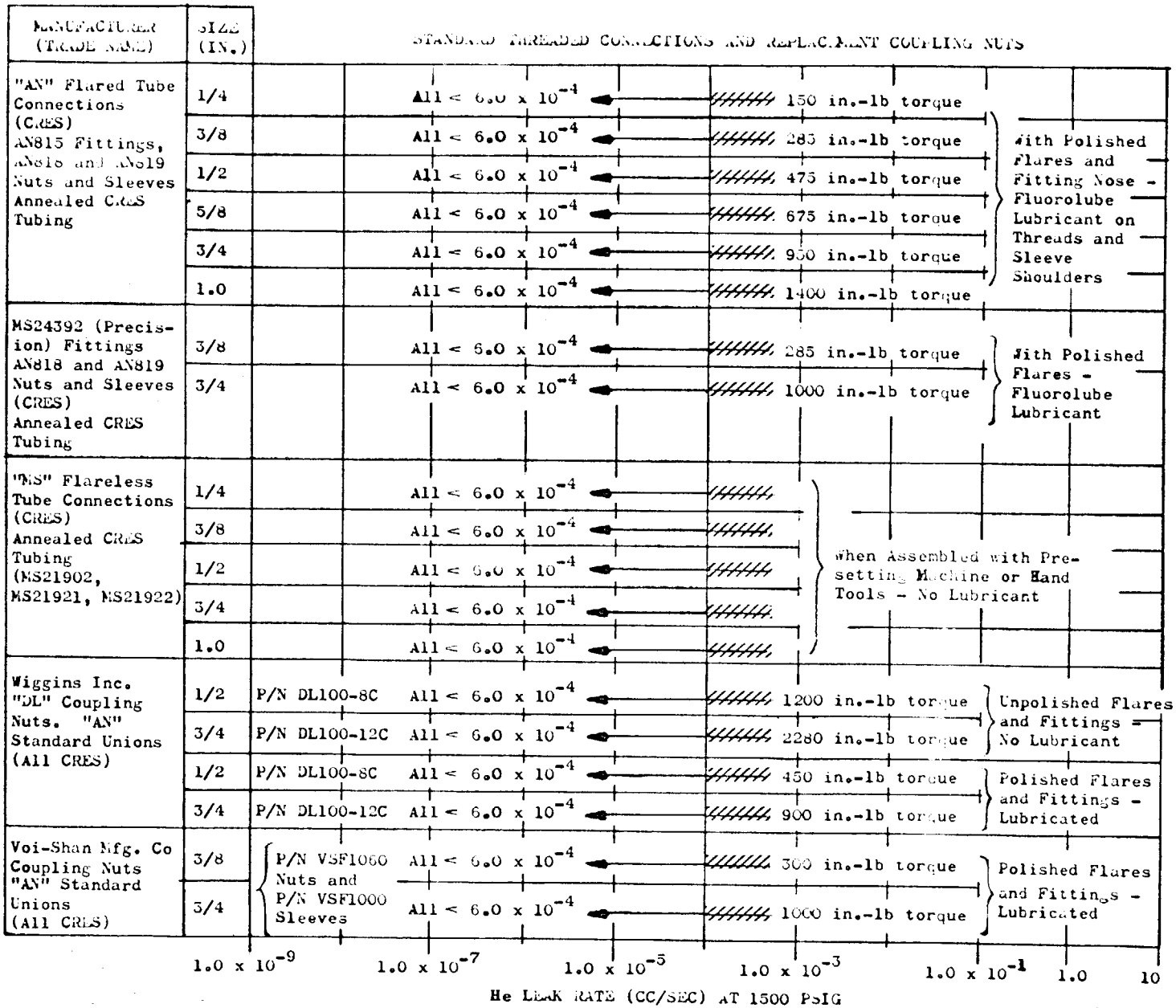


Figure 88

Initial Leak Test Results (Ambient Temperatures)

Sheet 2 - Standard Threaded Connections and Replacement Coupling Nuts

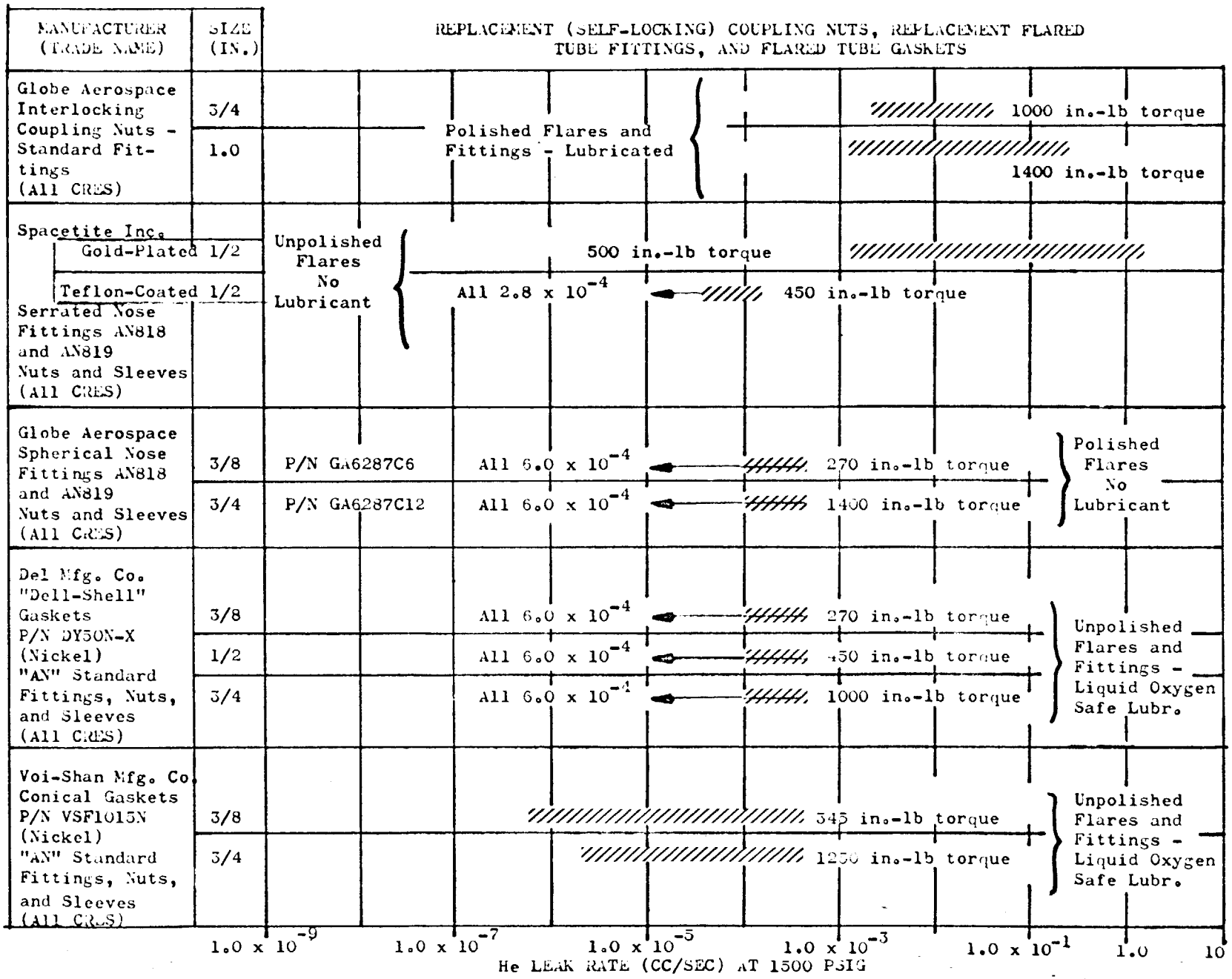


Figure 88

Initial Leak Test Results (Ambient Temperatures)

Sheet 3 - Replacement (Self-Locking) Coupling Nuts, Replacement Flared Tube Fittings, and Flared Tube Gaskets

Figure 88

Initial Leak Test Results (Ambient Temperatures)

Sheet 4 - Standard Flared Tube Connection Boss Gaskets

STANDARD FLARED TUBE CONNECTION BOSS GASKETS

MANUFACTURER (TRADE NAME)	SIZE (IN.)		
Teflon O-Rings (TFE) (Unlubricated) "AN" Bulkhead Fitting and AND10050 Bosses (All CRES)	1/4	Std. C/S-P/N AS1119J1-4	All < 6.0×10^{-4} — 135 in.-lb torque
	1/4	Oversize C/S (46% by Volume)	All < 6.0×10^{-4} — 135 in.-lb torque
	3/8	Std. C/S-P/N AS1119J1-6	All < 6.0×10^{-4} — 270 in.-lb torque
	3/8	Std. C/S (Glass-Filled)-P/N 2118-906	All < 6.0×10^{-4} — 270 in.-lb torque
	3/8	Oversize C/S (-55% by Volume)	All < 6.0×10^{-4} — 270 in.-lb torque
	1/2	Oversize C/S (24% by Volume)	All < 6.0×10^{-4} — 450 in.-lb torque
"AN" Straight Union Fittings and AN10050 Bosses (All CRES)	1/4	Oversize C/S (46% by Volume)	All < 6.0×10^{-4} — 135 in.-lb torque
	3/8	Oversize C/S (55% by Volume)	All < 6.0×10^{-4} — 270 in.-lb torque
	1/2	Oversize C/S (24% by Volume)	All < 6.0×10^{-4} — 450 in.-lb torque
Reid Enterprises (Omniseal) (Unlubricated) Polished Fittings (All CRES)	3/8	Bulkhead Fitting Gaskets-P/N R1090	< 6.0×10^{-4} — 300 in.-lb torque
	3/8	Straight Union Fitting Gaskets-P/N R1091	All < 6.0×10^{-4} — 270 in.-lb torque
	3/4		All < 6.0×10^{-4} — 900 in.-lb torque
Harrison Mfg. Co. (K-Seal) "AN" Fittings and AND10050 Bosses (All CRES)	1/4	P/N MC252C4T	All < 6.0×10^{-4} — 180 in.-lb torque - S-122 Lubricant on Threads Only

1.0×10^{-9} 1.0×10^{-7} 1.0×10^{-5} 1.0×10^{-3} 1.0×10^{-1} 1.0 10

He LEAK RATE (CC/SEC) AT 1500 PSIG

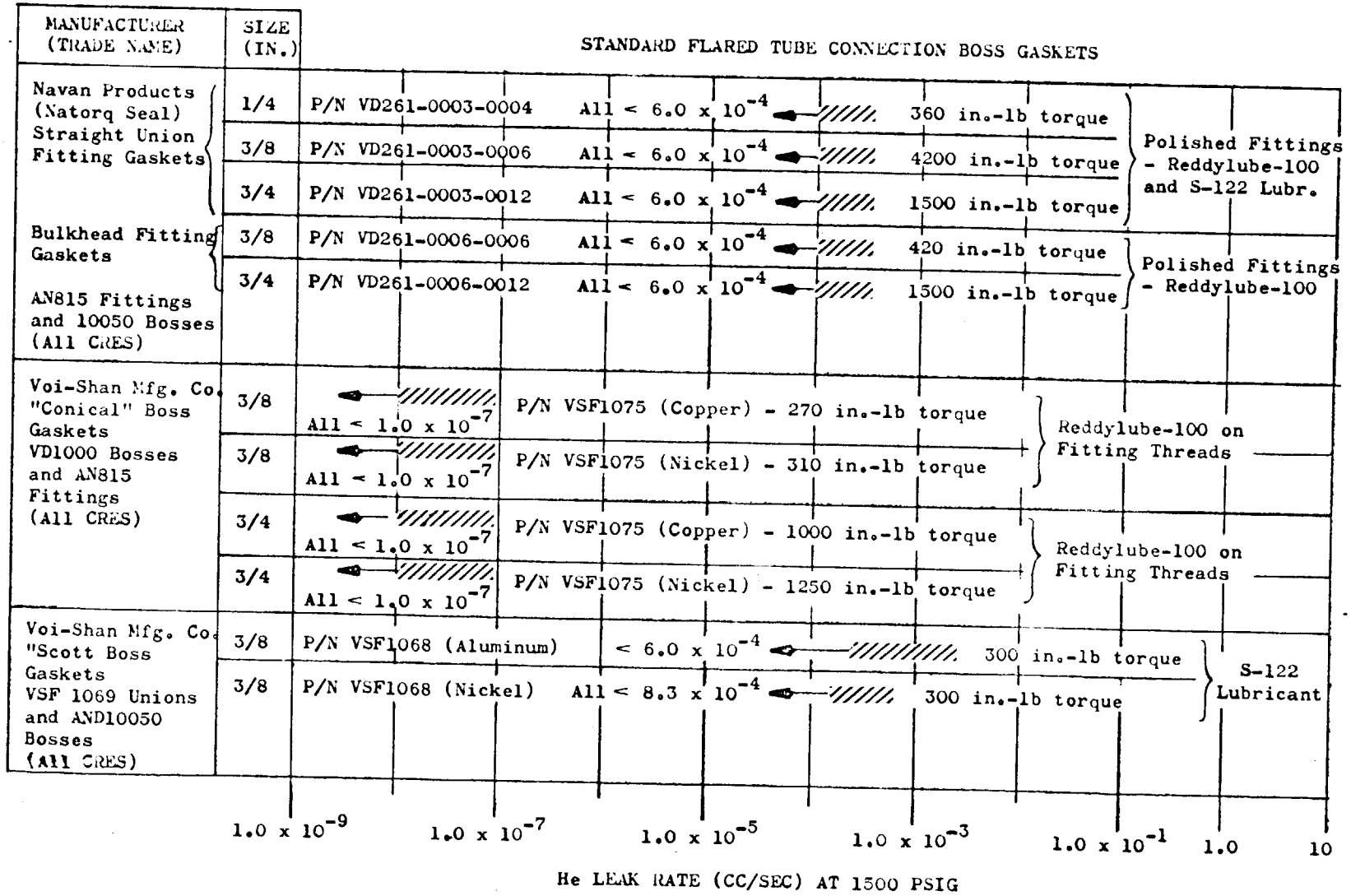


Figure 88
Initial Leak Test Results (Ambient Temperatures)
Sheet 5 - Standard Flared Tube Connection and Boss Gaskets

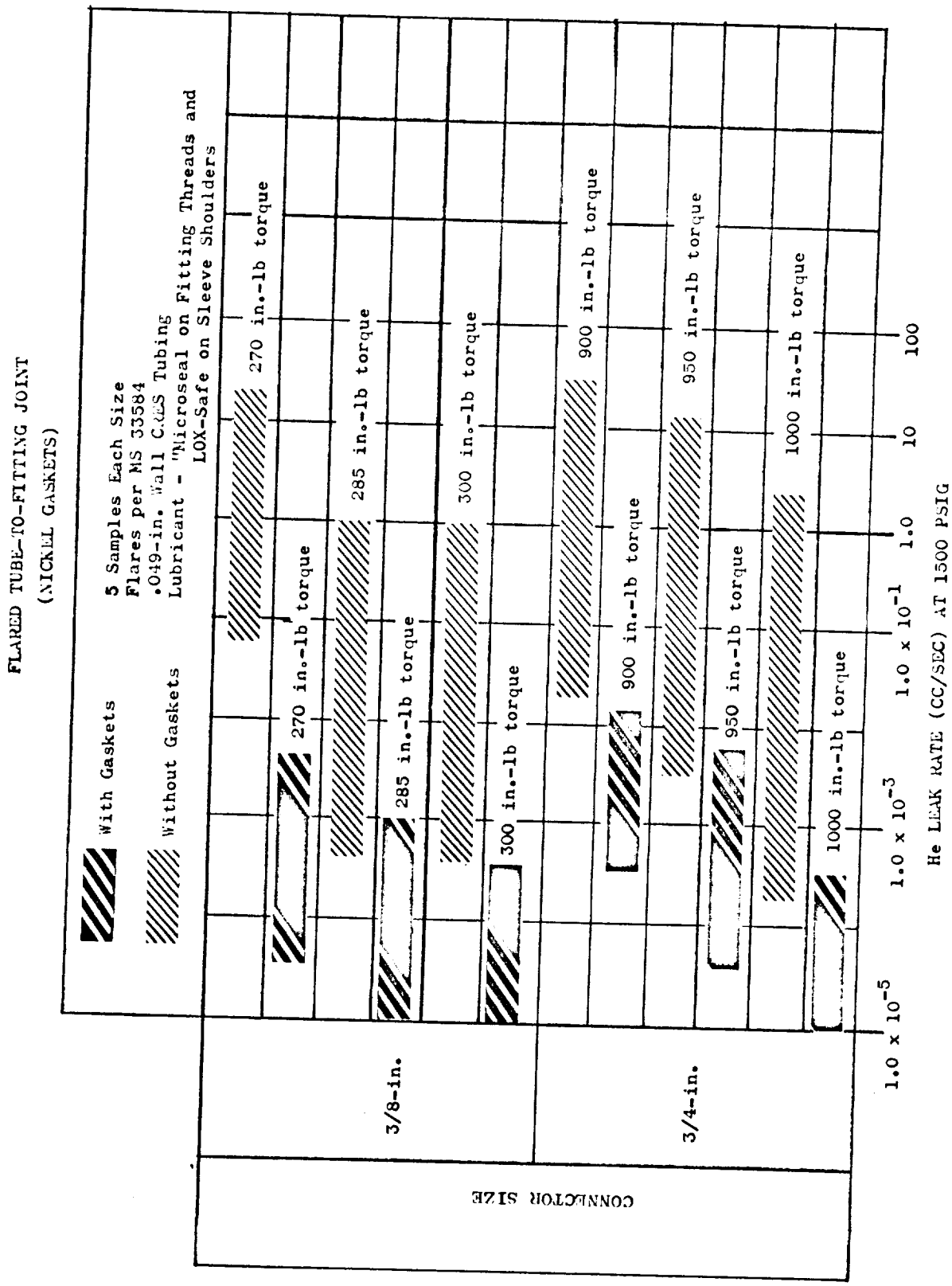


Figure 89
Leak Rate Comparisons, Voishan Conical
Gaskets and "AN" Standard B-Nuts

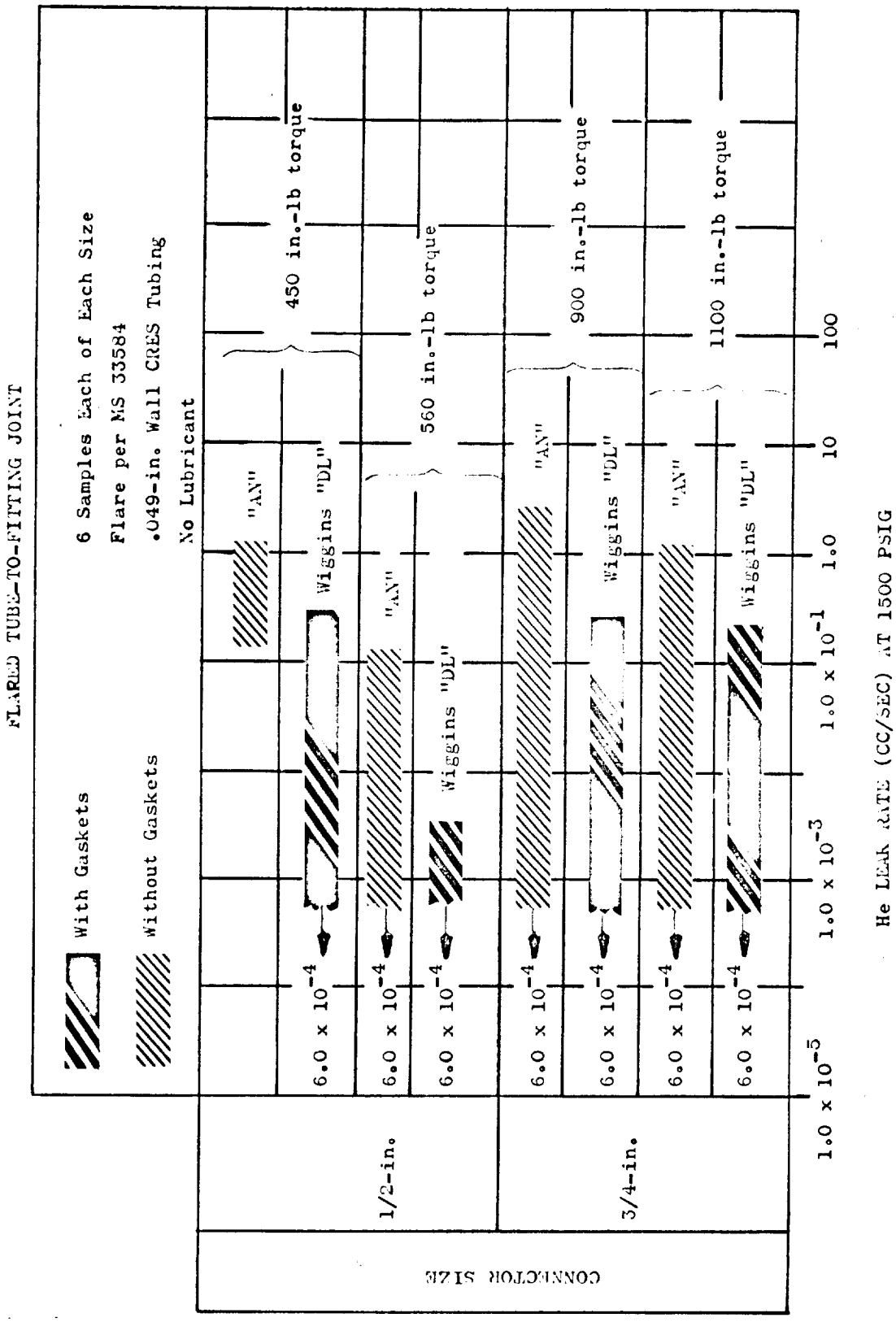


Figure 90
Leak Rate Comparisons, Wiggins "DL" Connector Nuts and Standard B-Nuts

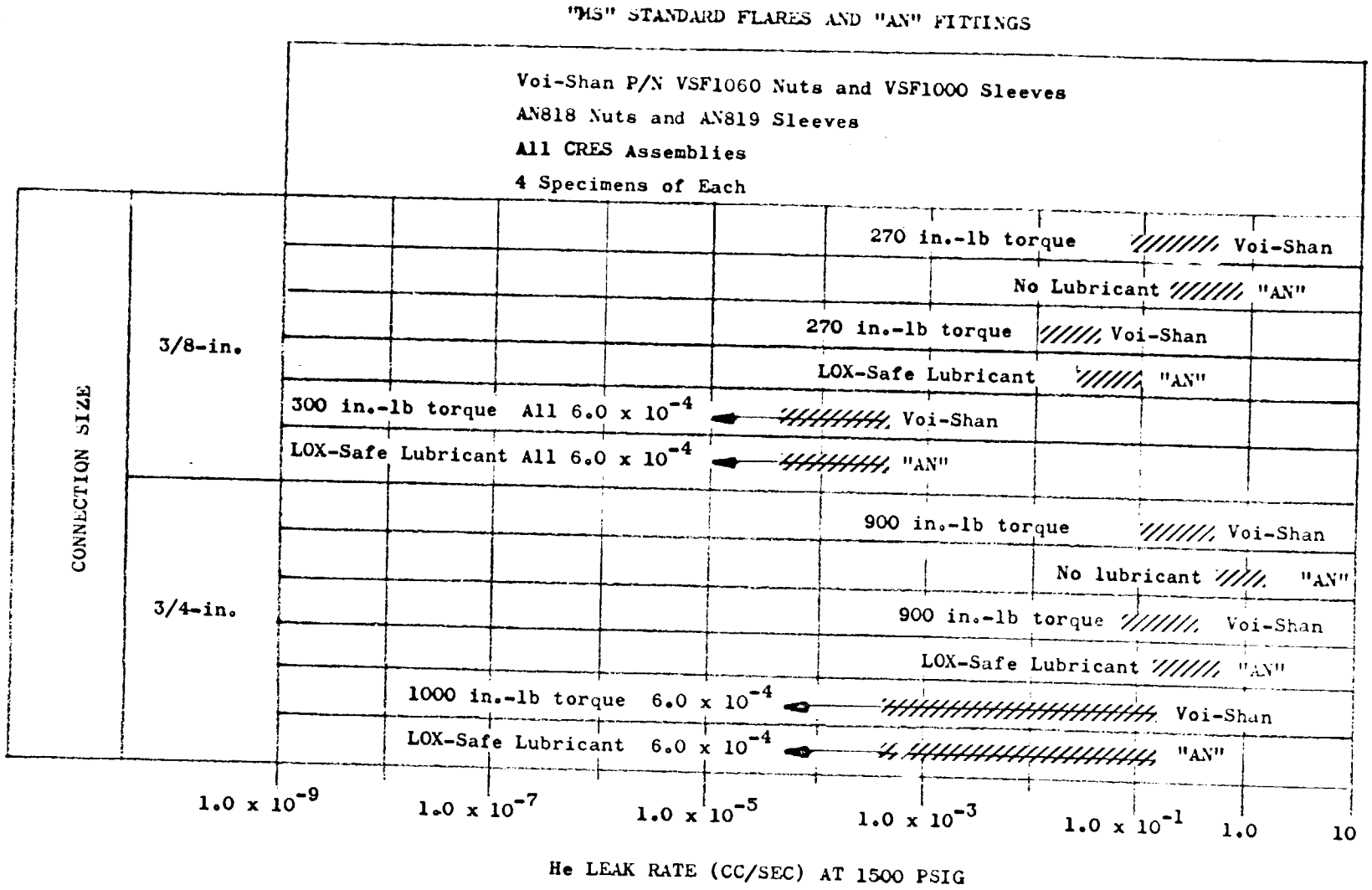


Figure 91
Leak Rate Comparisons, Replacement Voishan Lightweight Nuts and "AN" Standard B-Nuts

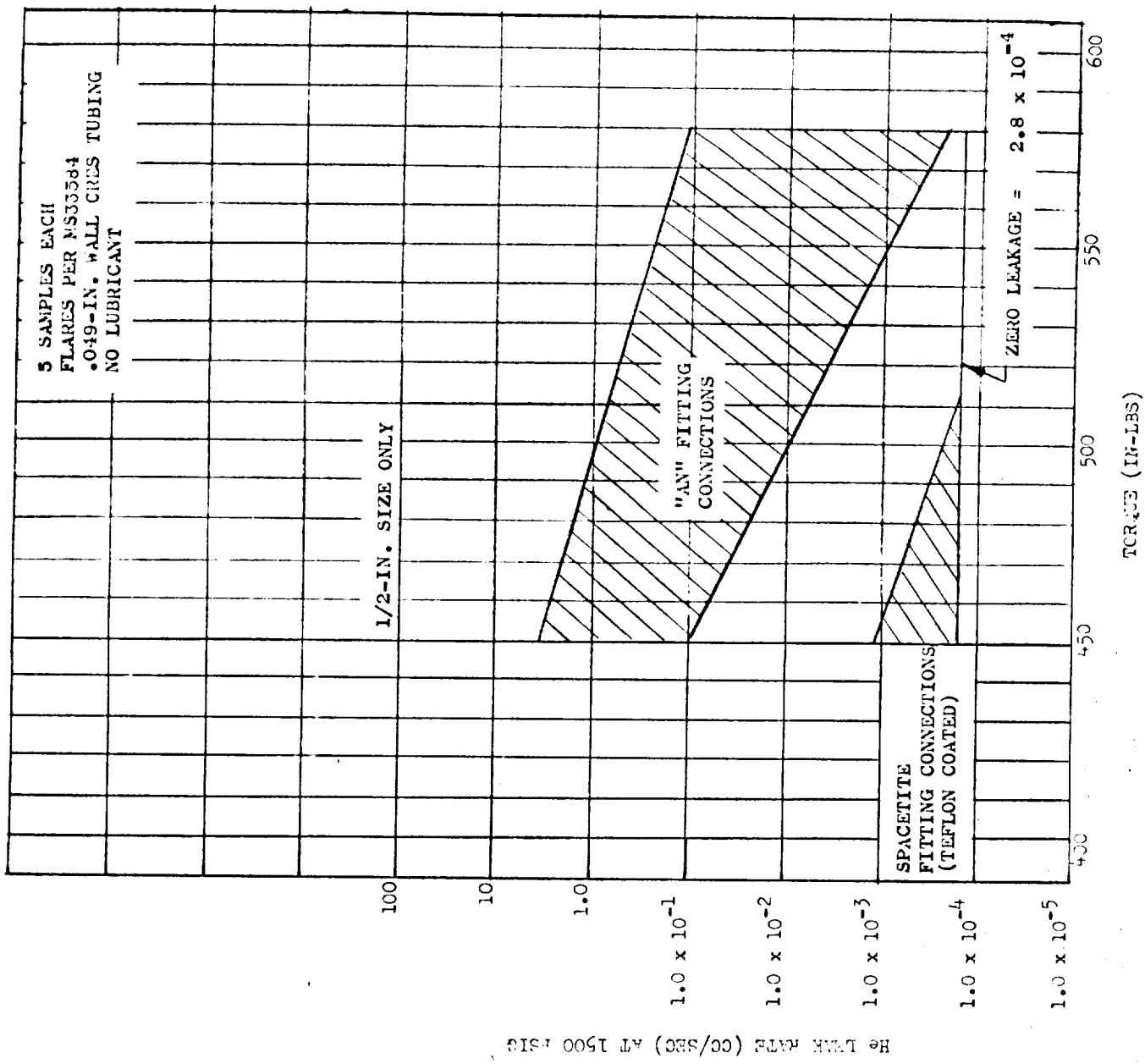


Figure 92
Leak Rate Comparisons, Spacetite Serrated Nose Fittings and "AN" Standard Fittings, Leak Rate vs. Torque

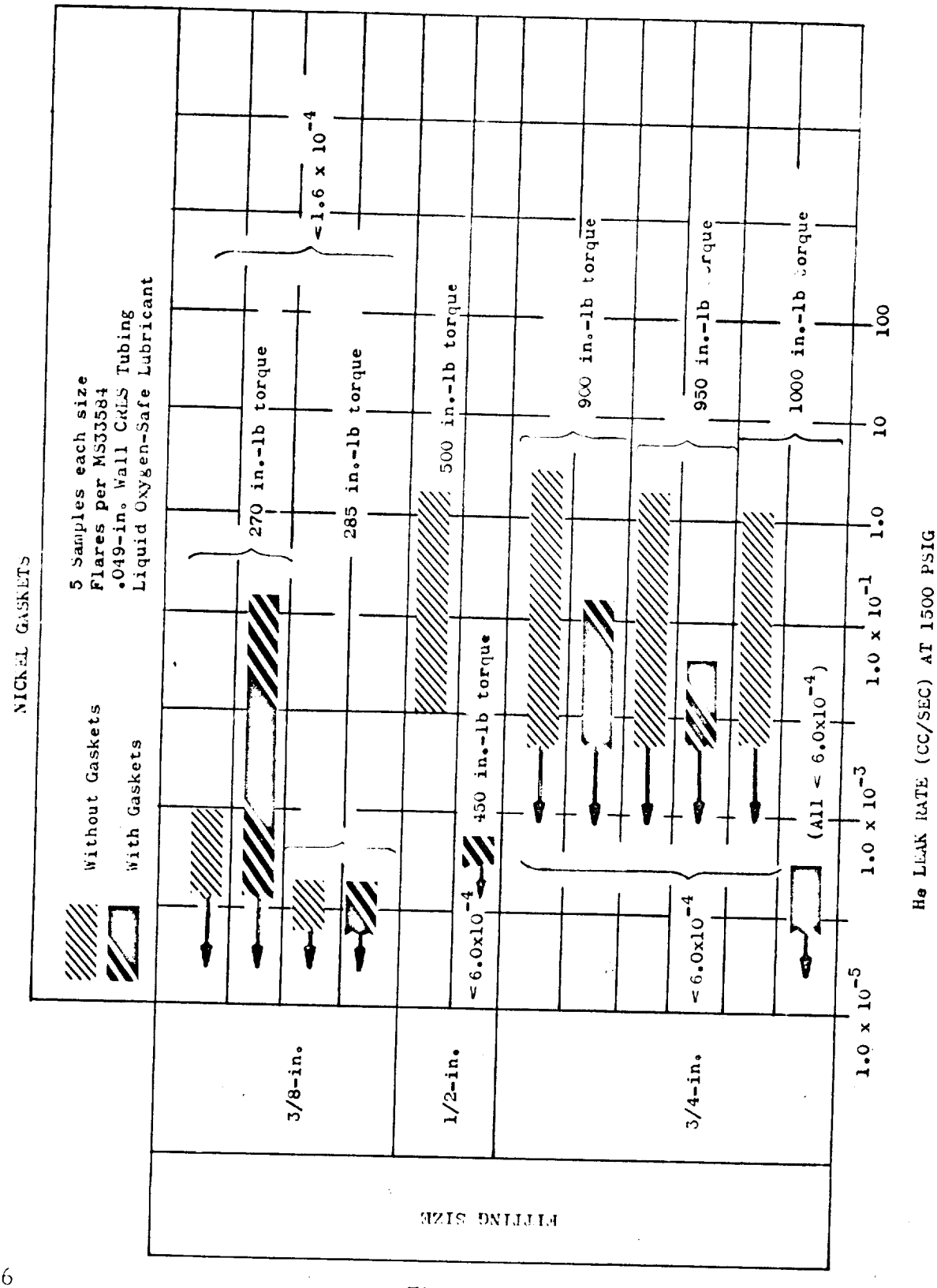


Figure 93
Leak Rate Comparisons, Del-Shell Gaskets and "AN" Standard Connections (Flared Tube-to-Fitting Joints)

SPACETITE SEALS INC. SERRATED NOSE FITTING

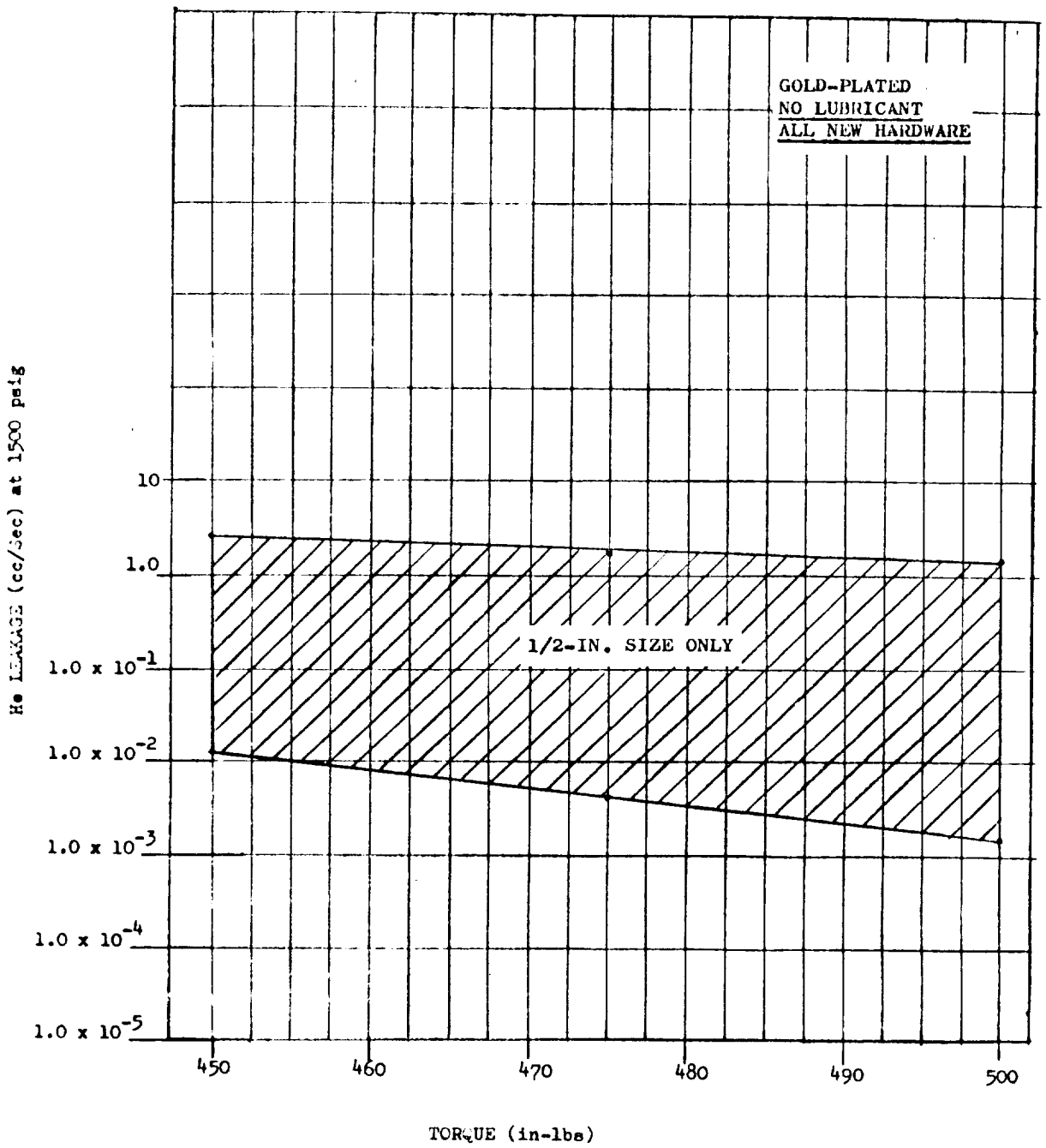


Figure 94
Torque vs. Leakage
Sheet 1 - Spacetite Seals, Inc., Serrated Nose Fitting

VOI-SHAN CONICAL BOSS GASKETS

(VSD1000 BOSSES & AN815C FITTINGS)

RLDDYLUBE - 100 LUBRICANT ON MALE THDS. ONLY

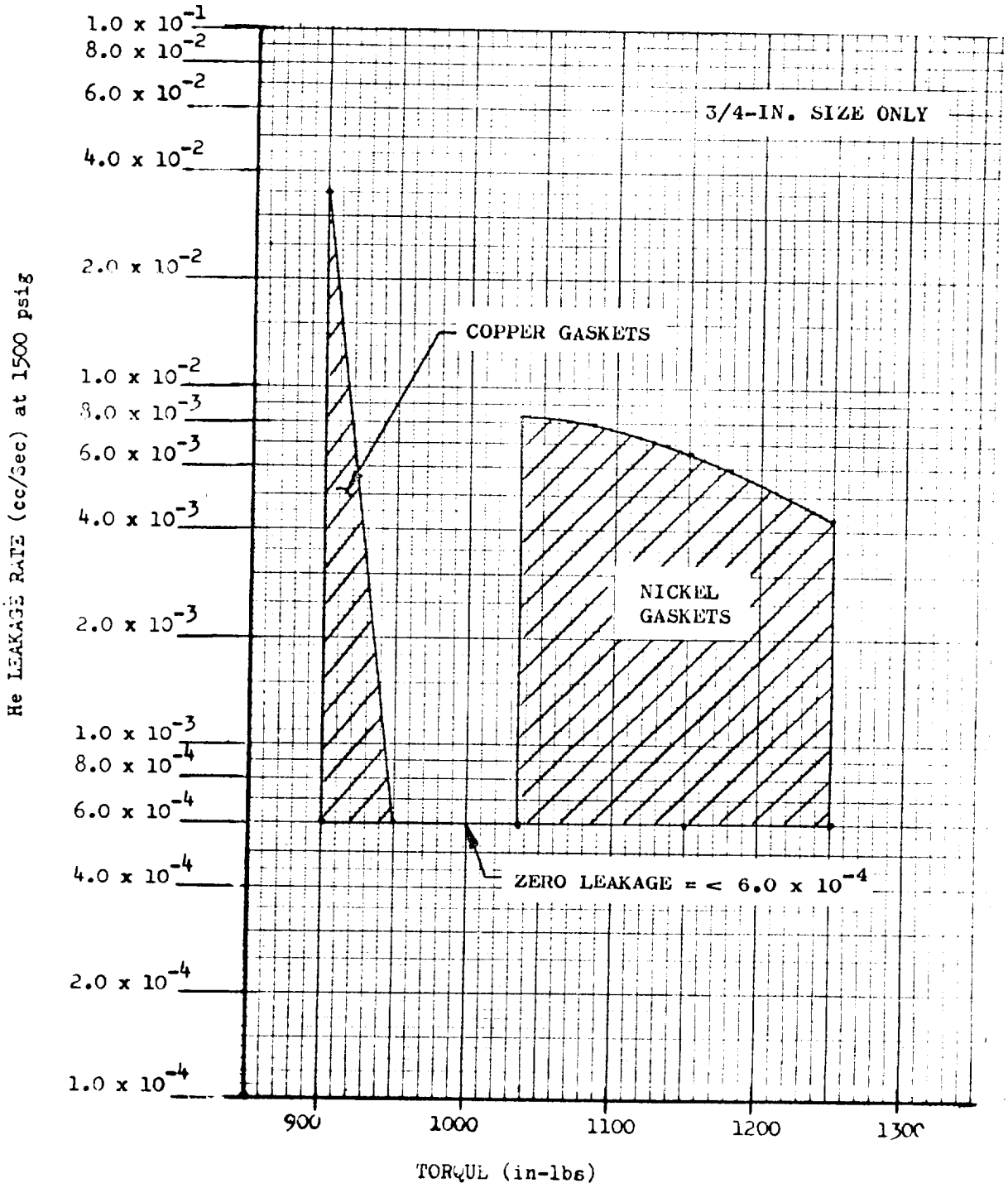


Figure 94
Torque vs. Leakage
Sheet 2 - Torque vs. Leakage, Voishan Conical Boss Gaskets

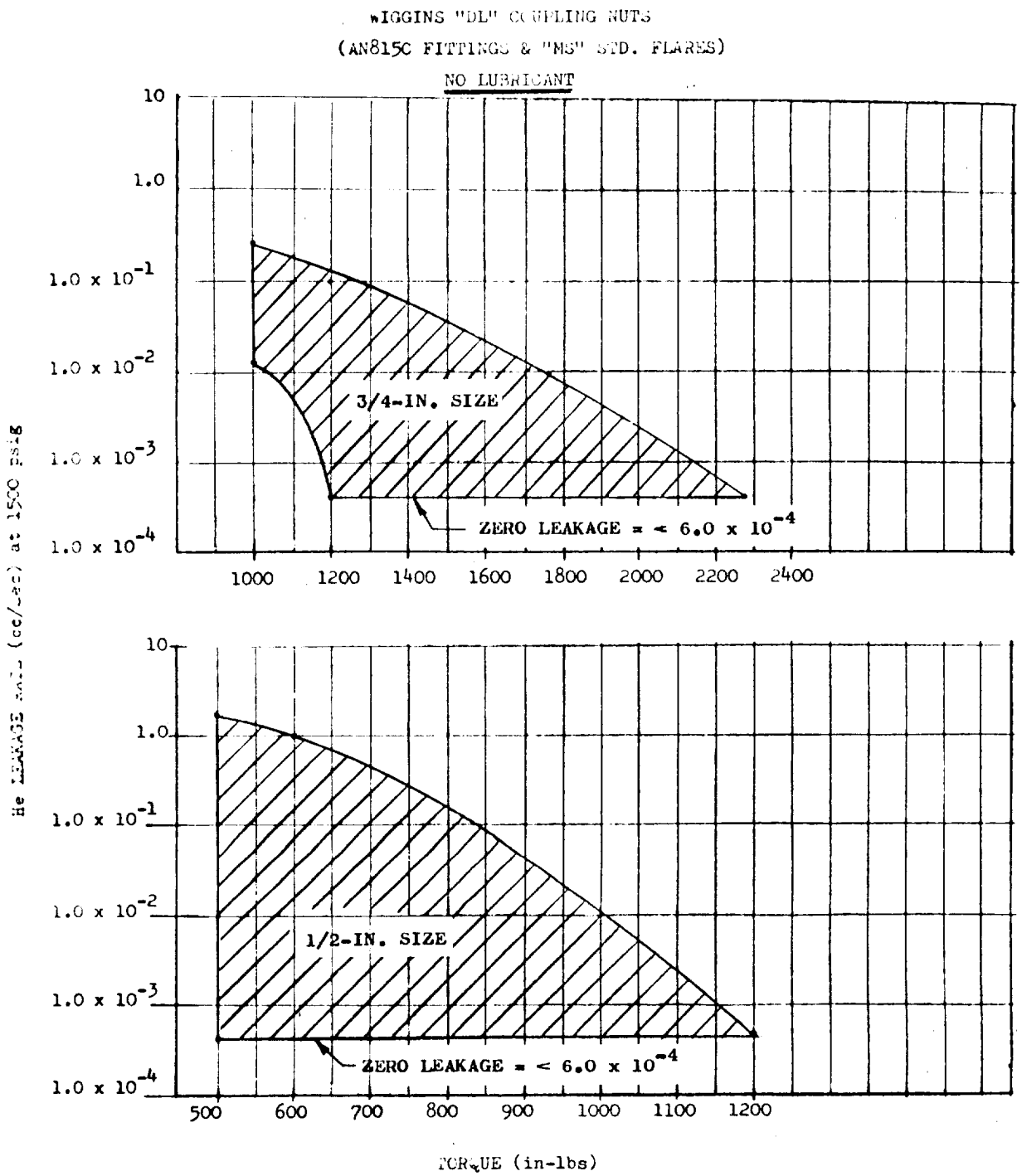


Figure 95
Torque vs. Leakage
Sheet 1 - Wiggins "DL" Coupling Nuts

STANDARD FLARED TUBE CONNECTIONS

UNFOLLISHED (MMS STD) FLARES AND (AN 818)

FITTING SURFACE

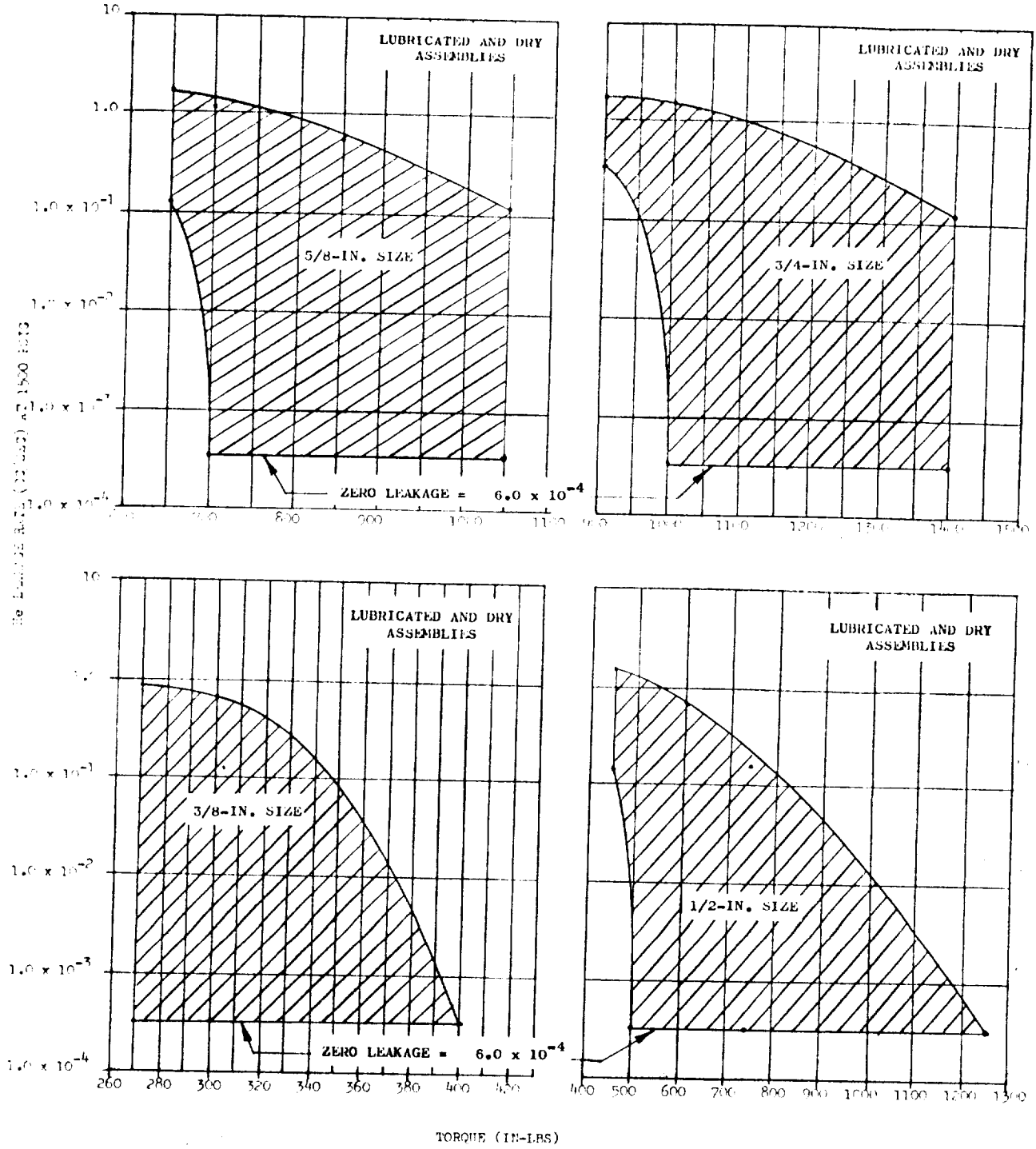


Figure 95
Torque vs. Leakage
Sheet 2 - Standard Flared Tube Connections

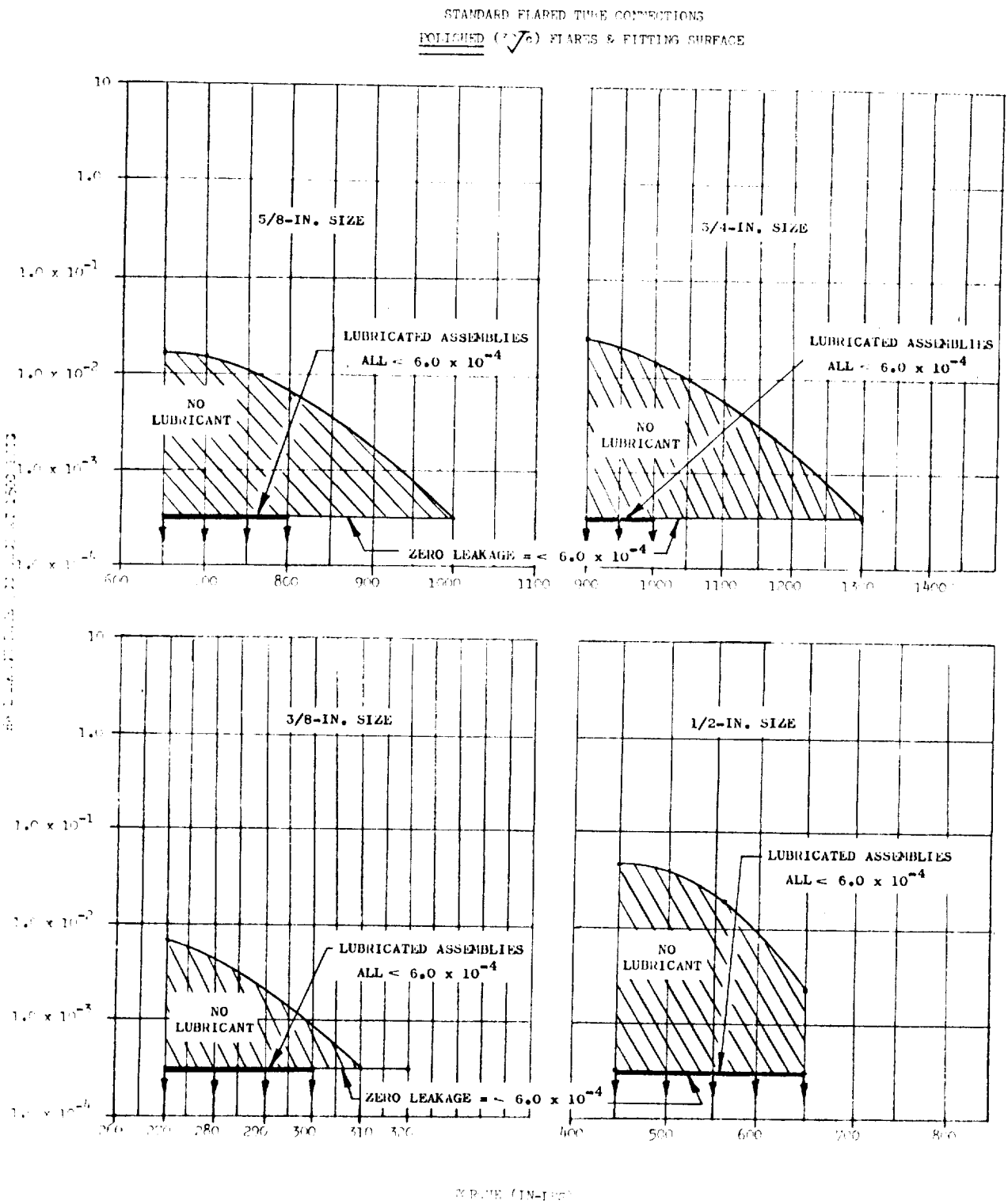


Figure 95
Torque vs. Leakage
Sheet 3 - Standard Flared Tube Connections

5. Pressure Cycling Tests

The results of those pressure cycling tests performed with the connections are shown on Figure No. 96. No failures (separation or permanent distortion) were encountered with the connections tested.

6. Thermal Shock Test

a. Low Temperature Thermal Shock Test

Results of the low temperature thermal shock tests performed with the connections are shown on Figure No. 97. No failures (separation or permanent distortion) were encountered in the connections tested. Torque relaxation was encountered with some of the threaded connections and this is shown on Figure No. 98. For those joints which showed high leakage rates during the thermal cycling test, a special low temperature test was conducted to determine the limiting low temperature range. The curves of the leakage rates for these connections were attained at various temperatures and are shown on Figures No. 99, No. 100, and No. 101.

Tests with standard and oversized cross-section Teflon O-rings in the standard flared tube fitting bulkhead and straight union-to-boss connections indicate that to maintain gaseous helium leakage rates of less than 1.6×10^{-3} cc/sec (no bubbles in three minutes) when temperatures are reduced, the volume of the O-ring must be increased. The standard (ARP568-900 series) size O-rings are not completely constrained in the boss-to-fitting gland. Cold flow of the teflon (at ambient temperature) and the contraction characteristics of the material cause the standard cross-section gasket to leak almost immediately as the temperature is lowered. As the cross-section diameter is increased (with an associated increase in volume), the Teflon is increasingly constrained in the gland and the temperature at which leakage occurs is decreased. However, the necessary increase in volume is different for the bulkhead union joint than it is for the straight union joint and varies with the nominal size of the connection. Also, the mean diameter of the O-ring must remain the same as the standard O-ring to prevent extrusion of the Teflon between the fitting and boss faces. The induced assembly problems were previously discussed. For service temperatures down to -50°F, the 1/4-in., 3/8-in., and 1/2-in. joints required a 55% minimum volume increase for the bulkhead joint and a 46% minimum volume increase for the straight union joint.

b. High Temperature Thermal Shock Test

No high temperature thermal shock tests were conducted prior to the program termination.

7. Repeated Assembly Test

The results of the repeated assembly tests with the connections tested are shown on Figures No. 102 through No. 110. The repeated assembly tests proved the importance of thoroughly cleaning the connection

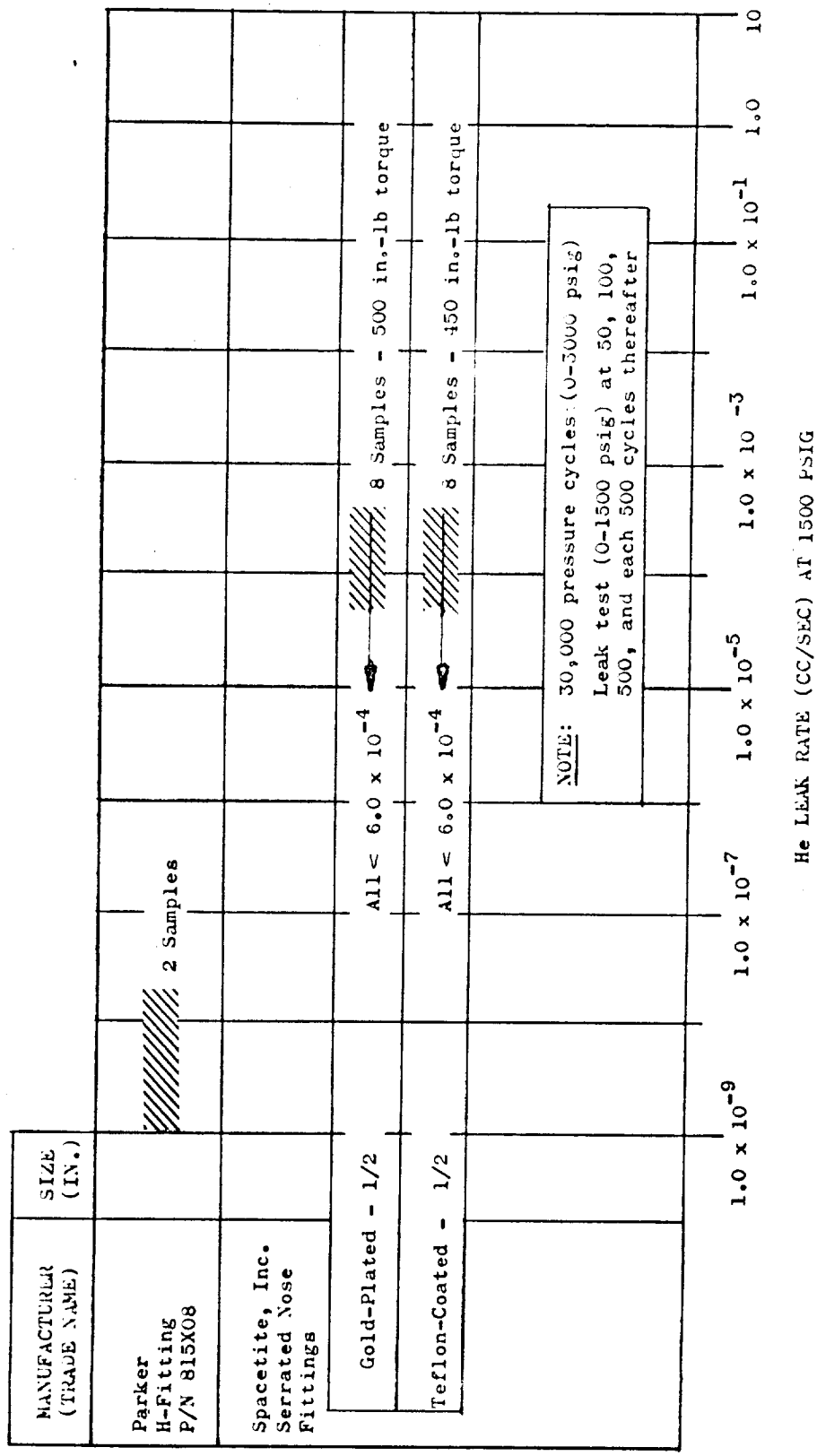


Figure 96

Results of Pressure Cycling Tests (Ambient Temperature and Results of Thermal Cycling Tests - 320°F to + 200°F

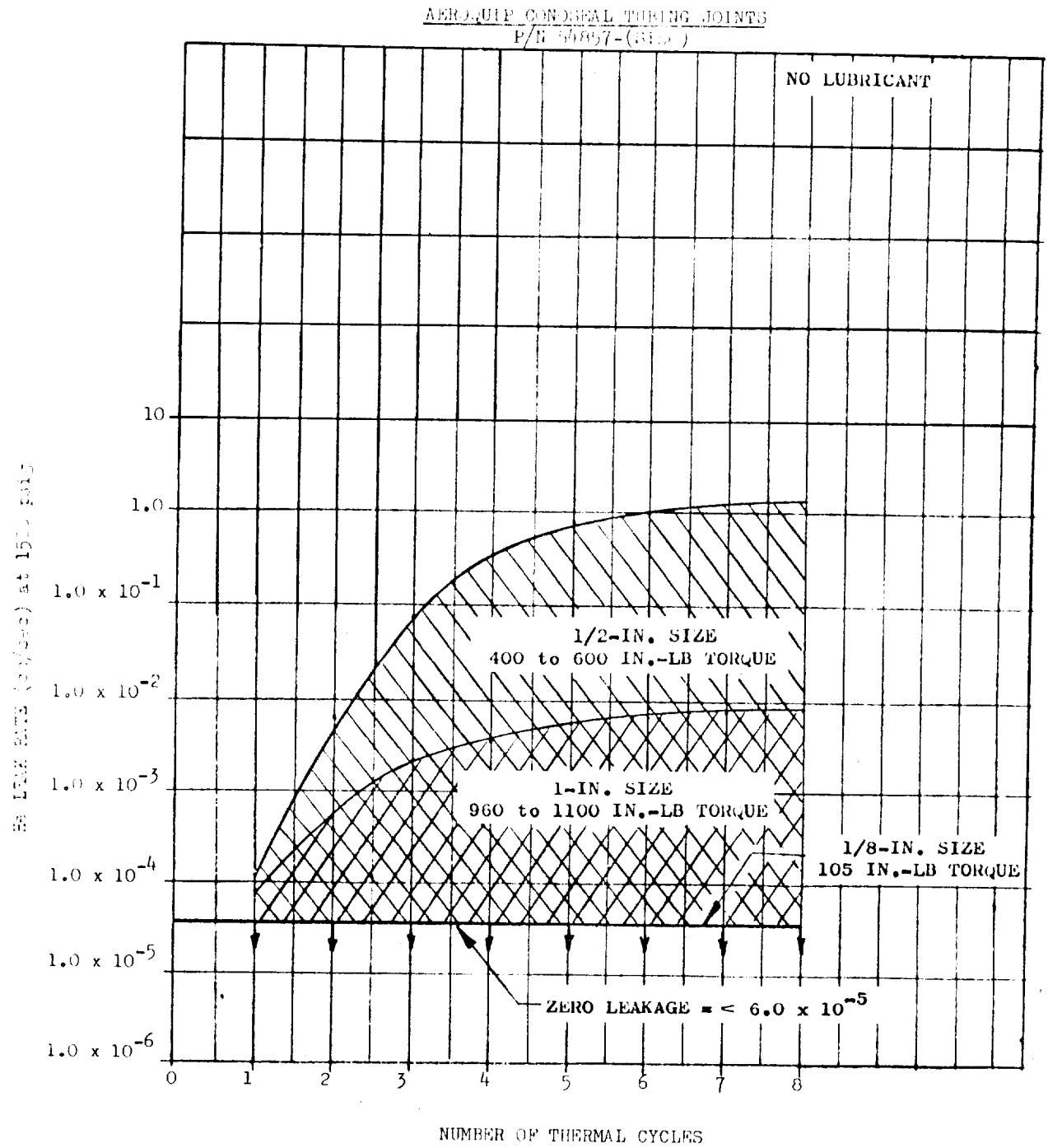


Figure 97
Results of Low Temperature Thermal Cycling Tests
Sheet 1 - Leakage vs. Number of Thermal Cycles

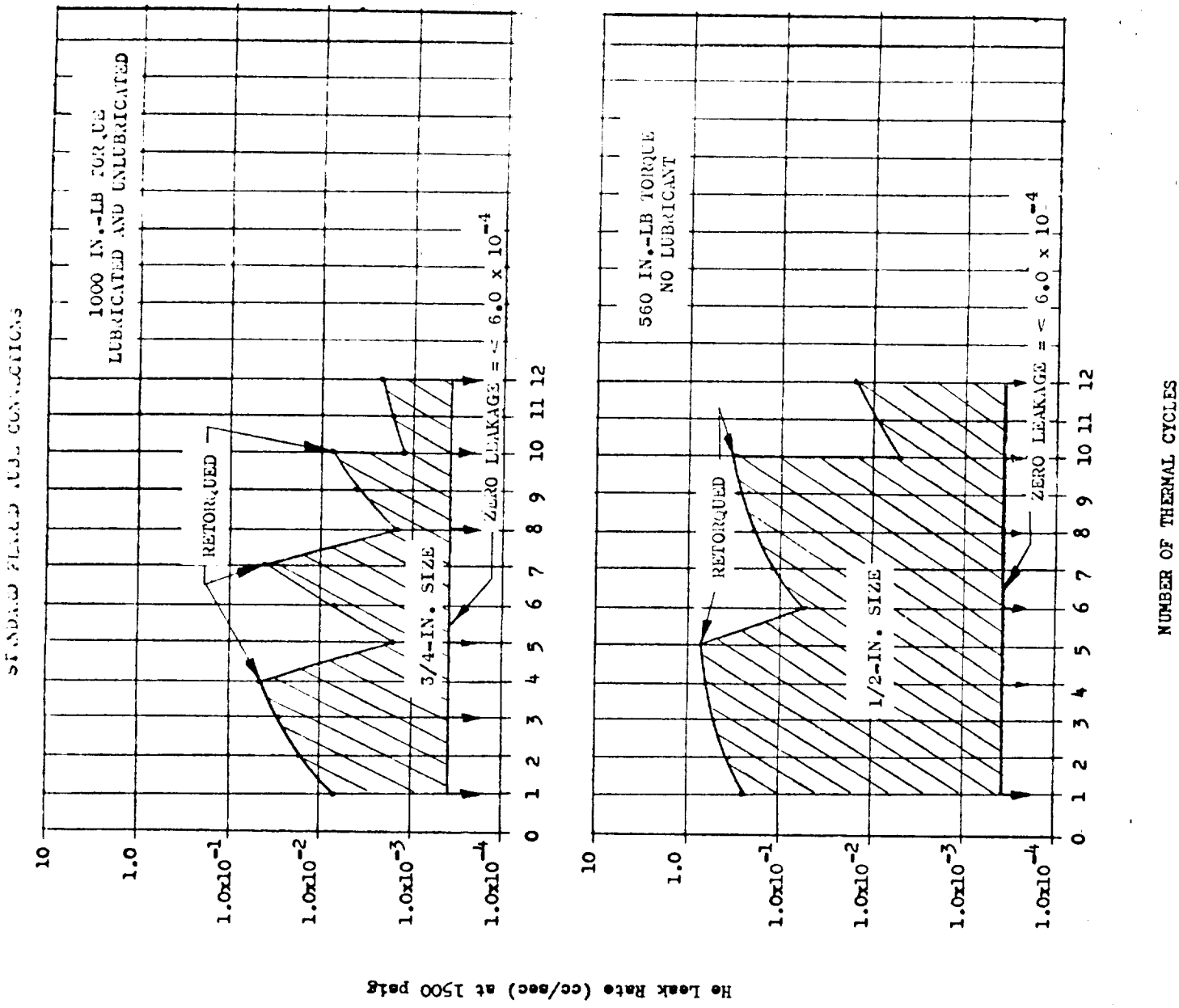


Figure 97
Results of Low Temperature Thermal Cycling Tests
Sheet 2 - Leakage vs. Number of Thermal Cycles

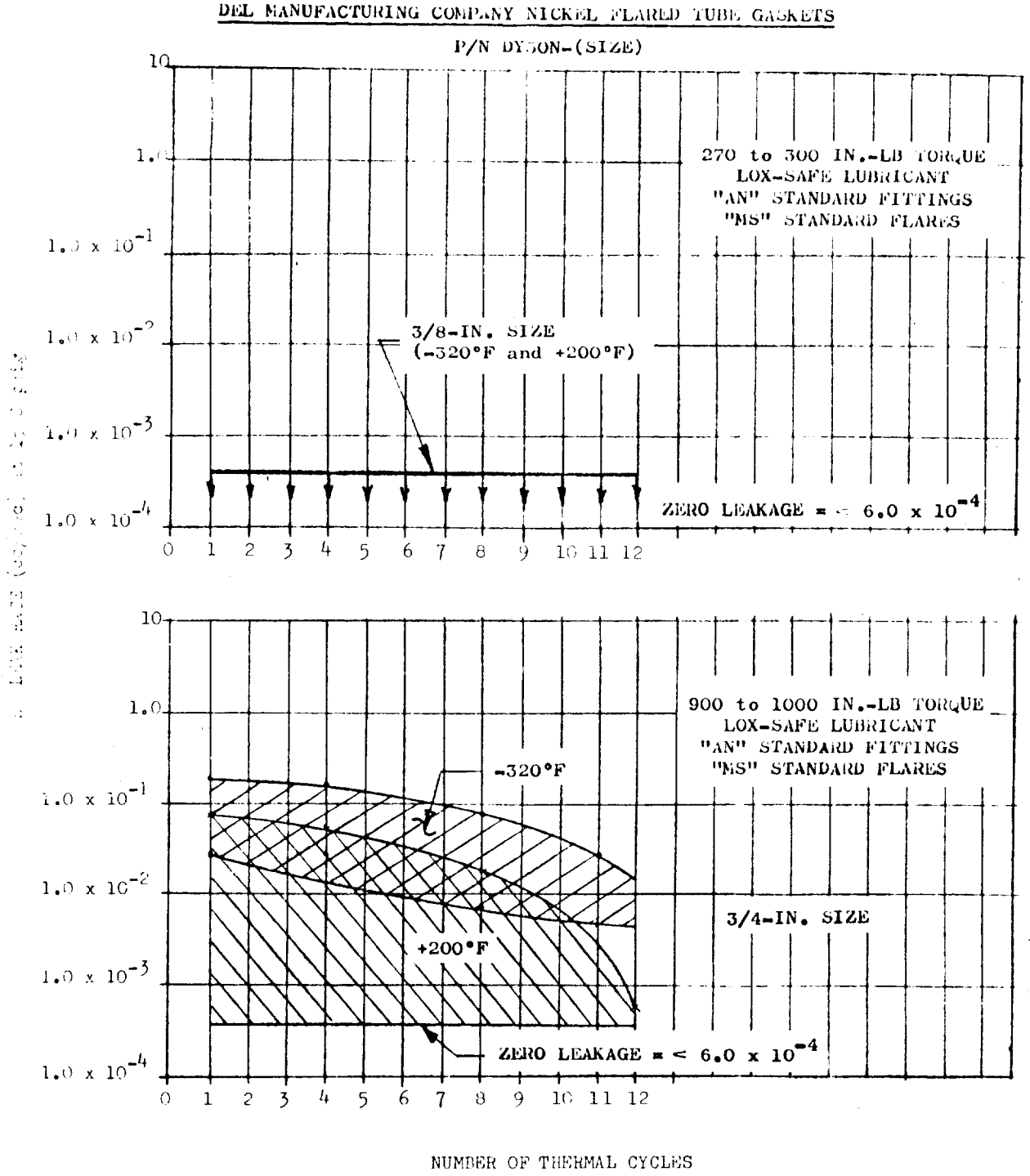


Figure 97
Results of Low Temperature Thermal Cycling Tests
Sheet 3 - Leakage vs. Number of Thermal Cycles

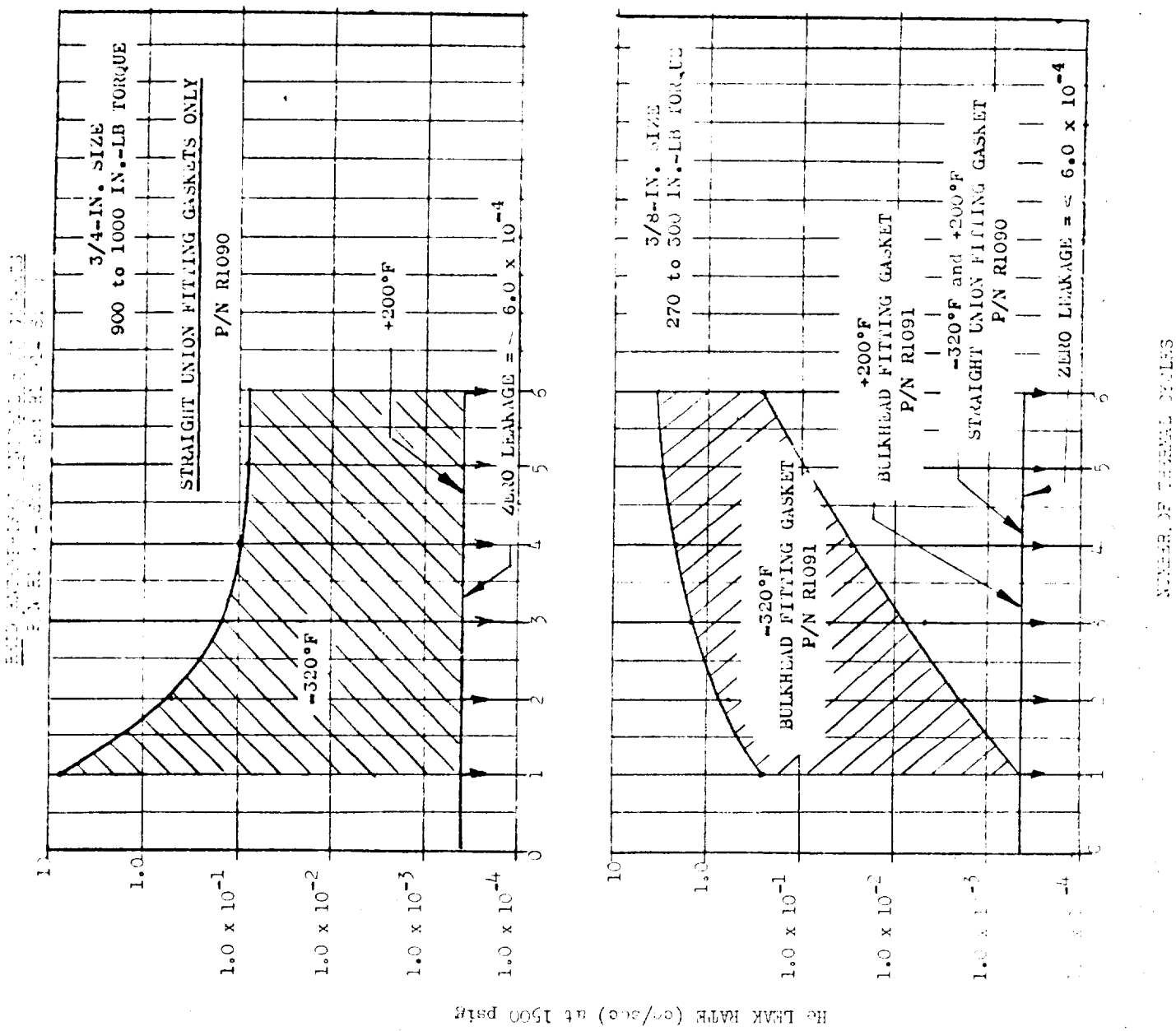


Figure 97
Results of Low Temperature Thermal Cycling Tests
Sheet 4 - Leakage vs. Number of Thermal Cycles

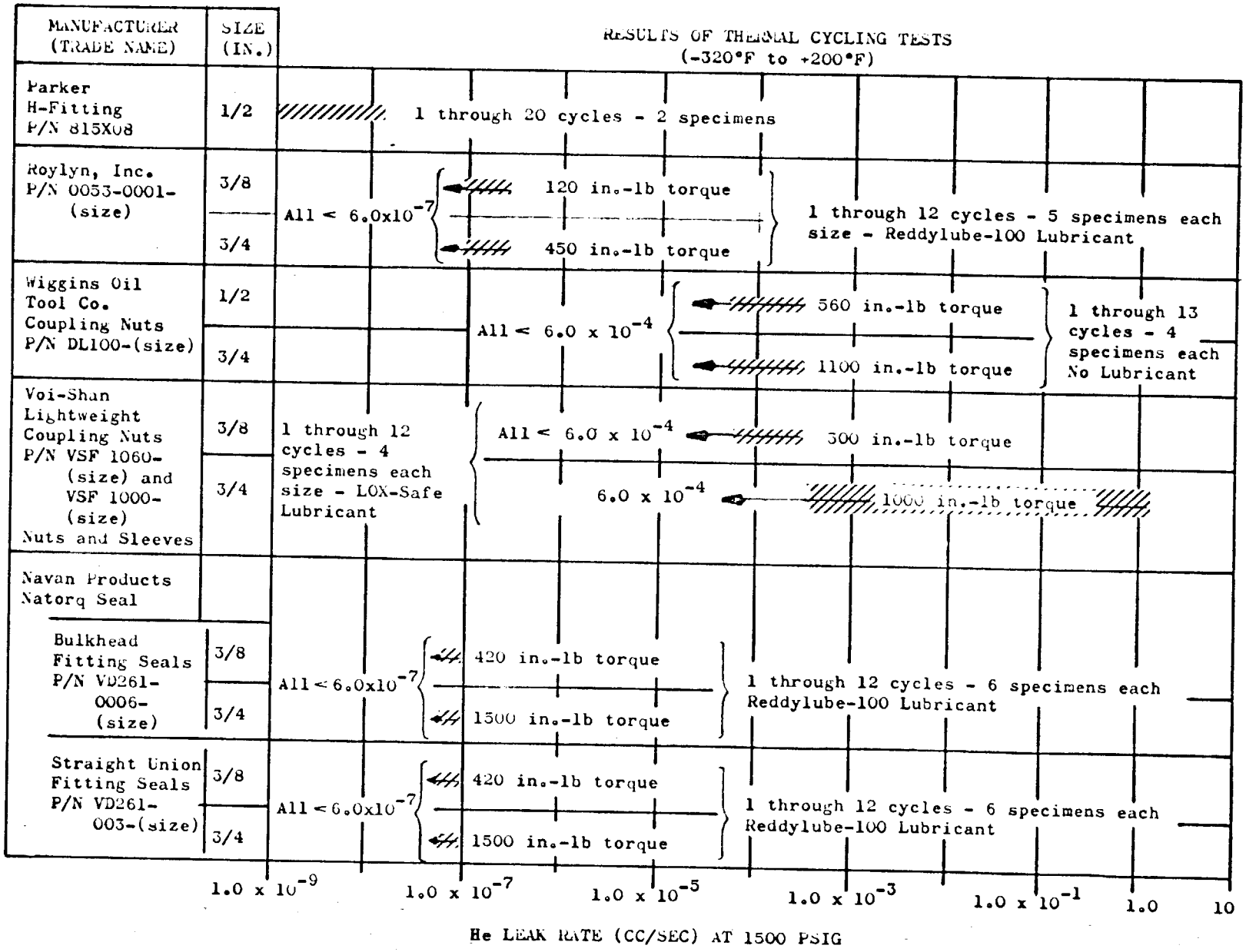


Figure 97

Sheet 5 - Results of Low Temperature Thermal Cycling Tests

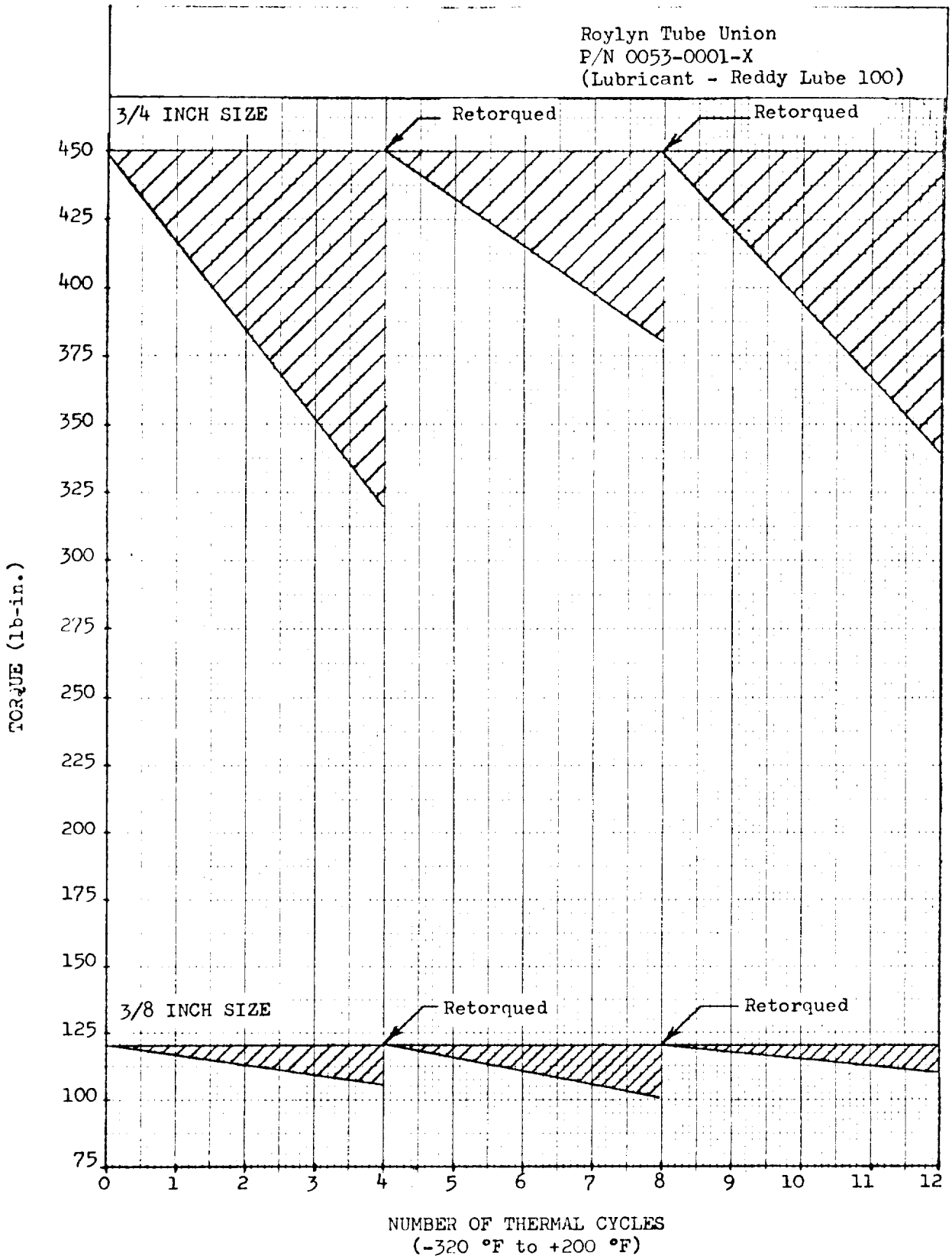


Figure 98
Torque vs. Number of Thermal Cycles

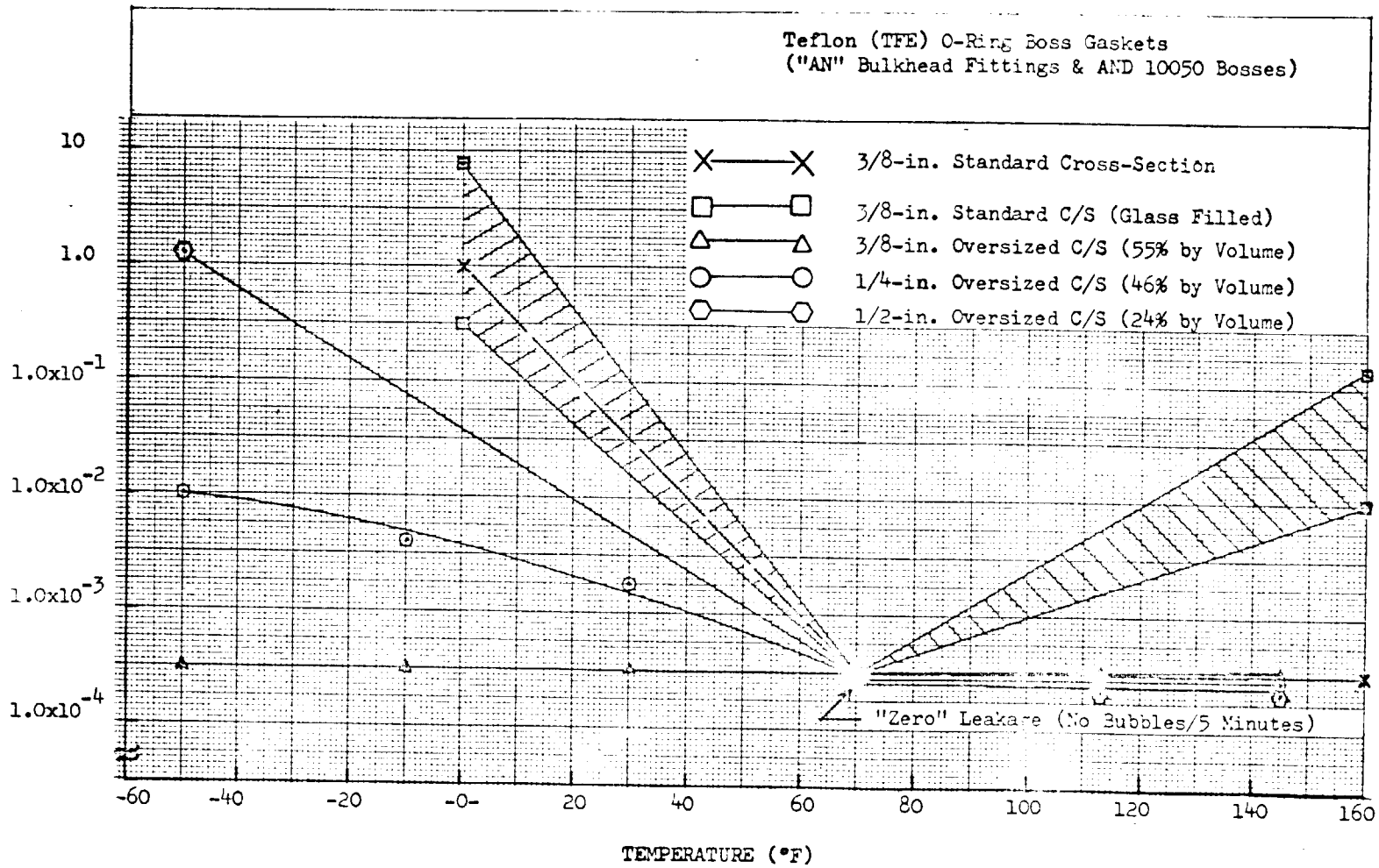


Figure 99
Leak Rate vs. Temperature (TFE)

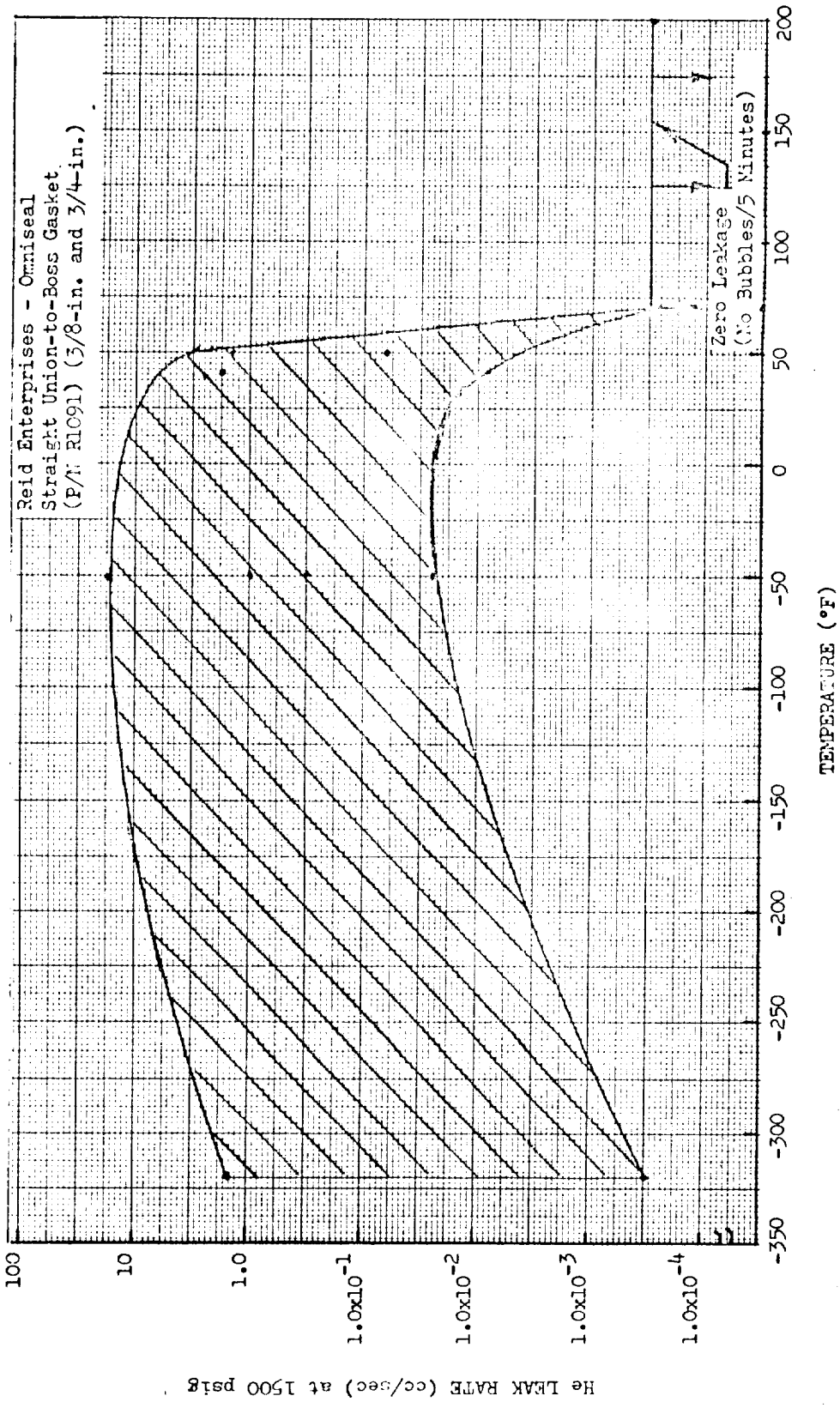


Figure 100
Leak Rate vs. Temperature (Reid Enterprises)

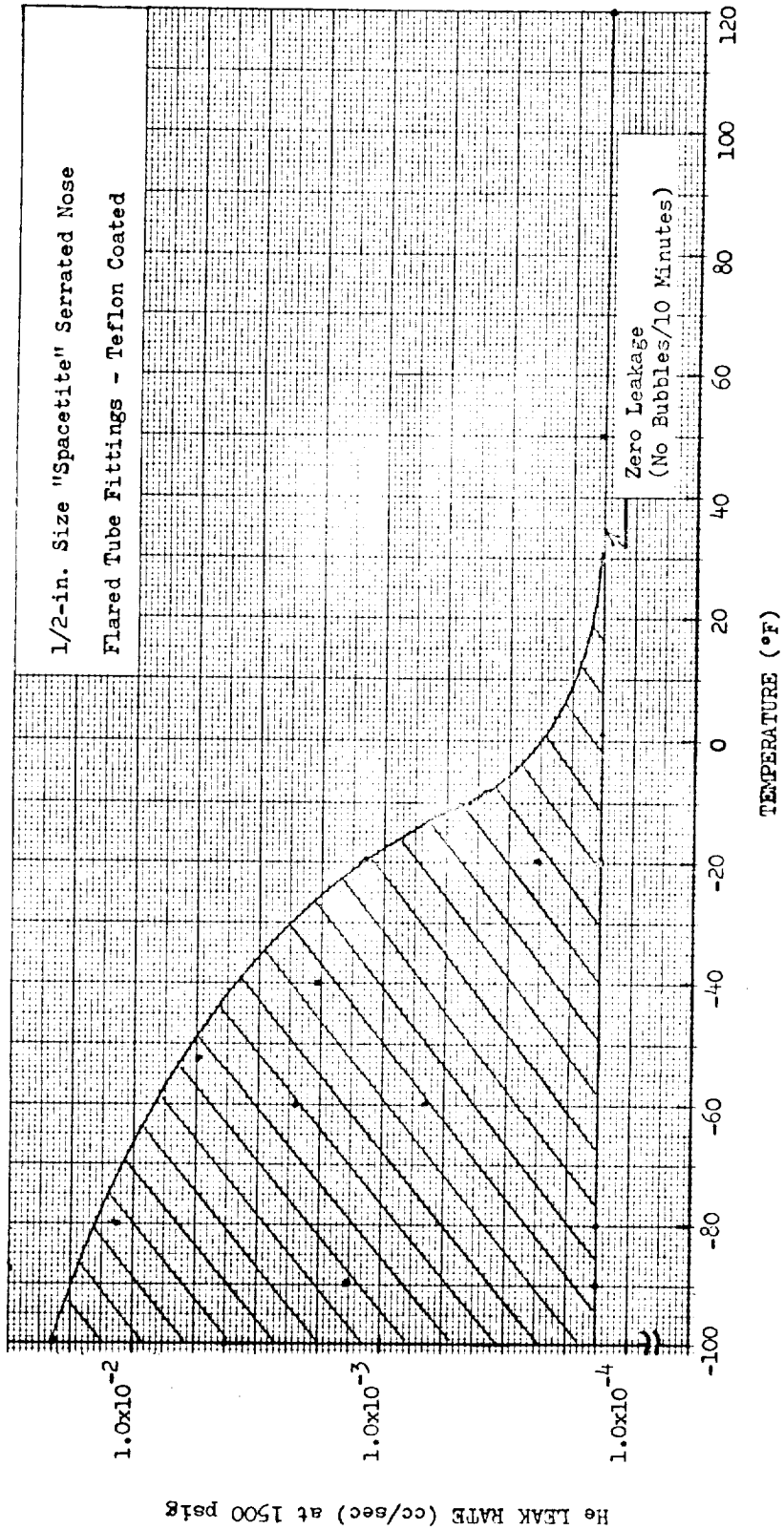


Figure 101
Leak Rate vs. Temperature (Spacetite)

AEROQUIP CORP. "CONOSEAL" TUBING JOINTS

P/N 54857 - (Size)

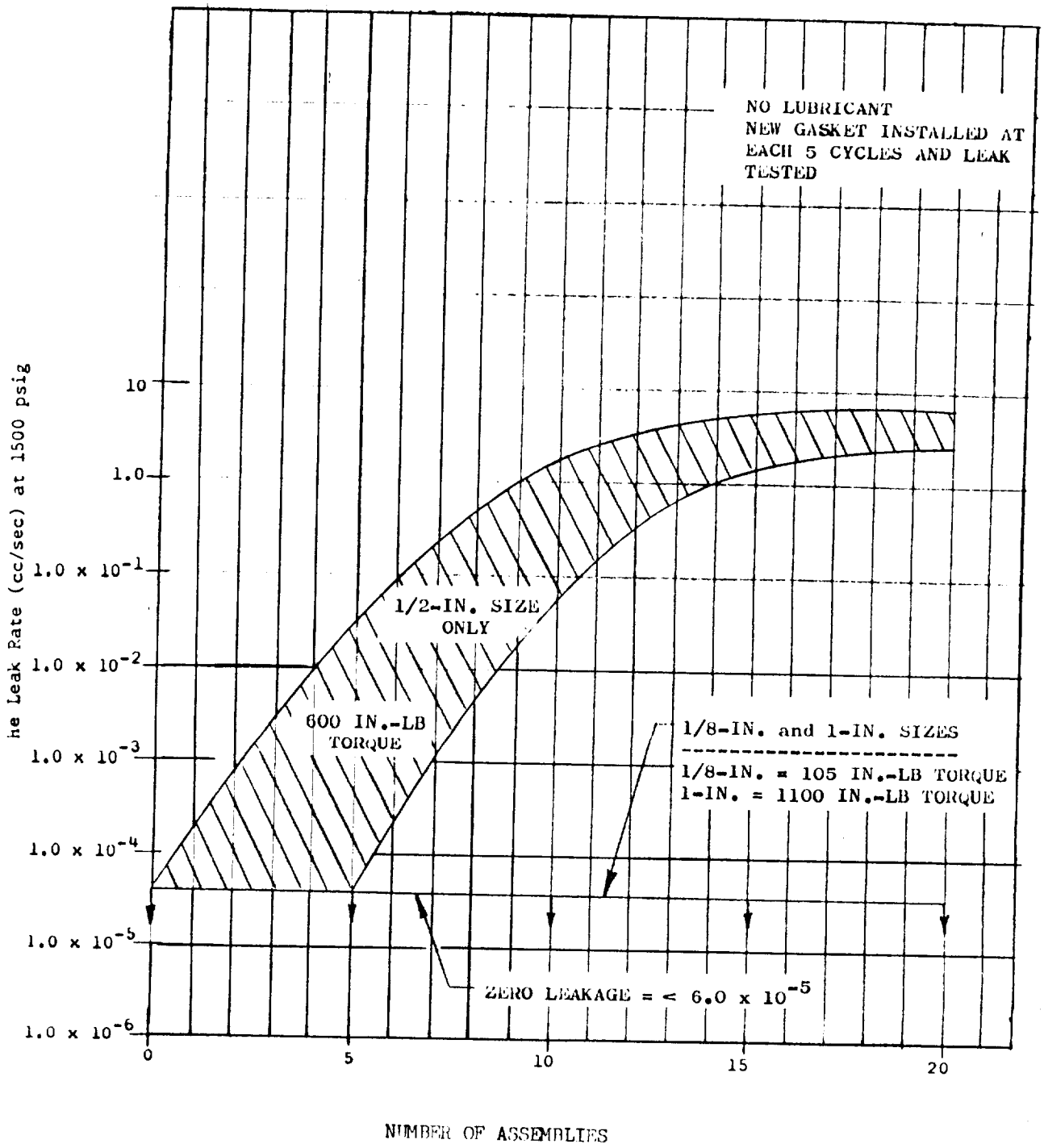


Figure 102
Leakage vs. Number of Assemblies (Aeroquip Corp.)

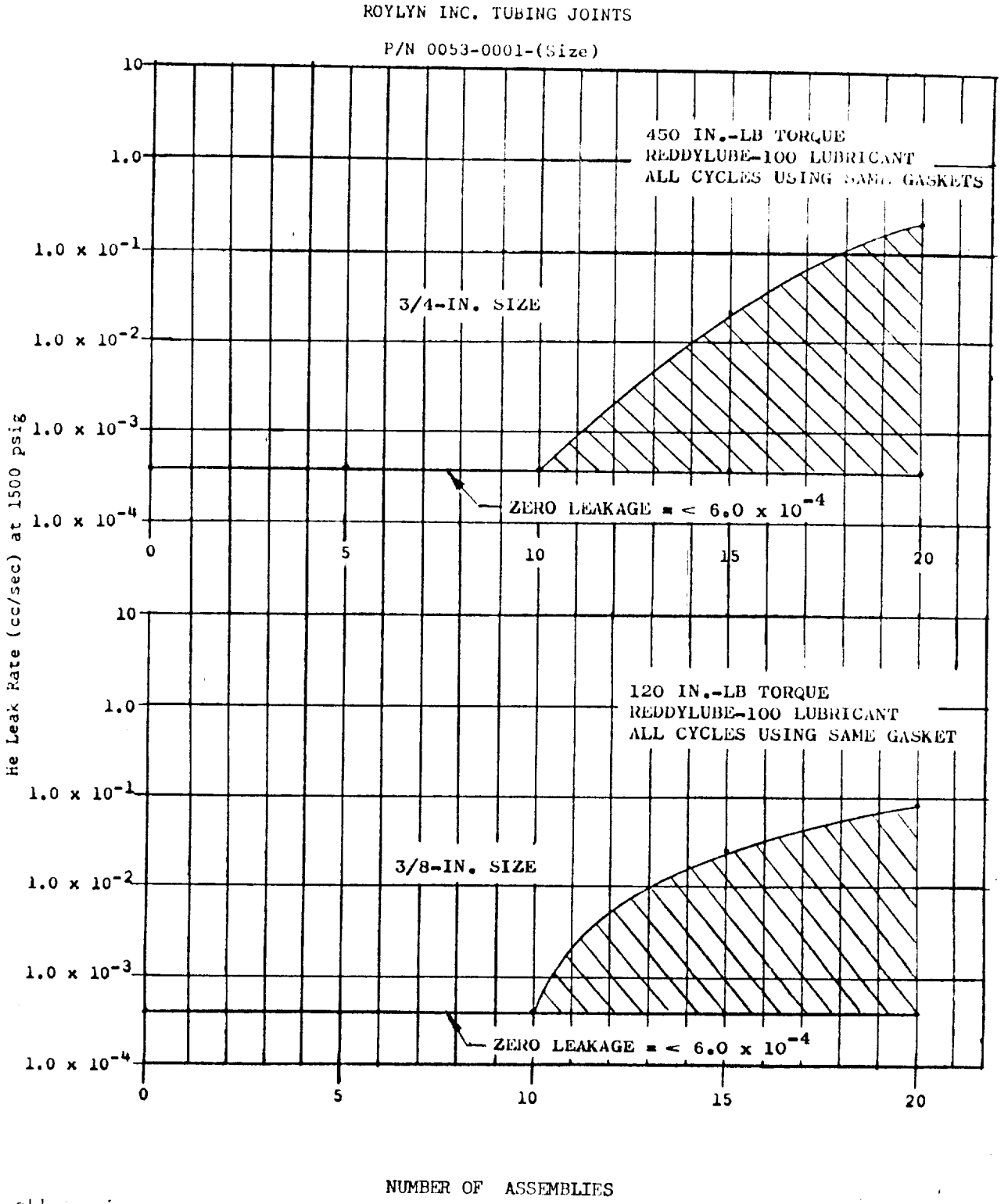


Figure 103
Leakage vs. Number of Assemblies (Roylyn, Inc.)

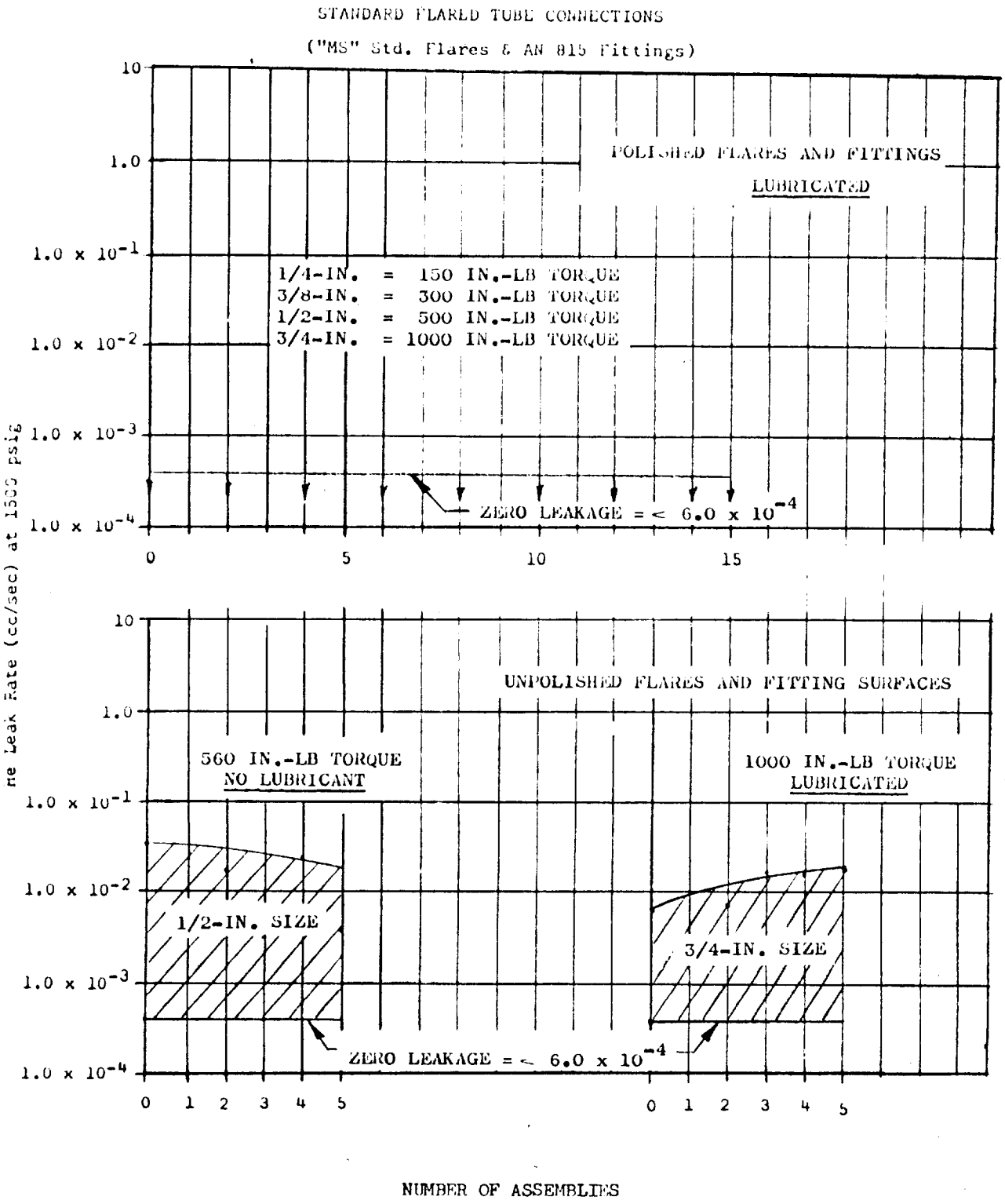


Figure 104
Leakage vs. Number of Assemblies (Standard Flared Tube Connections)

"MS" STANDARD FLARELESS TUBE CONNECTIONS
3 SPECIMENS EACH SIZE

NO LUBRICANT

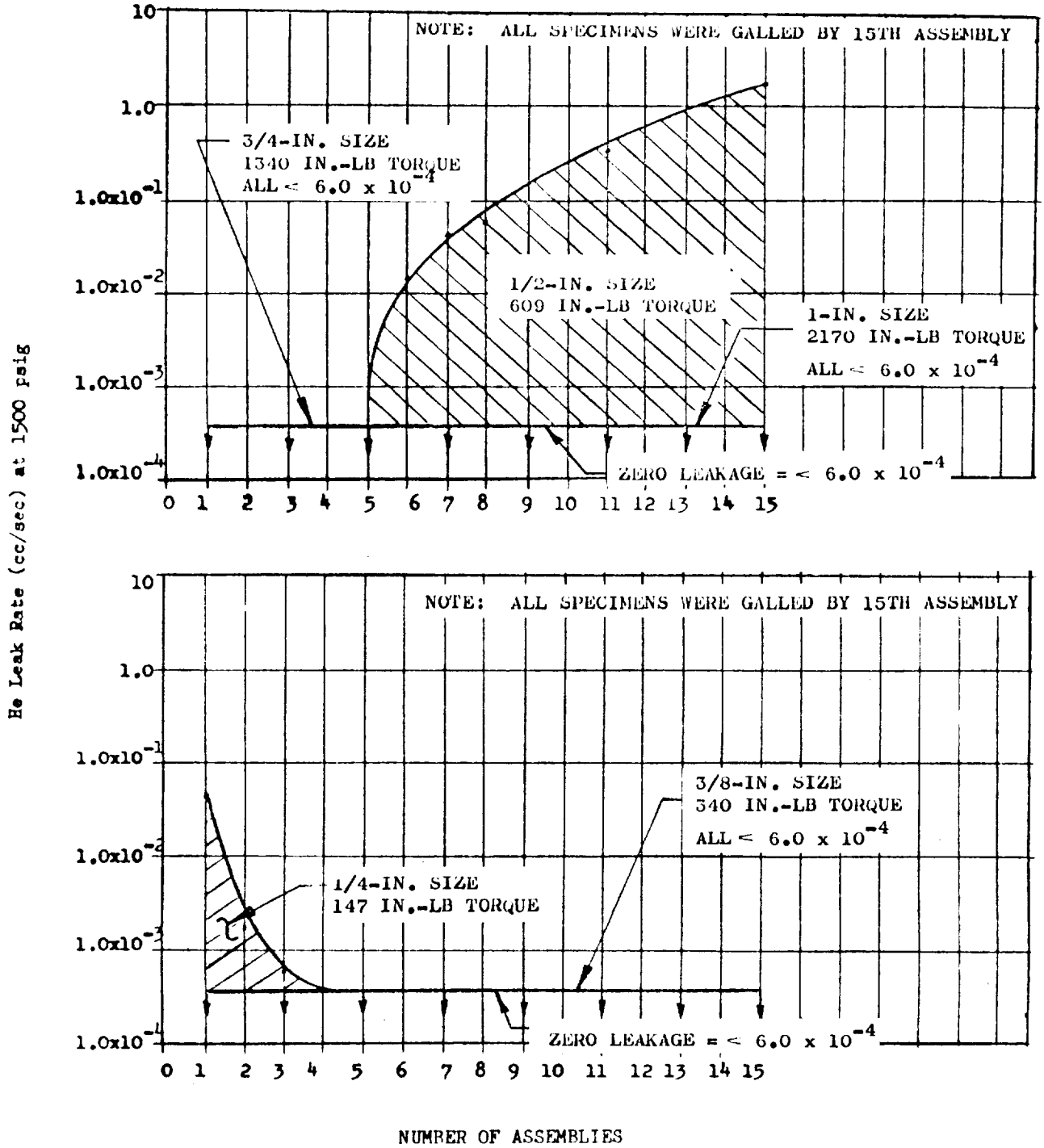


Figure 105
Leakage vs. Number of Assemblies "MS"
(Standard Flareless Tube Connections)

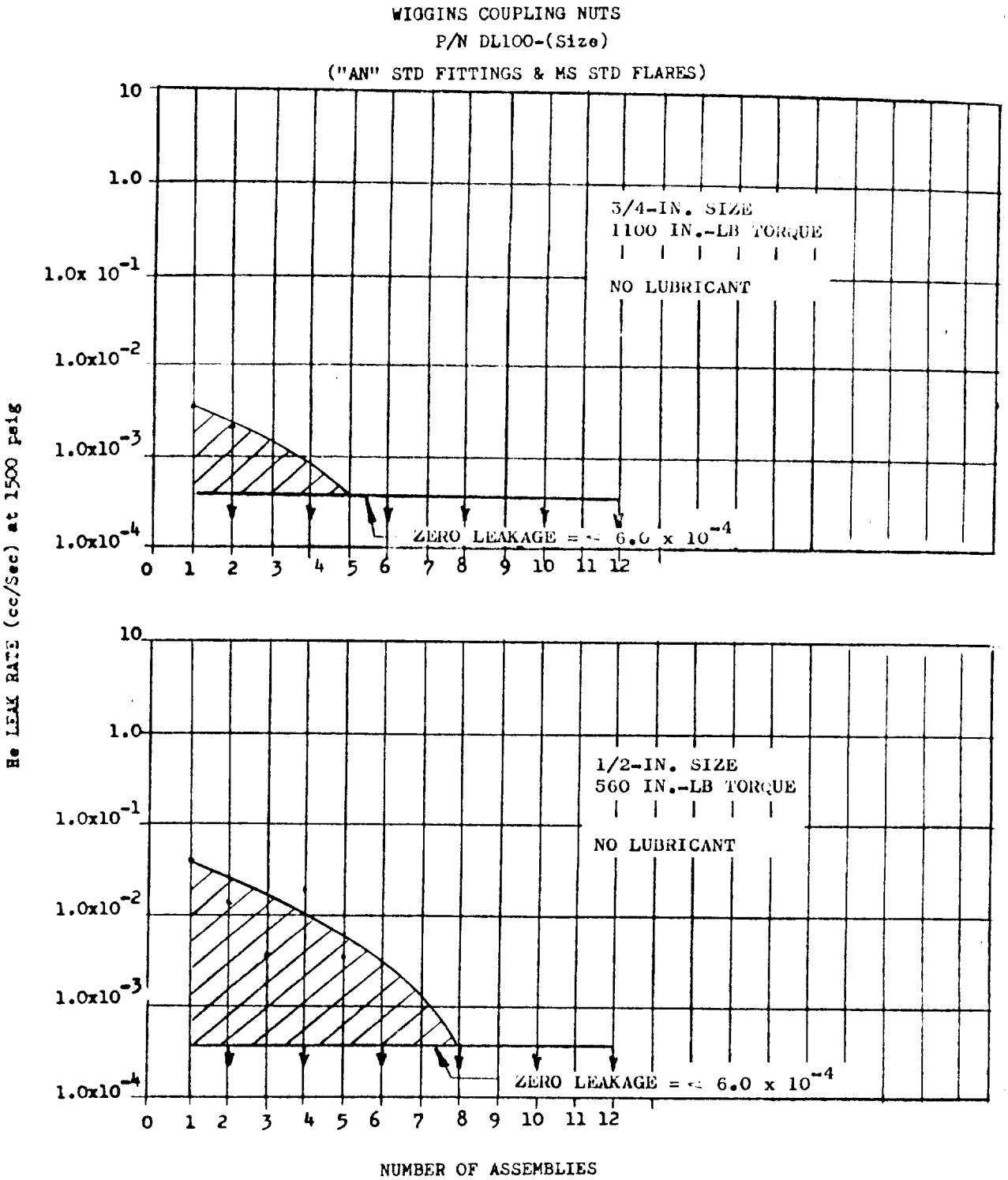


Figure 106
Leakage vs. Number of Assemblies (Wiggins)

SPACETITE SEALS INC.

SERRATED NOSE FLARED TUBE FITTING

CRES WITH GOLD PLATING

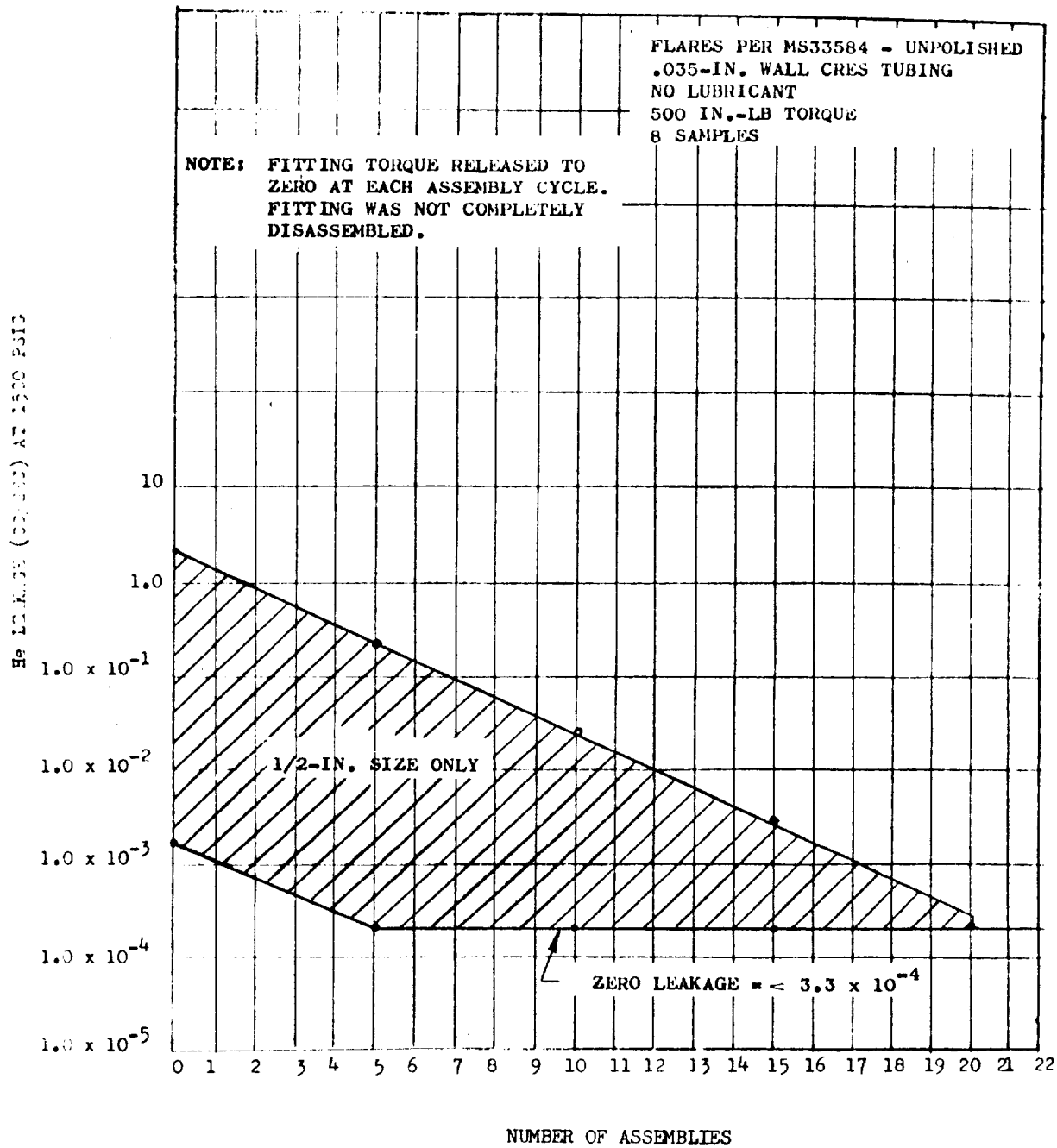


Figure 107
Leakage vs. Number of Assemblies (Spacetite)

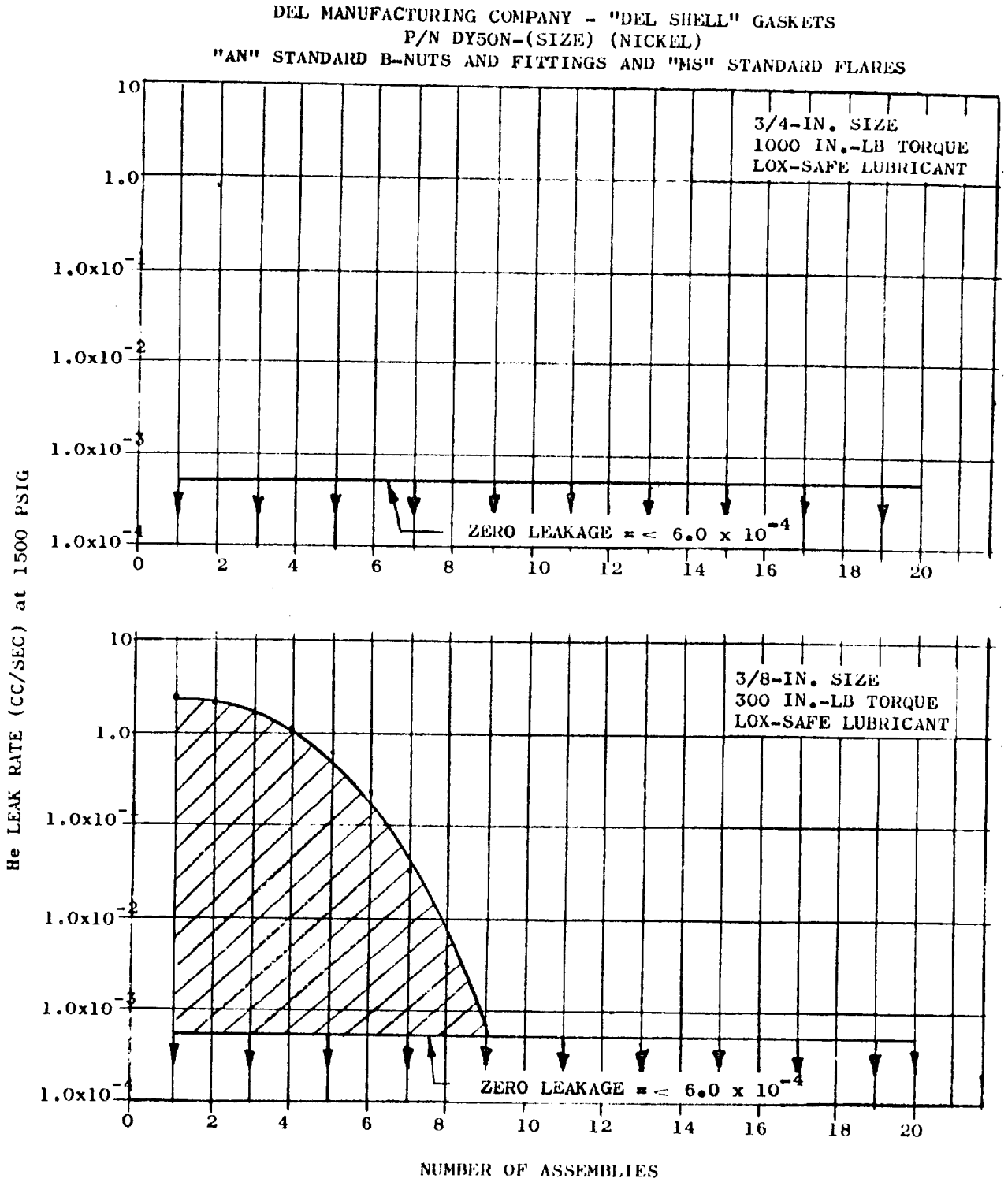


Figure 108
Leakage vs. Number of Assemblies (Del Mfg. Co.)

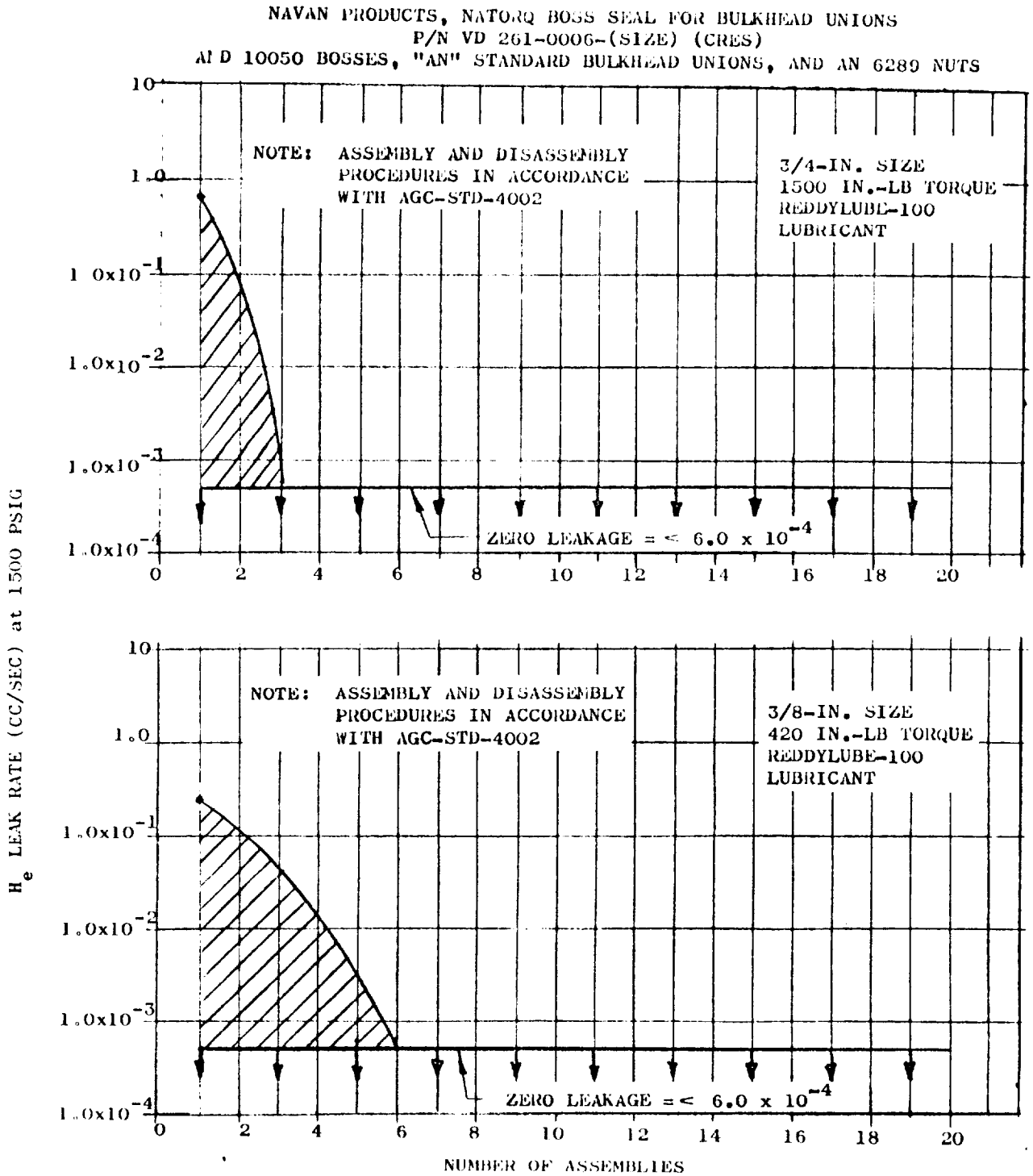


Figure 109
Leakage vs. Number of Assemblies (NAVAN Products)

MANUFACTURER (TRADE NAME)	SIZE (IN.)	REPLACEMENT COUPLING NUTS, FITTING-TO-FLARED TUBE GASKETS, AND FITTING-TO-BOSS GASKETS			
Voi-Shan Mfg. Co. Lightweight Coupling Nuts P/N VSF1016-(size) and P/N VSF100-(size)	3/8	LOX-Safe Lubricant "AN" Standard Fittings and "MS" Standard Flares	All < 6.0×10^{-4}	300 in.-lb torque	1 through 20 cycles
	3/4		All < 6.0×10^{-4}	1000 in.-lb torque	
Voi-Shan Mfg. Co. Conical Boss Gaskets P/N VSF1075x-(size)	3/8	Reddylube-100 Lubricant "AN" Standard Fittings P/N VSD1000-(size) Bosses	All < 6.0×10^{-4}	Copper-270 in.-lb torque	1 through 20 cycles New gasket installed each cycle.
	3/8		All < 6.0×10^{-4}	Nickel-310 in.-lb torque	
	3/4		All < 6.0×10^{-4}	Copper-1000 in.-lb torque	
	3/4		All < 6.0×10^{-4}	Nickel-1250 in.-lb torque	
Voi-Shan Mfg. Co. "Scott" Boss Gaskets P/N VSF1068x-(size)	3/8	S-122 Lubricant AND10050 Bosses and VSF1069 Unions	All < 6.0×10^{-4}	Aluminum-300 in.-lb torque	1 through 20 cycles New gasket installed each cycle
	3/8		All < 6.0×10^{-4}	Nickel-300 in.-lb torque	
Navan Products Natorq Boss Gaskets for Straight Union Fittings P/N VD261-003-(size) (CRES) "AN" Fittings and AND 10050 Bosses	1/4	S-122 Lubricant	All < 6.0×10^{-4}	360 in.-lb torque	1 through 20 cycles
	3/8	S-122 Lubricant	All < 6.0×10^{-4}	360 in.-lb torque	
	3/8	Reddylube-100	All < 6.0×10^{-4}	420 in.-lb torque	
	3/4	Reddylube-100	All < 6.0×10^{-4}	1500 in.-lb torque	

1.0×10^{-9} 1.0×10^{-7} 1.0×10^{-5} 1.0×10^{-3} 1.0×10^{-1} 1.0 10

He LEAK RATE (CC/SEC) AT 1500 PSIG

Figure 110
Assembly Cycling Test Results

and lightly lubricating the male thread before each assembly. Both cleaning and lubrication were essential in preventing galling of the threads. The sealing areas of the connection must be thoroughly clean to prevent dust or other particles from scoring or galling the critical sealing areas of both parts of the joint. The thread must be thoroughly clean to remove particles that may cause galling between threads. Also, cleaning prevents accumulation of lubrication that can contaminate the system. It further provides more repeatable axial load (sealing stresses) versus torque relationships in the connection.

The Conoseal gaskets are not reusable; therefore, new gaskets were used in these connections at each assembly. No lubricants were used on the gaskets and no galling or scoring of the mating surfaces occurred when the sealing area was thoroughly cleaned before each assembly. The male threads and the contact area between the nut and the female flange were throughly cleaned and lightly lubricated at each assembly. No galling or scoring was noted in these areas.

The gasket in the Roylyn connection (a hardened CRES material) is supposedly reusable; however, enough permanent set had occurred after three to five cycles to cause the gasket to wedge into the connection halves which made reassembly very difficult. As shown on Figure No. 103, leakage rates increased after the tenth assembly cycle. Some galling and scoring of the mating surfaces was evident after the twentieth cycle. The male thread and contact area between the nut and the female flange was thoroughly cleaned and lightly lubricated at each assembly. No galling was noted in these areas.

The standard flared and flareless fittings also required thorough cleaning and light lubrication of the male thread as well as the contact area between the nut and the sleeve to prevent galling. The same was required for the replacement coupling nuts and the Spacetite (serrated nose) fittings.

Only the Natorq seal and the elastomeric O-ring required lubrication of the contact area of the gasket. Lubrication is not usually desirable for gaskets as this prevents intimate contact between the gaskets and the mating surfaces. However, the Natorq gasket will gall the fitting and even more seriously, the boss surface if a good high compression lubricant is not used. Lubrication is required on the elastomeric (but not the Teflon) O-ring gaskets to prevent damage to the O-ring during assembly.

The Voishan conical fitting-to-flared tube gasket usually "hung up" in the B-nut after the second or third assembly cycle and could not be removed. However, the joint could usually be reassembled in this condition. This is not desirable from a field service aspect because a second gasket could be inadvertently installed and cause torque losses or restrictions in the line.

Both the Voishan conical boss gaskets (VSF1075) and the Voishan "Scott" boss gaskets (VSF1068) had suffered some cutting damage after

two to five assembly cycles. These gaskets were replaced at each assembly in all subsequent assembly cycles. No lubricant was used on either the nickel or copper gaskets before assembly and no galling of the mating surfaces was noted.

8. Vibration Tests

Results of those vibration tests completed with various connections are shown on Table XVI. Tests with the swaged fitting, the Wiggins connector nuts, and the Natorq boss seals were conducted at ambient temperature. After the initial resonant frequency search, each of these joints was vibrated for one hour at each of the predominant resonant frequencies. The stresses encountered at the joint at each of the predominant frequencies was calculated and is tabulated on Table XVI. Vibration tests with the standard flared and flareless tube connections were conducted at -320°F. The test set-up and input displacement were designed to provide a predetermined stress at the connection during these tests. All of the tests were conducted at 1750 cycles per minute and were continued until a failure or leakage was noted. The displacements and the associated stresses are also shown on Table XVI.

9. Salt Spray Tests

The salt spray testing was discontinued before any useful data could be obtained; therefore, no data are presented for these tests.

10. Burst Tests

Burst tests were conducted with only one type of connection (the Aeroquip "SpaceCraft" Brazed Connection) before the program was terminated. The joint tested was a 0.750-in. straight tube union. The 0.032-in. wall, 347 CRES tubing burst before failure of the connection with 10 specimens at 7800 psig to 8500 psig during the initial test. The tubing length was reduced to as short a section as possible with one specimen to induce failure in the connection rather than the tubing. The connection in this specimen failed at 11,400 psig. Two additional specimens were tested in a tensile test machine (no pressure) and the tubing failed in a typical tensile fracture at 7160 lb and 7200 lb with no affect upon the actual connection. These tests are detailed in a separate report[38].

D. CONCLUSIONS

The most leak-free and trouble-free small tubing connections are the welded, brazed, or swaged ones. They are also the most lightweight of the connections. If reliable zero leakage is a requirement, the system design should be such that these connections can be used although they require space around the connection to gain access for the assembly and disassembly tools. If disconnection is desired, the length of the component bosses must be ascertained for a predetermined number of disconnections because most of them have

(38) ibid.

TABLE XVI
(Sheet 1 of 2)
VIBRATION TEST RESULTS

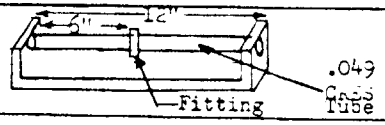
Mfr. & Type Connections	Lube	Size	No. Spec.	Resonant freq. & Amplitude or G Loading	Max. Stress @ Joint	Time to Failure (Min.)	Leak Rate Range (cc/sec)	Remarks
Parker "H" fitting (swaged)	N/A	1/2"	2	1170 cps @ 20 g's		No failure after 30 minutes	$<2.4 \times 10^{-9}$ after test	
Std. Flared Tube	Dry	1/4"	3	1750 CPM @ 1/16" displacement	40,000 psi	No failure after 330 minutes	$<6.0 \times 10^{-4}$ after test	Test temp. (-320°F) torque 135 in-lbs. polished fittings & flares
"AN" Connections			3	1750 CPM @ 3/32" displacement	59,800 psi	No failures after 1040 minutes	$<6.0 \times 10^{-4}$ after test	Test temp. (-320°F) torque 135 in-lbs. polished fittings & flares
		3/8"	3	1750 CPM @ 1/16" displacement	40,000 psi	No failures after 330 minutes	$<6.0 \times 10^{-4}$ after test	Test temp. (-320°F) torque 135 in-lbs. polished fittings & flares
			3	1750 CPM @ 3/32" displacement	59,800 psi	No failures after 1040 minutes	$<6.0 \times 10^{-4}$ after test	Test temp. (-320°F) torque 135 in-lbs. polished fittings & flares
		1/2"	3	1750 CPM @ 1/16" displacement	40,000 psi	No failures after 330 minutes	$<6.0 \times 10^{-4}$ after test	Test temp. (-320°F) torque 135 in-lbs. polished fittings & flares
			3	1750 CPM @ 3/32" displacement	59,800 psi	No failures after 1040 minutes	$<6.0 \times 10^{-4}$ after test	Test temp. (-320°F) torque 135 in-lbs. polished fittings & flares
Std. Flareless Tube Connections "MS"	Dry	1/4"	3	1750 CPM @ 1/16" displacement	40,000 psi	No failures after 330 minutes	$<6.0 \times 10^{-4}$ after test	Test temp. (-320°F) sleeve set to tube with hand setting tool.
			3	1750 CPM @ 3/32" displacement	59,800 psi	No failures after 1040 minutes	$<6.0 \times 10^{-4}$ after test	Test temp. (-320°F) sleeve set to tube with hand setting tool.

TABLE XVI
(Sheet 2 of 2)
VIBRATION TEST RESULTS

Mfr. & Type Connections	Tube	Size	No. Spec.	Resonant freq. & Amplitude or "G" Loading	Max. Stress @ Joint	Time to Failure (Min.)	Leak Rate Range (cc/sec)	Remarks
Std. Flareless Tube Connections "MS" (cont'd)	Dry	3/8"	3	1750 CPM @ 1/16" displacement	40,000 psi	No failures after 330 minutes	$< 6.0 \times 10^{-4}$ after test	Test temp (-320°F) sleeve set to tube with hand setting tool.
			3	1750 CPM @ 3/32" displacement	59,800 psi	failed(crack in sleeve groove) @ 222 minutes	> 1.0 cc/sec after test	Test temp (-320°F) Sleeve set to tube with hand setting tool.
			3	1750 CPM @ 3/32" displacement	59,800 psi	failed @ 236 minutes	> 1.0 cc/sec after test	All new Hardware Test temp (-320°F) Sleeve set to tube with hand setting tool.
			3	1750 CPM @ 3/32" displacement	59,800 psi	failed @ 364 minutes	> 1.0 cc/sec after test	All new Hardware Test temp (-320°F) Sleeve set to tube with hand setting tool.
		1/2"	3	1750 CPM @ 1/16" displacement	40,000 psi	No failures after 330 minutes	$< 6.0 \times 10^{-4}$ after test	Test temp (-320°F) Sleeve set to tube with hand setting tool
			3	1750 CPM @ 3/32" Displacement	59,800 psi	Failed(cracked @ sleeve groove) @ 278 minutes	> 1.0 cc/sec after test	Test temp (-320°F) Sleeve set to tube with hand setting tool
			3	1750 CPM @ 3/32" displacement	59,800 psi	Failed @ 203 minutes	> 1.0 cc/sec after test	New Hardware Test temp (-320°F) Sleeve set to tube with hand setting tool
			3	1750 CPM @ 3/32" displacement	59,800 psi	Failed @ 95 minutes	> 1.0 cc/sec after test	New Hardware Test temp (-320°F) Sleeve set to tube with hand setting tool
Natorq boss gasket VC261-	S-122	1/4"	1	540 cps to 860 cps .4 to .7 g's		No failure after 30 mins	$< 6.0 \times 10^{-4}$ after test	No leakage or torque relaxation.
			1	2500 cps @ .004" displacement		No failure after 30 mins	$< 6.0 \times 10^{-4}$ after test	No leakage or torque relaxation
			1	100 cps @ 5.0 inches/sec		No failure after 30 mins	$< 6.0 \times 10^{-4}$ after test	No leakage or torque relaxation
			1	55 cps @ .004" displacement		No failure after 30 mins	$< 6.0 \times 10^{-4}$ after test	No leakage or torque relaxation

to be cut for each disconnection. This extra boss length is not required for the Aeroquip "SpaceCraft" brazed connection because it can be cut at the fitting itself; then, debrazed, re-dressed, and re-brazed.

The proprietary threaded connections vary with regard to leakage rate capabilities, reliability, and weight depending largely upon the manufacturer. Methods for welding, brazing, or swaging the connection to the tubing can present some problems, but no serious ones were encountered in this program when normal care was taken in welding (i.e., use of heat-sinks and careful welding procedures) and the brazing or swaging techniques were in accordance with the supplier's recommendations. The use of most of these as component connections is very difficult as compared with the standard threaded connection. Before these connections are used in any system, they should be thoroughly tested under the system operating conditions and environment.

Generally, the standard flared and flareless tube threaded connections are best suited to aircraft hydraulic systems. They are not adequate for the exotic fluids and hot gas service of space systems. This is particularly true for the newer high pressure (3,000 psig to 6,000 psig) systems now being proposed. The "MC" standards may lead to improvements, but no comparison tests were made before the M-1 seal evaluation program was terminated. However, it is the author's opinion that it is difficult to produce a satisfactory joint with any of the standard connections.

Generally, the replacement coupling nuts and fittings for the standard flared tube connections did not yield a distinct improvement in the connection. Actually, in some instances, the connection can be much worse; however, there were a few exceptions to this. Although the Wiggins "DL" coupling nut did not completely eliminate torque relaxation, the capability for higher torque without distortion of the nut can reduce leakage problems. Also, at the higher torque values, the torque relaxation problem is somewhat reduced. The higher torque induces a problem of distortion in the nose of the standard flared fitting; the fitting nose tends to distort as the compression load is increased above the standard values. The Globe Aerospace "spherical" nosed fitting can accept higher compression loads because of added material in the nose of the fitting. The spherical surface also provides somewhat higher sealing stresses with the same loading. The Voishan conical gasket can also solve leakage problems at the fitting-to-flare interface. However, it induces a new problem because the gaskets can become stacked in the nut and restrict the lines or prevent proper "grip" of the tubing flare.

In some of the more exotic fluids and the hot gas systems, metal gaskets between the boss and fitting are usually necessary. Each of the proprietary metal gaskets available for this joint present their own peculiar problems. All require improvement in the fitting and boss surfaces over the normal standard requirements if very low leakages are required. Torque requirements for some of these gaskets (i.e., Natorq) cause stresses in the fitting that exceeds the material yield strength. It is the author's opinion

that the best concept for this joint is the Voishan conical boss seal. Although this design does present the problem of gaskets possibly becoming "stacked" in the bottom of the boss, this problem has less significance than the problems solved by its use. The torque required does not overstress the fitting because fitting stresses became compression instead of tension. Also, only the fitting threads are lubricated, which reduces or eliminates the problem of system contamination with the lubricant. Teflon O-rings can be used in the standard AND-10050 boss-to-fitting joint if the service temperature range is between approximately 0°F and 500°F and if the volume of the O-ring is carefully designed to completely fill the gland volume. Different O-ring volumes are required for the same size straight union or bulkhead union joint. Assembly tools are required to fit the O-ring over the fitting threads without damaging the O-ring.

E. RECOMMENDATIONS

If reliable zero leakage connections are desired or required, the system should be designed so that the welded, brazed, or small tubing joints can be used. Each type of connection will require investigation to determine the clearances necessary around the connection for assembly and disassembly tools as well as the minimum boss lengths needed for the components if the connections must be cut out of the system. The Aeroquip "SpaceCraft" fitting is the only brazed or swaged connection that can be disassembled without removing a length of the boss or tubing. However, the "SpaceCraft" connections are currently limited to CRES materials, as are most of the brazed connections. Specifications and standards should be developed for the brazed connections because they are well along in their development. These specifications and standards should include qualification specifications, fitting standards, tubing (and boss) preparation and requirements standards, cleaning and packaging requirements standards, as well as assembly and disassembly specifications.

The current "AN" or "MS" standard tubing connections need considerable improvement to be satisfactory for most Aerospace systems. These connections could not be recommended where reliable zero leakage is required. The "MC" standard connection definitely would be an improvement in the flared connection because of the improved surfaces in all parts of the connection as well as improvement over other weaknesses of the "AN" connection. The "MC" connections are considerably more expensive, which is not the most important consideration in manrated system, and they still present the same basic problem as the standard "AN" connection. These are the mechanics involved in loading of the flare and the possible overstress of the fitting for metal gaskets in the boss-to-fitting joint. It is the author's opinion that the connection shown on Figure No. 111 would tend to eliminate these basic problems. The "DL" coupling nut and sleeve provides more direct loading upon the tubing flare. Also, the "beefed-up" part of the nut, which provides the loading for the sleeve and flare, permits increased torque over the standard values without overstressing the nut. The tubing flare would need to be the "MC" standard or at least an improved standard over the military standard flare. The

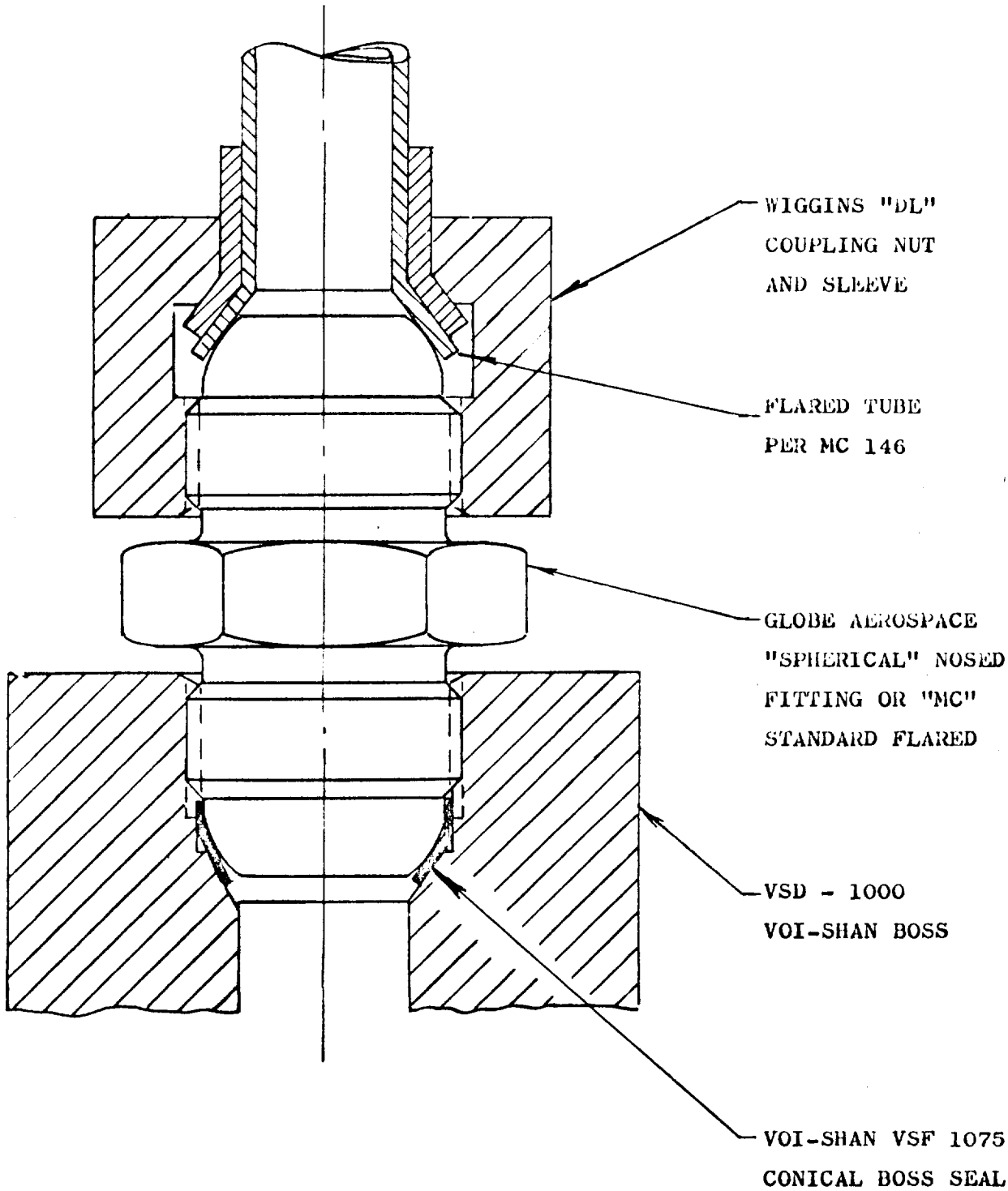


Figure 111
Recommended Flared Tube Connection

spherical nosed fitting would provide higher sealing stresses with the same loading (torque) and a slight increase in compression strength of the nose. One possible problem that could be induced is fatigue of the flare-to-tube radius from impulse cycling. A standard would be necessary for the boss (equivalent to the VSD-1000) and the necessary increased torque values for the boss-to-fitting joint.

Completion of the statistical testing devised for the M-1 Program is recommended to gain information for comparison with the various types of connections. The results of this testing could serve as a basis for new specifications and standards for the welded, brazed and swaged connections, as well as being used as a basis for improving the threaded connections.

The development of standardized test procedures and test methods is also recommended. It was expected that much of the necessary information for standardization of test procedures and test methods could be obtained from the statistical testing plan. Connections could be separated as to pressure rating, system fluid, and other factors. They could also be qualified with the standardized procedures and test methods. If such a standardized program were initiated, a more realistic comparison between testing could be made from tests conducted by different evaluators.

V. DYNAMIC SEALS

A. TECHNICAL DISCUSSION

1. Basic Requirements for a Dynamic Seal

Ideally, the perfect dynamic seal would be obtained in the same manner as the static seal (complete plastic flow or deformation between the seal and mating surfaces) except that it would also be necessary to maintain this condition with relative motion between the seal and mating surfaces. Obviously, the only way possible to achieve this condition would be to have the mating surfaces absolutely smooth with no irregularities (an absolute zero PTV surface finish). Otherwise, the interface contact surfaces would shear at the start of motion between the surfaces and the seal would be lost. Methods for sealing one type of relative motion between parts (reciprocating) are available that will provide the lowest leakage rates and not require the intimate rubbing contact between the parts. These are the bellows that are welded at each end to one of the parts, or the flexible plastic or elastomeric diaphragms. However, in most cases, a satisfactory dynamic joint for a particular application is the result of a series of compromises between permissible (and desired) leakage rates, mating surface finishes, flatness, roundness, hardness, seal materials, and many other factors that can affect the life of the seal.

The two basic considerations in the dynamic joint design is the allowable leakage rate and the required operating life of the joint. Initial leakage rate decreases when the sealing stress is increased, but the life of the joint decreases at the same time. Leakage rate and life of the joint is also a function of mating surface finish, rubbing speeds, lubrication, temperatures, pressures, deflection and distortions in the joint, as well as many other factors. These will be discussed subsequently.

2. Components of the Dynamic Seal Joint

The number of components in the dynamic seal joint varies with the type of seal (or joint) selected. Basically, there are three parts to the "contact" joint; the housing, the moving member (which contains the prepared grooves or surfaces for the seal), and the seal. The controlled clearance (or labyrinth) joints have no seals as such. Other types of joints require additional parts or pieces to hold or properly position the seal. All of these components must be considered together in the initial design of the joint. Each element of the design depends upon the others to obtain a satisfactory leakage rate as well as joint life. The seal must be selected based upon the fluid to be sealed, the allowable leakage rate, wear resistance, operating temperatures and pressures, as well as the other environmental conditions that will be encountered in the joint. Then, the mating parts must be designed and constructed in a manner that will provide optimum utilization of the seal. Each design selected has definite machining requirements (i.e., surface finishes,

roundness, flatness, parallelism, and clearances) to provide the desired squeeze, clearances, and other conditions compatible with the associated leakage rate and life of the joint. Each design will have definite limitations as regards the amount of deflection or distortion (which results in such changes as squeeze and clearances, eccentricities, and parallelism) that can be accommodated while maintaining the leakage rate and providing the desired life. All of the components must be designed to meet these requirements under all operating conditions and environments of the joint.

3. Variables in the Dynamic Joint

The number of possible variables and combinations of variables in the dynamic seal joint exceed those possible in the static seal joint. Essentially, the same variables are present in the dynamic joint as for the static joint except that they are increased because the relative motion between parts presents additional factors affecting the operation and storage life of the joint (such as friction, galling, surface speeds, heat outputs, and many other factors). Obviously, it would be impractical to attempt a discussion of all these possibilities. However, some of the most common designer considerations are presented on Table XVII.

4. Types of Dynamic Seals

There are two basic types of dynamic seals; diametral seals and face seals. Examples of these two types as well as a combination of the face and diametral seals are shown on Figures No. 112 and No. 113. As the names imply, the diametral seal seals between two diameters and the face seal seals between two faces. Within these two types are subtypes which are classified according to their use, design, or application. The diametral seal can be either a "contact" (definite contact between mating surfaces) or "no-contact" (i.e., controlled clearance, labyrinth, and slingers) seal and its application can be for rotary, reciprocating, or oscillating devices. Face seals can also be "contact" or "no-contact" types and they can be applied to rotary or oscillating devices.

5. Uses, Advantages, and Disadvantages of the Various Dynamic Seal Types

Each of the subtypes of dynamic seals are inventions for specific applications. The controlled clearance and labyrinth designs are very useful in control valves for liquid (i.e., lubricating oils and hydraulic fluid) systems and provide very low friction, long-life, and reasonable leakage rates. The materials and hardness of the mating parts must be selected to prevent seizing or galling and clearances are extremely critical. Usually, the mating parts are lapped-in pairs for the more critical applications. The labyrinth (as well as other directed or controlled flow seals) and the "hard-faced" face seals are the only dynamic seals available for use in extremely high speed turbomachinery. Again, clearances are highly important for the directed or

TABLE XVII

VARIABLES AND THEIR EFFECT UPON THE DYNAMIC SEAL JOINT

VARIABLE	EFFECT
A. SEAL	
Configuration	Groove machining; installation, handling, and storage procedures and requirements; friction; leakage rate; wear rates; deflection capabilities
Material	Compatibility, permeability, machining, leakage rate, friction, required sealing stresses, mating surface preparation, permissible operating temperature, seizing or galling, corrosion, lubrication
Material Modulus and Strength	Permissible pressures, required sealing stresses, deflection capabilities clearances, parallelism, permissible temperature
Material Flexural Strength	Permissible pressures and temperatures, deflection capabilities, wear rates
Material Creep, Cold Flow, Compression Set Characteristics	Leakage rates, storage life, wear rate, required machining, initial sealing stresses, friction, deflection capability
Seal Surface Finish	Leakage rates, required sealing stresses (squeeze), wear rates, permissible surface speeds, seizing or galling, mating surface preparation, friction
Roundness, Concentricity, Flatness	Leakage rates, unequal sealing stresses (squeeze), wear rate, permissible surface speeds
Coatings	Compatibility, leakage rates, required sealing stresses, wear rate, permissible surface speeds, seizing or galling, friction, mating surface preparation
Material Contraction Characteristics	Sealing stresses, friction, wear rates, leakage rates
B. HOUSINGS AND SHAFTS	
Material	Compatibility, electrolytic corrosion, permeability (castings, forgings, and raw stock), friction, seizing or galling, permissible operating temperature, lubrication, wear rate
Material Hardness	Friction, seizing or galling, wear rates
Material Modulus and Strength	Deflection rates, clearances, parallelism, squeeze (sealing stresses), permissible pressure and loads, permissible temperature
Material Creep	Squeeze (sealing stresses), leakage rate, wear rate, friction clearances, concentricities, parallelism
Material Contraction Characteristics	Squeeze (sealing stresses), leakage rate, wear rate, friction, clearances, concentricities, parallelism
Groove Surface Finish	Required sealing stresses (squeeze), leakage rate, wear rate, friction
Groove Roundness, Concentricity, Flatness	Unequal sealing stresses (squeeze), leakage rate, wear rate, clearances
Groove Surface Coatings	Leakage rate, required sealing stresses, wear rate, seizing or galling, friction, fluid compatibility, corrosion
Surface Particle Contaminants	Leakage rate, wear rate, friction, seizing or galling
C. ENVIRONMENTS	
System Fluids	Gasket material properties, sealing stresses, leakage rate, friction, wear rate, surface finishes (corrosion), deflection capabilities of gasket
System Pressure	Sealing stresses, leakage rate, deflection in joint, friction, wear rate, clearances (seizing and galling)
Fluid Temperature	Sealing stresses (differential contraction), material creep and cold flow, friction, wear rate, permissible pressure (material strength), deflection capabilities, clearances (seizing and galling), leakage rate
External Loads	Joint deflection, sealing stresses, seizing and galling, wear rate, leakage rate
Vibration	Sealing stresses, leakage rate, seal material properties (fluctuation in stresses), clearance (seizing and galling)

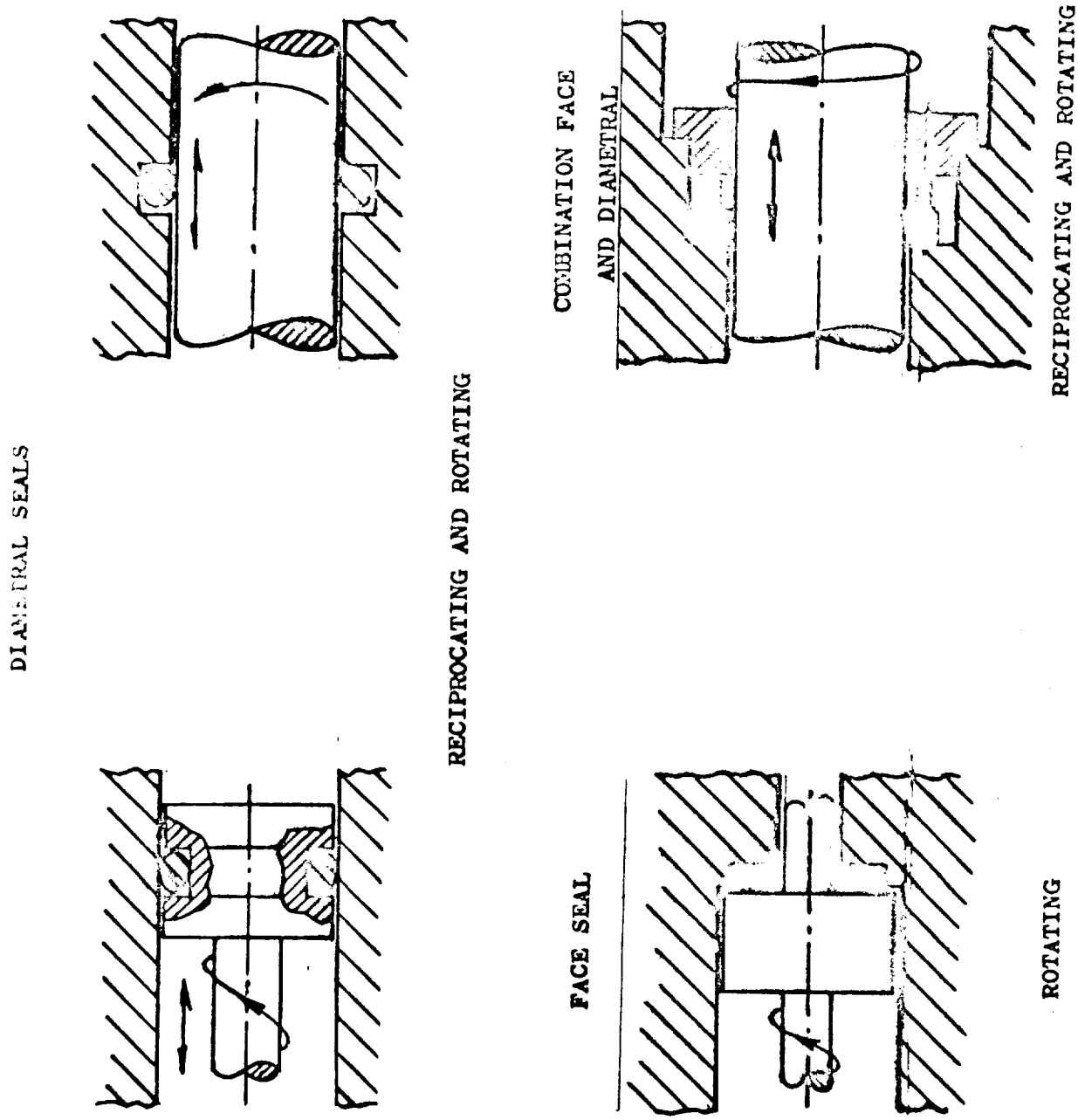
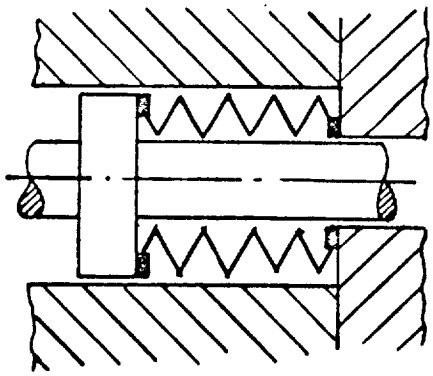


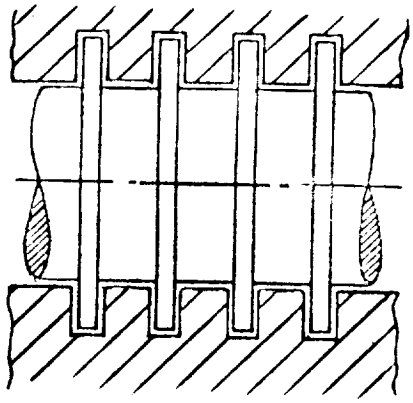
Figure 112
Dynamic "Contact" Seal Joint Configuration

BELLOWS



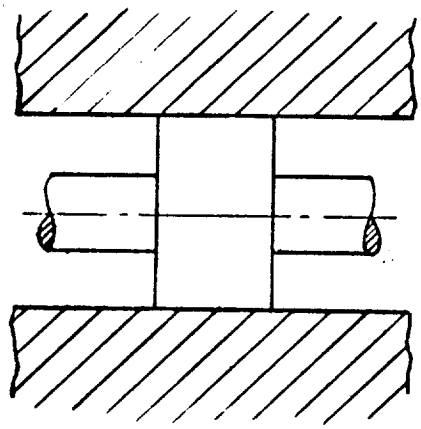
RECIPROCATING

LABYRINTH



ROTATING

CONTROLLED CLEARANCE



RECIPROCATING

Figure 113
Dynamic "No Contact" Seal Joint Configuration

controlled flow types. Materials that prevent galling must be selected for the "hard-faced" face seals. Also, flatness and finish of the mating surfaces are extremely critical because face pressure variations can become serious problems. The stiffness or flexibility of the loading components along with the mating material can cause "chatter" between the faces at various rubbing speeds.

The discussions in this report are limited to the low speed "contact" dynamic seal used in valves and other low speed devices primarily intended for reciprocating shafts and pistons. Where rubbing speeds are relatively low, much lower leakage rates can be obtained with the "contact" dynamic seals. The initial leakage rate is a function of the interface sealing stresses (squeeze or interference), mating surface finish, and relative alignment of the two parts. However, the higher the sealing stresses, the shorter the life of the joint becomes because the seal will wear more rapidly. Also, the higher the seal loading, the greater is the friction. As a result, the forces required to start and maintain motion increase. Mating surface finish affects friction, life, and the sealing stresses required to obtain the desired leakage rate. Thus, the problem becomes one of many compromises between material selection, mating surface finishes, machining requirements, sealing stresses (squeeze), friction, as well as many other factors that affect leakage rates and seal life.

6. Engineering Evaluation

a. Capabilities

Before starting a joint design for a particular application, the various types of seals must be evaluated with respect to the application. It is necessary to know the available materials and coatings, the machining requirements, service temperatures, resistance to the fluid being sealed, required alignment between parts, and many other variables. Information of this type was collected for many different types of dynamic seals and is presented on Figure No. 114. Because this was only a preliminary review, no special grouping was attempted. The information presented was gained from available literature, discussions with the suppliers, and catalog information. The information, while somewhat limited, was used to select the types of gaskets used and tested. It was planned to conduct evaluation tests with many other types to gain additional design information, but the program ended before this could be accomplished.

b. Design Practices

The forces tending to separate or cause leakage and affect the operating life of the dynamic joint are proof (or burst) pressures and both external dynamic and static load input (including tension, compression, bending, and shear load).

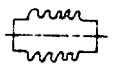
MANUFACTURER	SEAL CONFIGURATION	SEAL MATERIAL	APPROX. COST 1/2" DIA / 4" DIA.	RECOMMENDED SURFACE FINISH	GLAND CONFIGURATION STD. O-RING	GLAND CONFIGURATION SPECIAL	MAXIMUM OPERATING PRESSURE PSIG	OPERATING TEMPERATURE RANGE °F	UNI-DIRECTIONAL SEALING	APPLICABLE TO BOTH INTERNAL & EXTERNAL USE	SEAL FRICTION LOAD LBS/CIRCIN STATIC	SEAL FRICTION LOAD LBS/CIRCIN DYNAMIC	MAX. ALLOWABLE SURFACE SPEED FT/MIN AXIAL	MAX. ALLOWABLE SURFACE SPEED FT/MIN ROTATIONAL	REMARKS
Del Mfg.	Del Seal	Teflon Teflon Coated SS	$4.75 to $21.50 $18.70 to $67.40	8 32		Yes	10,000	-320 to 500	No	Yes					-
Belfab Corp.	Bellows Seal 	Metal SS Inconel Monel etc.													
Wiggins Oil Tool Co.	Bar-X Seal	Inconel X		5			11,000 (Hydro)		Yes	Yes					
Rudolph Krueger Co.	(Delta) Seal	Teflon Nylon Kel-F Lead, Gold Silver	50 in qty 200	8/16	No	Yes	4000	-280°F +500°F	No	Yes		5.1 at -320°F	3000 fpm		2 1/2" dia. max. size to date 8/62 (500 to 800 std cc/min @ 1200 psi OH_2) and -320°F
Gits Bros. Mfg. Co.	Ring Seal	Metal Inconel X Stainless Steel		28-32 for static application 2 for dynamic			20,000 Static Test	-450°F +1500°F	No			5.5* at 0 psi	40*	40*	* N Based upon test data.
Cleveland Graphite Bronze		Non-Elastomeric		x 8 rms XX 4 rms			X 1200 XX1500 For Gas X1250 XX2000 For Liquid	X-423° to +1600°F XX-423° to +500°F	Yes for types "E" & "H" No for Type "F"					X 36000 XX30000 Max Run to date	X Shaft Seals XX Face seals
John L. Dore Co.		Fluoro-green E-600		16					No	Yes	0.070 to 0.095	0.070 to 0.095	800 at 160°F	800 at 140°F	
Mace Corp.	T Ring	25% glass filled teflon					10,000	-450°F to +500°F	No	Yes					

Figure 114
(Sheet 1 of 3)
Dynamic Seal Review

MANUFACTURER	SEAL CONFIGURATION	SEAL MATERIAL	APPROX COST 1/2" DIA. / 4" DIA.	RECOMMENDED SURFACE FINISH	GLAND CONFIGURATION STD. O-RING	GLAND CONFIGURATION SPECIAL	MAXIMUM OPERATING PRESSURE PSIG	OPERATING TEMPERATURE RANGE °F	UNI-DIRECTIONAL SEALING	APPLICABLE TO BOTH INTERNAL & EXTERNAL USE	SEAL FRICTION LOAD LBS/CIRC IN STATIC	SEAL FRICTION LOAD LBS/CIRC IN DYNAMIC	MAX ALLOWABLE SURFACE SPEED FT/MIN AXIAL	MAX ALLOWABLE SURFACE SPEED FT/MIN ROTATIONAL	REMARKS
Reid Enterprises, Inc.	Omniseal	Teflon* Jacketed Stainless Spring		16	Yes			-450 to 500	Yes	Yes	70 to 120 lbs per circumference inch		600	600	
Creavey Seal Co. And Haskell Seal Co.	Astra-Seal Creavey Seal	Teflon over SS with internal filler		63	Yes		5,000	-450 to 500	No	Yes					Not recommended for reciprocating applications
Hydrodyne Corp.	Skinner	SS Bellows* Teflon Wedge				Yes	4,000	-65 to 1200	Yes	Yes					
Balsells Engr. Co.	Bal Seal			16	Yes		10,000	-450 to 400	Yes	Yes					
Garlock, Inc.	Slotted C-Ring	Teflon			Yes			-450 to 500	Yes	Yes					
Allegheny Plastics	X Seal	Teflon*						-450 to 500	NO	Yes					
Crane Packing Co.	U Cup	Teflon		15			1,500		Yes	Yes					
AGC	Hat Seal AS1023-	Kel-F		4		Yes	1,000	-320 to 165	Yes	No			175	175	Information based on AGC proven application;
* Denotes other materials available.															* P = Bearing load in psi on projected area X surface speed in feet per minute.

Figure 114
(Sheet 2 of 3)
Dynamic Seal Review

MANUFACTURER	SEAL CONFIGURATION	SEAL MATERIAL	APPROX. COST 1/2" DIA / 4" DIA.	RECOMMENDED SURFACE FINISH	GLAND CONFIGURATION: STD. O-RING	GLAND CONFIGURATION: SPECIAL	MAXIMUM OPERATING PRESSURE PSIG	OPERATING TEMPERATURE RANGE °F	UNI-DIRECTIONAL SEALING	APPLICABLE TO BOTH INTERNAL & EXTERNAL USE	SEAL FRICTION LOAD LBS/CIRC.IN.: STATIC	SEAL FRICTION LOAD LBS/CIRC.IN.: DYNAMIC	MAX ALLOWABLE SURFACE SPEED FT/MIN.: AXIAL	MAX ALLOWABLE SURFACE SPEED FT/MIN.: ROTATIONAL	REMARKS
Crane Packing Co.	V Ring	Teflon Asbestos					750	-100°F to +650°F	Yes						
National Seal Division of Federal-Mogul-Bower Bearings, Inc.	Bud Oil Seal	Synthetic Rubber		10 to 25			5 - 10 psi	-80°F to +400°F	Yes	No					
National Carbon Co.	Molded/Machined (Config. Optional)	Nat. Carb. Co., Grade CCA-72	3" $550 in 500 lot	16		Yes		-269 d_o + 160	Yes	Yes			*		Friction factor of .05 to .15 *Limited by PV factor of 20,000 to 50,000 for non-lubricated operation.
Pure Carbon Co.	Molded/Machined (Config. optional)	Carbon/ Graphite		10 rms		Yes	1500	-0 to 1100°F	Yes	Yes					PV factor of 15,000 dry to 250,000 lubricated. Attacked by strong oxidizing agents, H_2O_2, Fluoride, HNO_3, etc. Compatible to radiation environment.

Figure 114
(Sheet 3 of 3)
Dynamic Seal Review

Usually, the joint is subjected to a static differential pressure above the maximum expected pressure to demonstrate that the joint can withstand the maximum expected differential pressure across the joint at operating temperatures. Factors are included to compensate for the various material properties at operating temperatures. In addition, the housing, valve body, or other parts of the component can be expected to take system proof pressure without detrimental distortions or effects upon the joint. All these forces must be known to properly design the joint. The system proof pressures and proof differential pressures will depend upon the safety factors desired for the particular application, but are usually the same as those indicated in the Static Seal discussion. Generally, the dynamic joint is not expected to be operable after the application of burst pressure. However, if the ultimate-to-yield strength ratio of any of the materials of the joint is less than the ratio of design burst-to-proof pressures, the burst pressure can become one of the real design loads rather than differential proof pressure. All external loads must also be scrutinized to determine how they are affected by proof pressure, operating temperatures and other environments. Compression (or column) loads, tension loads, bending loads, and shear loads on shafts, pistons, and other running gear will be extremely important factors in designing for minimum leakage and best life. Errors and miscalculations in any one of these design loads can make the difference between a satisfactory and an unsatisfactory joint design.

Some dynamic seal designs can accept differential pressures in both directions across the joint (called bi-directional in this report) while other types of dynamic seals or joints can take differential pressures in only one direction across the joint (called uni-directional in this report). Some design data and methods for eliminating problems with dynamic O-rings and other squeeze type packing are available in the existing literature(39)(40). Some information is also available for other types of packings in the existing literature(41)(42)(43). However, no single reference is available that provides methods for analysis or the effects of the many variables upon seal life and leakage rate for all types of dynamic seals or joints. The most complete coverage of the subject found by the author to date is contained in three

(39) Everett, M. H. and Gillette, H. G., "Squeeze-Type" Packings, Product Engineering Seals Book, Precision Rubber Products, June 11, 1964
(40) Catalog No. 5700, Rubber Products Division, Parker Appliance Co.
(41) Main, M. M. and McKillop, G. R., Simple Compression Packings, Product Engineering Seals Book, Crane Packing Co., June 11, 1964
(42) Smith, J. N., Lip-Type Packings, Product Engineering Seals Book, Rubber and Leather Products, June 11, 1964
(43) Shepler, P. R., Split-Ring Seals, Product Engineering Seals Book, Metals Products Division, Koppers Co., Inc., June 11, 1964

reports[44][45][46]. These documents constitute a literature survey of the subject of dynamic seals along with some preliminary work for design analysis of some types of dynamic seals.

The basic variables affecting "contact" seal life and leakage are sealing stresses (load, squeeze, or interference), surface (or rubbing) speed, surface finish (and lay), and lubrication (or lubrication qualities) between the two surfaces. Once these factors have been determined to gain the best leakage rate and seal life, the problem becomes one of determining how the many other variables affect these four basic variables. In one example, changes in temperature will cause differential contraction between the parts of the joint and an associated change in sealing stresses which can cause galling (over-stressing and resultant change in surface finish) or separation at the sealing interface and a change in the leakage rate and wear rate (life). Another example is the increase or decrease in the mating surface finish which can cause either an increase or decrease in seal life as well as either an increase or decrease in the leakage rate. Finally, consider that operating pressures directly affect sealing stresses and therefore, seal life and leakage.

If the proprietary seals are selected, the supplier can provide some information useful to the designer in his preliminary design work. However, this information is usually not sufficient for making the design final. The best method for making any of the designs final with existing technology and analysis is to conduct development and evaluation tests of the selected joint configuration. These development and evaluation tests should include the variables of machining, design loads, temperatures, pressures, and other conditions. Statistical testing methods are a definite aid in gaining the most information from a minimum amount of testing.

The deflection capabilities of the seal selected must also be investigated and determined. Types of deflections that can be encountered in the joint are shown on Figure No. 115. These capabilities also affect the design of the other parts of the joint. Some information is available from several suppliers of the proprietary designs, but usually these capabilities must be determined by actual testing. Also, these capabilities can be limited by extrusion characteristics, material modulus, or other factors and will determine the permissible change in clearance, interference, squeeze, or other design factors for the mating parts.

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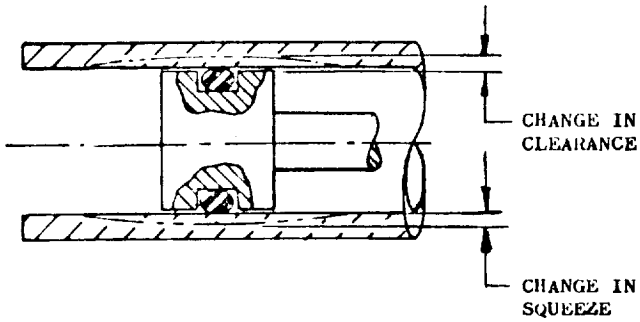
(44) Study of Dynamic and Static Seals for Liquid Rocket Engines, Final Report NASA Contract NAS 7-102 (February 1962 to February 1963), Volumes 1 (CR 50660), 2 (CR 50661), 3A (CR 50662), and 3B (CR 50663), Advanced Technology Laboratories, General Electric Co.

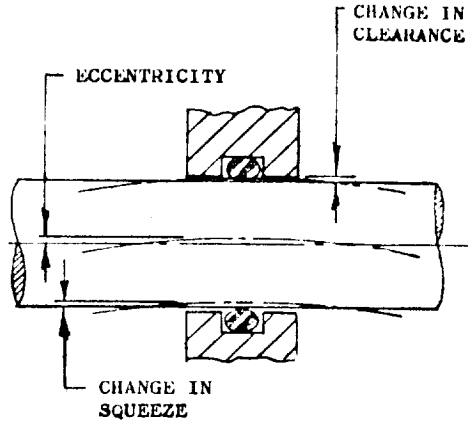
(45) Study of Dynamic and Static Seals for Liquid Rocket Engines Final Report NASA Contract NAS 7-102 (February to November 1963), Volumes 1, 2, and 3, Advanced Technology Laboratories, General Electric Co.

(46) Study of Dynamic and Static Seals for Liquid Rocket Engines, Final Report NASA Contract 7-102 (April to September 1964), Advanced Technology Laboratories, General Electric Co.



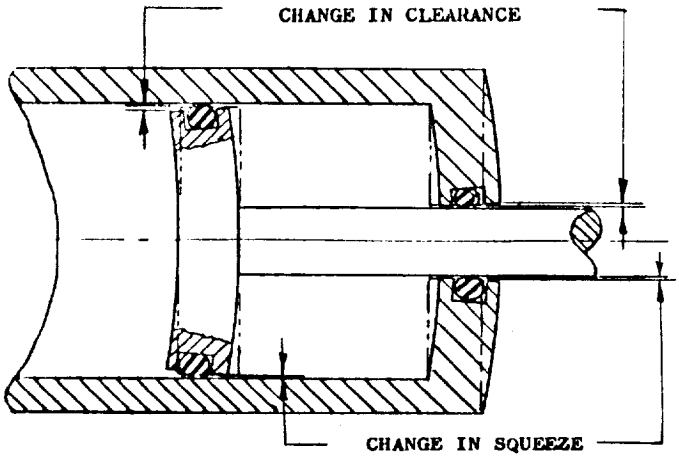
DIFFERENTIAL RADIAL

(Hoop expansion or contraction of housing or piston)

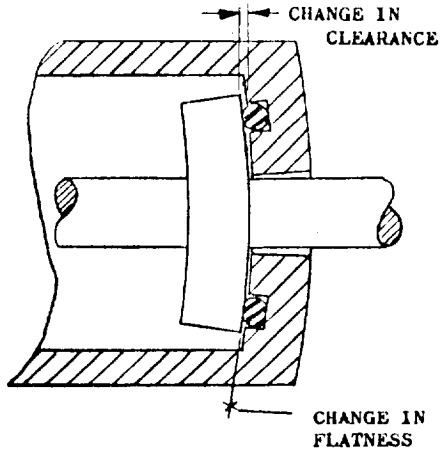


SHAFT BENDING OR WHIP

(Bending, shear, or column loading of reciprocating or rotating shafts and "whip" in high speed rotating shafts)



PISTON AND HOUSING



SHOULDER AND HOUSING

BENDING IN COMPONENTS

Figure 115
Types of Deflections in Dynamic Joints

7. Joint Design

The approach in designing a dynamic joint can be separated into three basic elements: seal selection; preparation of the mating surfaces (or grooves); and the shaft, housing (or body), or other necessary component design. These elements are strongly interrelated, but the approach provides a somewhat logical step-by-step method for preparing the preliminary design (or designs) for evaluation, development, and final analysis. This approach is used in the ensuing discussion.

a. Seal Selection

The first consideration in selecting the seal materials and configuration is the fluids to be sealed and the leakage rate. The effects of fluid exposure upon the seal material include all those presented for the gasket selection in the Static Seals discussion. In addition, the lubricating qualities (or lack of lubricating qualities) of the system fluid must be determined. If lubrication with other than system fluids is necessary, the lubricant must not only be compatible (i.e., not cause violent reactions or degrade when exposed) with the system fluids, system cleaning fluids, and materials, but it must remain essentially of the same consistency as the unexposed lubricant. Lubricants can harden or form hard globules when exposed to system fluids (or system cleaning fluids) and/or temperatures. As a result, they can present the same problems of scoring the contact surfaces as found with any other particle contamination.

Cold flow, creep, compression set, contraction, extrusion characteristics, elongation, and other material properties of the selected seal material at operating temperatures must be investigated as these properties can affect the sealing stresses at the sealing interface. They can cause complete failure of the seal material because of mechanical shock, vibration, and other environments to which the joint may be subjected.

Wear rates as well as static and dynamic friction between the material at the sealing stresses and surface speeds expected at both ambient temperature (for simple functional tests) and operating conditions must be investigated. Sealing stresses can either increase or decrease with changes in temperature causing a change in wear rates. Friction must be determined at both temperatures and the larger of these must be included in the loads required to initiate motion and/or obtain the desired surface speed.

b. Groove or Surface Machining Requirements

After the seal configuration and materials have been selected, the machining requirements for the mating parts must be determined. The method of holding (or providing proper sealing stresses) determines the machining requirements and space required for the mating parts. Groove shape (i.e., depth and width) provides (or affects) the squeeze necessary to gain

the initial sealing stresses. Eccentricities, out-of-roundness, or waviness will cause unequal sealing stresses and affect leakage rates and wear rates (seal life). Surface finishes affect the required initial sealing stresses (to obtain the desired leakage rate) and friction. This, in turn, affects wear rates (seal life) and the forces needed to initiate motion and obtain the desired operating time or surface speeds.

Machining requirements also affect the allowable (or permissible) deflections and distortions that the seal or joint can withstand without affecting leakage rates and seal life. These should be thoroughly investigated to assure that all the requirements are specified in the detailed drawings of each part of the joint.

c. Housing, Shaft, and Other Component Designs

The housing (or body), the shaft, and any other necessary components of the joint must be designed and constructed so as not to exceed the allowable deflection of the seals at all operating loads and conditions. Differential radial deflections at the joints caused by proof or burst pressures or operating temperatures can cause permanent seal distortion, changes in sealing stresses (squeeze or interference), and can affect leakage rates and seal life. Bending or "whip" in the shafts or pistons will cause increased sealing stress and wear, or lead to galling of the sealing surfaces with eventual joint failure.

8. Typical Design Problem for Aerospace Service

Good design of the dynamic seal joint is more difficult than the static seal joint because the movements between members increases the problems in attaining a good seal and maintaining the seal for the desired life of the joint. Also, methods of analysis are more difficult and not as refined as they are for the static seal joints. Good design for dynamic joints requires as good a preliminary analysis as possible, including complete attention to the detailed machining requirements and tolerances. Then, complete development testing must be conducted to gain insights into improved analysis methods as well as improvements in the joint before making the design final. The basic purpose of this discussion is to present one approach to preliminary design and the desired data from testing before making the final design.

The sample problem is based upon an assumed sliding shaft seal joint for a valve to be used in liquid oxygen. The known data and requirements for the joint are as follows:

a. System Fluid: Liquid Oxygen

b. Temperature: +120°F to -300°F

c. Pressures: Nominal Operating = 1500 psig
Maximum Operating = 1850 psig
Proof = 2250 psig
Burst = 3375 psig

d. Type of Operation: Reciprocating Shaft

e. Shaft Size: 2.00-in. Outside Diameter

f. Materials: Shaft = 347 CRES with a maximum operating speed of 50-in./sec
Valve Body = 6061-T6 Aluminum

g. Maximum Permissible Leakage Rate: 1.0×10^{-4} cc/sec/in. circumference

h. Cycle Life: 2000 cycles before exceeding permissible leakage rate

i. A secondary (or dual) seal arrangement is also required for the joint for two purposes; to prove that the primary seal meets the leakage requirements and to provide a collector for leakage past the primary seal. The inter-seal cavity is vented overboard to a safe area.

The seal should be selected based upon its capability to meet the maximum permissible leakage requirements (including permeability) over the required cycle life, compatibility with the system fluids, temperature effect upon the material(s), and compatibility with the mating surface materials (including such considerations as galvanic corrosion, seal/mating surface hardness relationships, and resistance to galling with mating rubbing surface).

Currently, this selection is difficult because insufficient data exist about leakage capabilities, wear rates (life), as well as other necessary information for most seal designs. Considerable preliminary development testing is necessary to acquire this information. Then, the surrounding or mating members must be designed to meet all the requirements for the seal that is finally selected. In this case, the seals selected are the "Omniseal" shaft seal designs. The Omniseal is a "pressure-energized," spring-loaded, Teflon-jacketed seal. The materials are compatible with the system fluids and experience has shown that they are capable of satisfying the leakage and cycle life requirements if the joint is properly designed.

Leakage from permeation of the system fluids through the Teflon jacket can be estimated from gaseous oxygen permeation rates as follows:

$$Q_p = K_p \frac{A_e P}{1.296 \times 10^{11} t_j}$$

(See Appendix A for a definition of all Symbols used.)

Assuming that the nominal 1/8-in. cross-section "Omniseals" are used:

$A_e \cong 1.75\text{-in.}^2$

$t_j = 0.020\text{-in.}$

$P = 2250$ psig

Kp at -250°F = 0.07 for TFE (see Figure No. 4). This value is selected at -250°F, which is the limit for Figure No. 4. It is assumed that the value at -300°F would be less.

Then:

$$Q_p = \frac{0.07(1.75)\ (2250)}{1.296 \times 10''}$$

$$= \underline{\underline{2.127 \times 10^{-8}\ \text{cc/sec.}}}$$

The allowable leakage $= 1.0 \times 10^{-4} \times 2\pi$

$$= \underline{\underline{6.3 \times 10^{-4}\ \text{cc/sec}}}$$

However, if it is required to meet the leakage requirements at ambient temperatures with gaseous oxygen, the permeation factor changes to 500 cc/100-in.^{2}/mil/atm-days and leakage from permeation would be approximately:

$$Q_p = \frac{500\ (1.75)\ (2250)}{1.296 \times 10''\ (0.020)}$$

$$= \underline{\underline{7.6 \times 10^{-4}\ \text{cc/sec}}}$$

As the temperature decreases to operating conditions, the Teflon jacket, the shaft, and the valve body contracts. The change in diameters is calculated from:

$$\Delta D_T = K_T\ D_{68}$$

The values for K_T in he following calculations are from Figure No. 18 and existing literature(47).

(47) Cryogenic Materials Data Handbook, op. cit.

The change in the shaft diameter would be:

$$\Delta D_T \text{ (shaft)} = 2.0\ (-2.8 \times 10^{-3}) = -5.6 \times 10^{-3}$$

$$= \underline{\underline{0.0056\text{-in. contraction}}}$$

The change in diameter of the groove in the valve body would be:

$$\Delta D_T \text{ (groove)} = 2.25\ (-3.8 \times 10^{-3}) = 8.55 \times 10^{-2}$$

$$= \underline{\underline{0.00855\text{-in. contraction}}}$$

The unsupported change in the gasket inside diameter and outside diameter would be:

$$\Delta D_T \text{ (gasket I.D.)} = 2.0\ (-1.8 \times 10^{-2}) = -3.6 \times 10^{-2}$$

$$= \underline{\underline{0.036\text{-in. contraction}}}$$

$$\Delta D_T \text{ (gasket O.D.)} = 2.25\ (-1.8 \times 10^{-2}) = -4.05 \times 10^{-2}$$

$$= \underline{\underline{0.0405\text{-in. contraction}}}$$

The body contracts 0.00295-in. or 0.003-in. more than the shaft and will change running clearances between these parts which must be accounted for in the body clearance diameter. (Care must be used in specifying the clearance diameter because if this clearance becomes too large, the jacket can extrude between the diameters at operating pressure.) The seal tends to contract into the shaft (increasing sealing stresses and wear rate) and away from the body. The seal spring preload and the shaft provides a resistance to contraction. However, hoop tension stresses do increase in the jacket and can cause failure under certain conditions (see Section III,C,1,h).

Changes in diameters and clearances caused by pressure must be determined. This is a particular problem with thin-walled actuators and flat-plate actuator caps or ends where the shaft feeds through the actuator. These deflections must be limited to prevent extrusion of the seal, metal-to-metal rubbing with possible galling of the sealing surfaces, as well as excessive changes in squeeze (sealing stresses). Deflections of the flat plate ends can be estimated from the maximum bending stress and deflection at the gasket sealing edge equations in Section III,A,7,d and deflections in a thin-walled actuator can be estimated from the hoop stress formulae:

$$f_t = \frac{PD_i}{2t} \quad \text{and} \quad \Delta D_p = \frac{f_t\ D_i}{E}$$

$$\text{or} \quad \Delta D_p = \frac{PD_i^2}{2tE}$$

(See Appendix A for a definition of all Symbols used.)

After the layout is completed, a more detailed analysis must be made. Also, the detailed machining dimensions for the grooves and shaft must be determined and specified on the drawings.

From this point, it is a matter of extensive development testing to determine the actual conditions at the joint and to gain additional data for improving the methods of analysis.

9. Installation and Handling Procedures and Problems

The installation procedures necessary for a particular seal (or joint) design must be considered in the initial design of the joint. Proper design practices for the O-ring installation are presented in Military Specifications[48] and other literature[49][50]. Requirements for the design and proper installation of joints using proprietary seals must be evolved in conjunction with the supplier. Many of the proprietary seal designs require proper design of the mating parts as well as special installation tools, equipment, or procedures to prevent distorting or otherwise damaging the seal or sealing surfaces during installation. These must be made available to assembly personnel along with instructions for their proper use. Cleanliness of the seal and other mating parts is always extremely important in obtaining the initial seal at assembly and providing satisfactory seal life.

Proper handling of the dynamic seals is at least as critical as handling of the static gaskets. In some cases, dynamic seals are even more susceptible to damage during the various stages of inspection, cleaning, moving, and storage. The same methods recommended for the proper cleaning, packaging, and handling of static seals are also recommended for the dynamic seals. In addition, it is suggested that any lubricant used be mutually agreed upon with the supplier and applied to the seal before packaging. The effect of moisture and storage upon the lubricant can be minimized by slightly evacuating the plastic bags used for packaging of the pre-cleaned seals before the bag is heat sealed. The reduction of problems in proper lubrication (and the effect upon seal installation and possible seal life) would be well worth the effort and initial cost of the operation.

(48) MIL-P-5514, Packings, Installation and Gland Design, Hydraulic, General Requirements for
(49) Everett, M. H. and Gillette, H. G., op. cit.
(50) Catalog No. 5700, op. cit.

B. TEST PROCEDURES

A thorough dynamic seal test program was planned during the M-1 Engine Program. This testing was planned for many types of reciprocating seal designs using a test cell designed specifically for the purpose of developing and evaluating seals in a size range from .50-in to 8.0-in. diamter. Tests were to include variances in pressure, temperature, surface speed, friction (static and dynamic), leakage rates, squeeze or interference (sealing stresses), life cycling, as well as many combinations of these variables. The basic purpose of these tests was to determine the best configurations for a wide variety of services as applied to the M-1 engine. Also, the test program was designed statistically in such a manner so as to determine the most reliable conditions for each type of seal. However, these tests were cancelled because of a reduction in the M-1 Program scope and no data were obtained.

Useful data were collected for two types of seals from the testing of some components for the M-1 liquid hydrogen/liquid oxygen engine system. These seal types were the proprietary Omniseal and a group of specially designed lipseals. The components involved were the M-1 gas generator oxidizer (GGOV) and fuel (GGFV) valves as well as the sleeve-type thrust chamber valves (TCV). In addition a test cell was fabricated during the early stages of the thrust chamber valves design and development. This test cell simulated the basic configuration of the component and was used as an aid in determining the best configuration for a particular lipseal design for the actual valve. The basic configuration of the types of seals as well as the component and systems in which they were used are shown on Figure 116. The configuration of the actual components (GGOV, GGFV, and TCV) was shown previously on Figures 77, 78, and 79. The thrust chamber valve simulator test cell is shown schematically on Figure 117. The test procedures used for these tests as well as the component development tests are detailed in the ensuing discussions.

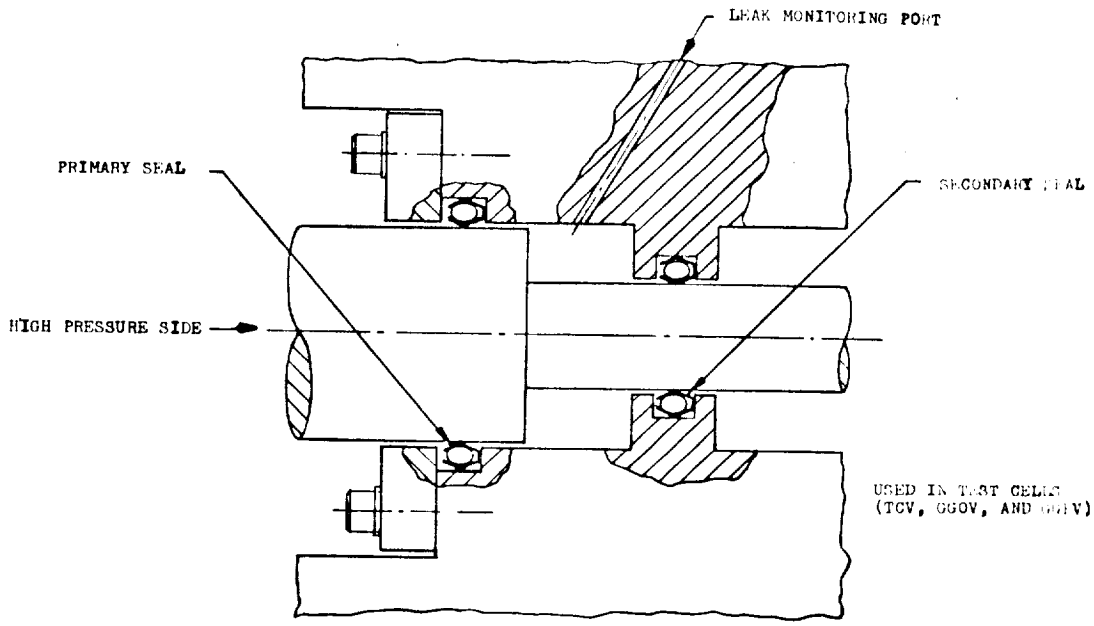
1. Test Cells

a. Proof Test

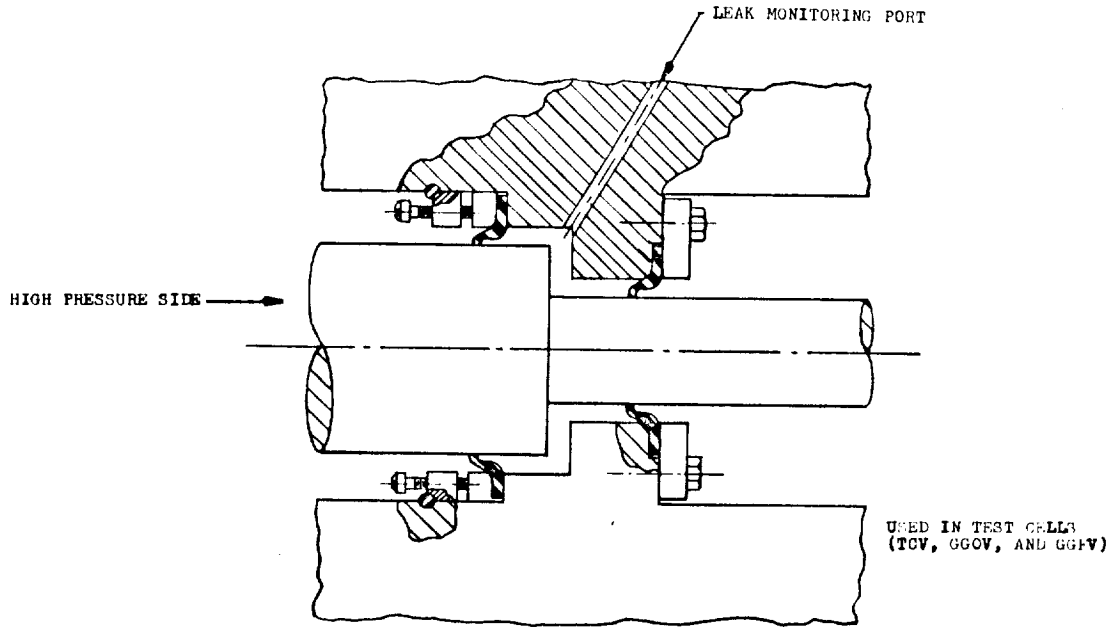
The seals were assembled in the test cells and proof tests were conducted at 1850 psig for three minutes at both ambient and liquid nitrogen temperatures. The test media used was gaseous nitrogen. The purpose of these tests was to assure that the joint would accept proof pressure without failure or detrimental distortion of the seal.

b. Leakage Test

Static leakage tests were conducted at incremental pressures between 0 psig to 1500 psig with the seals assembled in the test cells. The test media used were gaseous nitrogen, liquid nitrogen, and liquid hydrogen. Numerous static leakage tests were conducted with several seal configurations at various conditions of pressure and temperature during functional cycling, during friction tests, and at other types of operation.



OMNISEAL



LIPSEAL

Figure 116
Basic Configurations and Applications
of the Dynamic Seals Tested

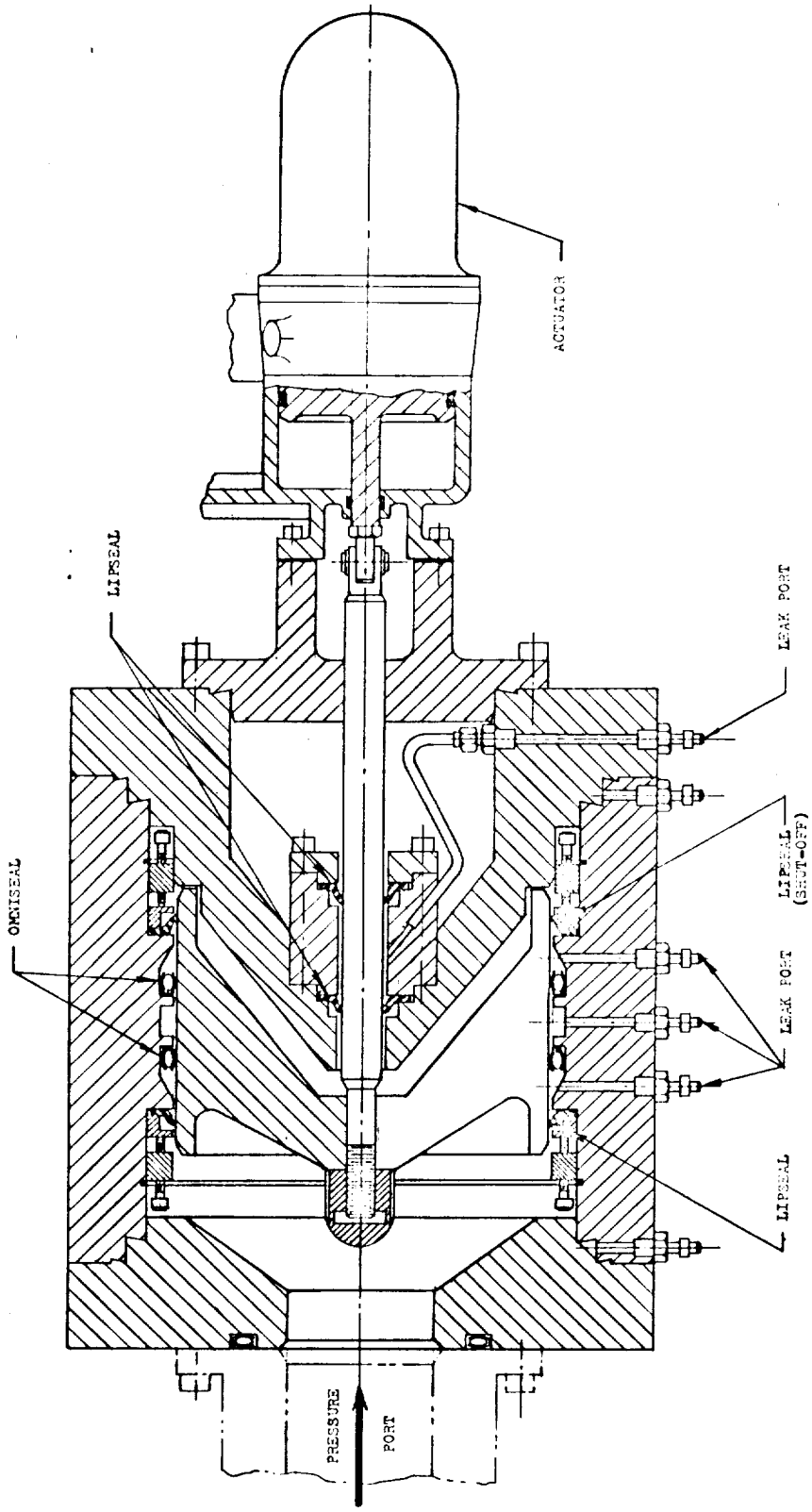


Figure 117
M-1 Thrust Chamber Valve Simulator Test Cell

c. Functional Cycling (Endurance) Tests

The lipseals were installed into the test cells and functionally cycled until failure or excessive leakage occurred. These tests were performed at ambient and liquid nitrogen temperature to gain information about wear rates at both temperatures. Leakage was monitored at various numbers of functional cycles. Cycle rates were adjusted to vary the sliding speeds between 2-in./sec and 10-in./sec. Pressures applied to the lipseal were also varied.

d. Friction Tests

Break-away (static) and sliding (dynamic) friction tests were conducted with some lipseal configurations. A schematic of the set-up for these tests is shown on Figure 118. Tests were conducted at ambient, liquid nitrogen, and liquid hydrogen temperatures. The basic method used was to conduct friction tests with the test cell at ambient and liquid nitrogen temperatures with all of the seals and equipment installed except for the lipseal. Then, the tests were repeated with a particular lipseal configuration installed. Friction for the lipseal was determined from the difference in friction loads both "with" and "without" the lipseal installed.

e. Burst Test

Burst tests were conducted with the large diameter lipseals installed into the test cells. It was not necessary that the component (or the seals) be functional after this test and the basic purpose of these tests was to determine whether the seal would fracture or "blow-out" when subjected to the design burst pressure of 2200 psig at liquid nitrogen temperature. Pressure was slowly applied to the test cell and held for a minimum of three minutes. In some cases, the seals were then subjected to leakage testing before disassembly, but usually the seal was removed and examined for permanent distortions or fracture locations. If a particular seal configuration satisfactorily passed this phase of testing without fracture or failure, a new seal was installed and slowly pressurized at liquid nitrogen temperature until fracture or failure occurred.

2. Components

a. Proof Tests

Initial static proof tests were conducted with the components to prove the ability of the component and various joints to withstand pressures above operating pressures without detrimental distortion or other affects. The magnitudes of proof test pressures were dependent upon the application for the particular component design. Test pressures were gradually applied over a minimum period of three minutes. Usually, no attempt was made to record leakage during these tests.

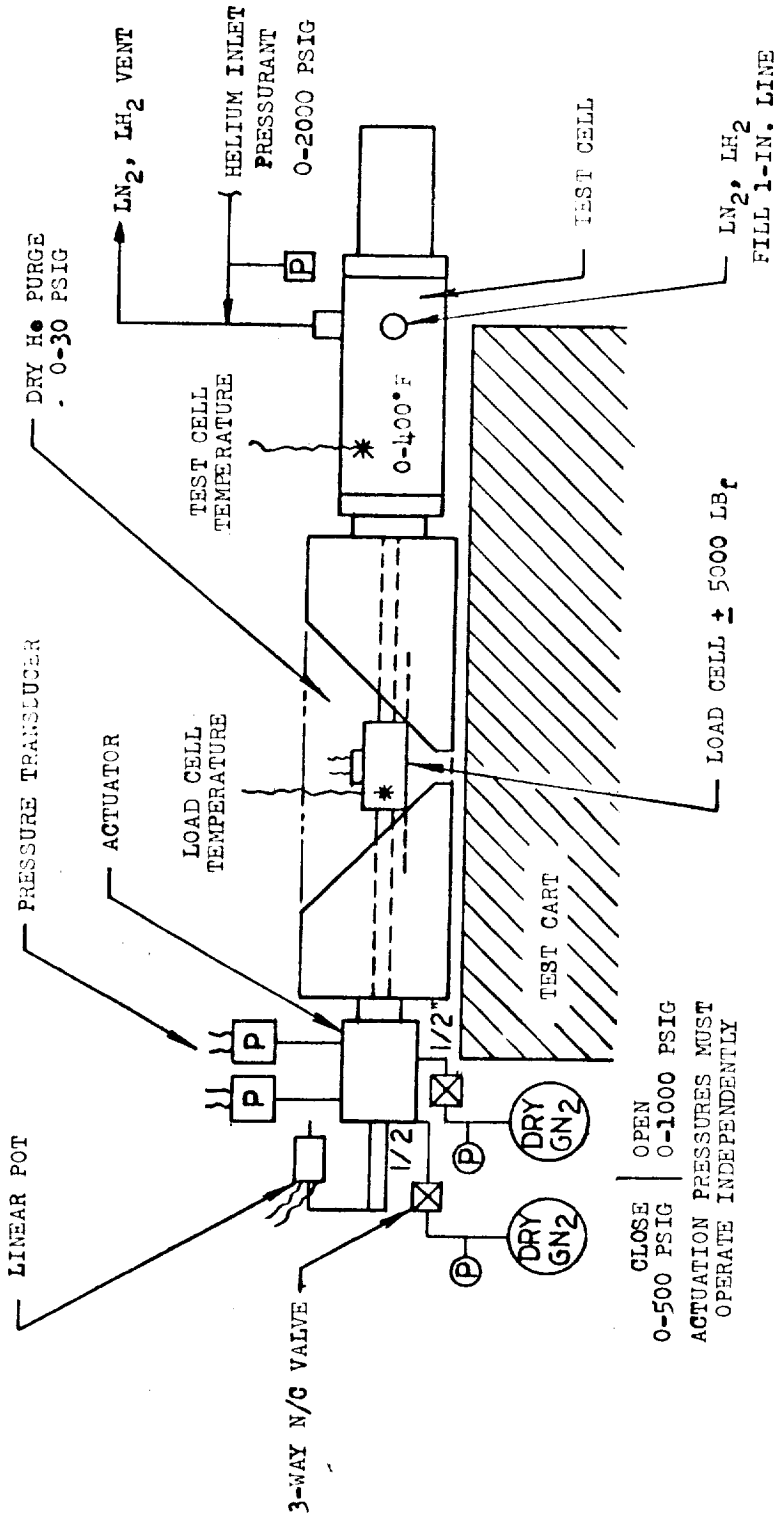


Figure 118
Friction Test Set-Up

b. Leakage Tests

Static leakage tests were initially conducted with the components and at various stages of the component testing and development. The maximum test pressures were again dependent upon the application for the particular component design. Pressure was applied in increments up to the maximum pressure and leakage rates were determined at each incremental pressure.

c. Functional Cycling (Endurance) Tests

Endurance tests were conducted with the component to demonstrate its capabilities in relationship to design requirements. These tests were performed at both ambient and operating temperatures. Leakage tests were conducted at predetermined cycles to gain an insight into the affect of cycling upon leakage rate.

d. Friction Tests

Some simple total seal friction tests were performed with the component to aid in determining actuator sizing, the effects of different loads upon the pressure at which movement starts, and the effects of different loads upon opening and closing times. The thrust chamber valve tests were performed to verify the empirical friction data obtained from the 11.0-in. lipseal test cell. These tests were performed at both ambient and liquid nitrogen temperature.

e. Vibration Test

Vibration tests were conducted using the gas generator valve with the lipseals installed (the selected seal configuration). These tests were performed in accordance with the model specification of the system for which the component was designed. The primary objective of these tests was to determine the effect of expected vibration levels upon the internal mechanism of the component. Leakage tests conducted before, during, and after the test also provided information about the lipseal capability. The vibration tests were conducted at both ambient and liquid nitrogen temperatures as follows:

(1) At ambient temperature, the component was vibrated in each of the three mutually-perpendicular planes with frequencies from 20 cps to 2000 cps at 1g RMS acceleration.

(2) The above test was repeated with the components maintained at liquid nitrogen temperatures.

(3) The component was then subjected to a random endurance vibration test in each of the three mutually-perpendicular planes at liquid nitrogen temperature (-300° $\pm$ 25°F).

f. Hot Firing Test

The gas generator valves were assembled to the mating hardware with the lipseals installed and actual operating tests were conducted with the completed assembly with system operating fluids (liquid hydrogen and liquid oxygen). Eleven tests were conducted.

C. TEST RESULTS

The configurations, part numbers, function, and use of the different seals tested are shown on Table XVIII. Since these tests were completed, the dynamic "Omniseal" design has been modified slightly (new Aeroquip P/N AR10104) to provide more equal stresses at both contact points and to reduce the over-all squeeze. It is not known what improvements in performance, if any, have been gained from this design modification. The specially-designed lipseals for the gas generator oxidizer and fuel valves provided reasonably satisfactory service; therefore, no further development was performed with these seals during the final phase of the M-1 Program. Two basic configurations of the 11-in. lipseal for the sleeve-type thrust chamber valve were evaluated; these were the "supported" (see Figure 119) and the "unsupported" (see Figure 120). The basic development of these two types of seals was done in the dynamic seal test cell (see Figure 117) and the final refinement was made in the actual thrust chamber valve component hardware. The results of the tests for the different seal configurations in test cells and components are discussed in the following paragraphs according to seal type.

1. Omniseals

All tests with the Omniseals were performed during development and testing of actual component hardware (M-1 gas generator oxidizer and fuel valves). These components are shown on Figures 78 and 79. No attempt was made in these tests to determine wear rate and friction, or the affect of pressure, temperature, and other factors upon these variables. Some simple "total" friction tests (for all of the component seals combined) were conducted to gain the necessary information for actuator sizing, actuation pressures, and valve timing; however, these data did not relate to friction performance of individual seals. Proof, leakage, and functional cycling (endurance) tests were conducted using the components with the seals installed. The results of these tests are discussed in the following paragraphs.

a. Proof Test

The 3-3/8-in. and 1-7/8-in. Omniseals (primary fluid seals) were subjected to static proof pressures of 2720 psig. The 1-1/8-in. and 11/16-in. Omniseals (leakage collectors or secondary seals) were subjected to a maximum of 1500 psig static proof pressure. These tests were performed with both gaseous nitrogen and liquid nitrogen. No seal failures were encountered during the proof tests. However, some instances of excessive leakage were encountered and these seals were replaced before the leakage tests were conducted.

TABLE XVIII

DYNAMIC SEAL CONFIGURATIONS TESTED

MANUFACTURER	NOMINAL SIZE I.D. (INCHES)	PART NUMBER	NOMINAL SQUEEZE ON CROSS-SECTION OR DIAMETRAL INTERFERENCE (IN.)	FUNCTION	SYSTEM FLUID	COMPONENT	SEAL MATERIALS
Aeroquip and Reid Enterprises (Omniseal)	3.375	R10105-237A3M	.017 (Squeeze)	Pintle Shaft Primary Seal	Liquid Hydrogen	M-1 Gas Generator Fuel Valve (GGFV)	Teflon (TFE) Jacket, Beryllium Copper Springs (3 Springs)
	1.875	R10105-134A3M	.0135 (Squeeze)	Pintle Shaft Primary Seal	Liquid Oxygen	M-1 Gas Generator Oxidizer Valve (GGOV)	
	1.125	R10105-122A3M	.0155 (Squeeze)	Pintle Shaft Secondary Seal	Liquid Hydrogen	M-1 Gas Generator Fuel Valve (GGFV)	
	.687	R10105-115A3M	.016 (Squeeze)	Pintle Shaft Secondary Seal	Liquid Oxygen	M-1 Gas Generator Oxidizer Valve (GGOV)	
Aerojet-General Corporation Unsupported Lipseal	3.375	703898-61	.058 (Interference)	Pintle Shaft Primary Seal	Liquid Hydrogen	M-1 Gas Generator Fuel Valve (GGFV)	Kel-F per AGC-44028
	1.875	703898-43	.031 (Interference)	Pintle Shaft Primary Seal	Liquid Oxygen	M-1 Gas Generator Oxidizer Valve (GGOV)	
	1.125	703898-25	.030 (Interference)	Pintle Shaft Secondary Seal	Liquid Hydrogen	M-1 Gas Generator Fuel Valve (GGFV)	
	.687	703898-7	.015 (Interference)	Pintle Shaft Secondary Seal	Liquid Oxygen	M-1 Gas Generator Oxidizer Valve (GGOV)	
Unsupported Lipseal	11.00	265114	Varied .100 to .150 (Interference)	Sleeve Seal	Liquid Oxygen and Liquid Hydrogen	M-1 Thrust Chamber Valve (TCV) and Test Cells	
		705458	Varied .100 to .150 (Interference)	Sleeve Seal			
Supported Lipseal	11.00	706080	Varied .100 to .150 (Interference)	Sleeve Seal			

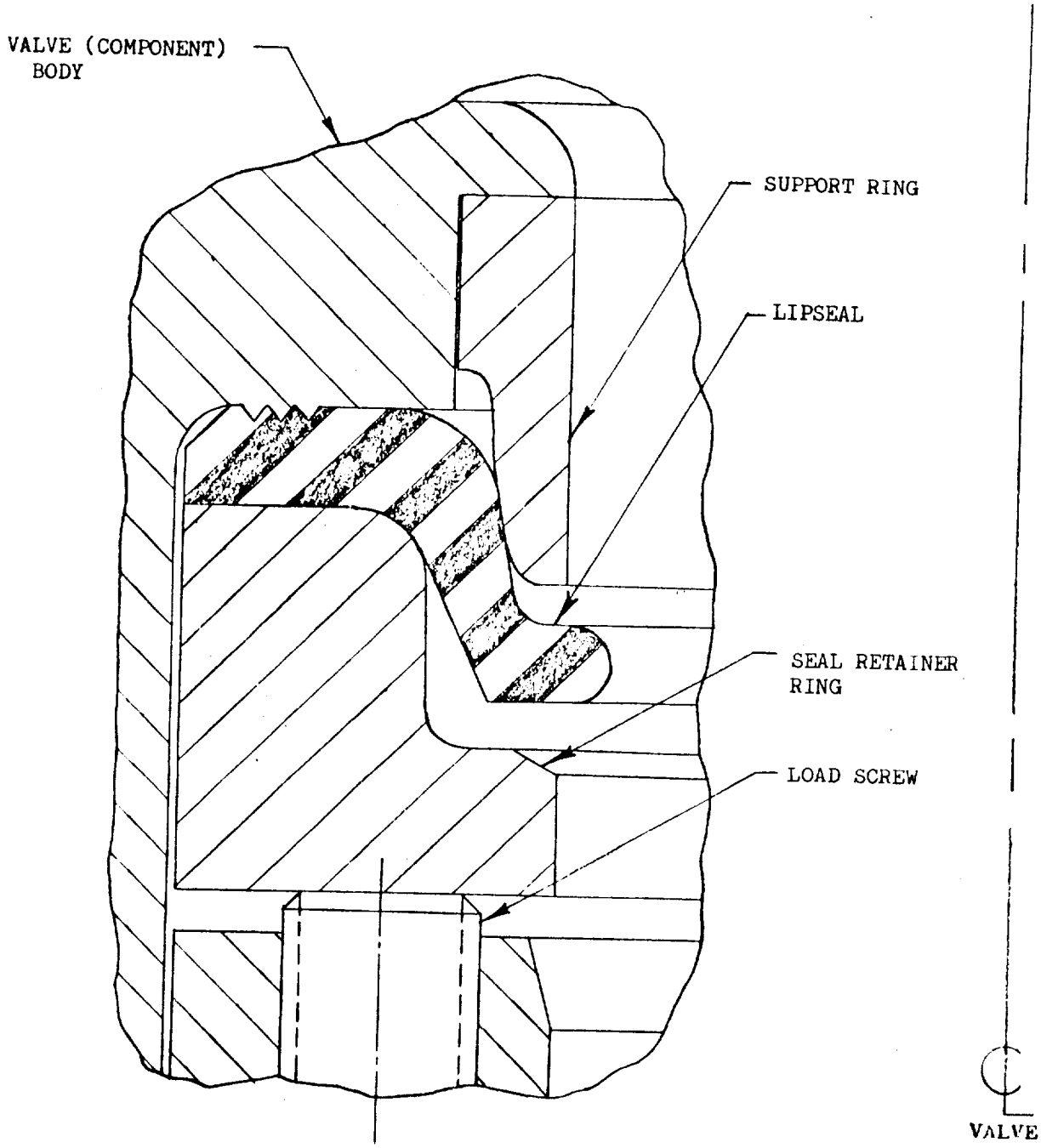


Figure 119
Supported Lipseal P/N 706080, M-1 Thrust Chamber Valve,
Installed Position Without Sleeve

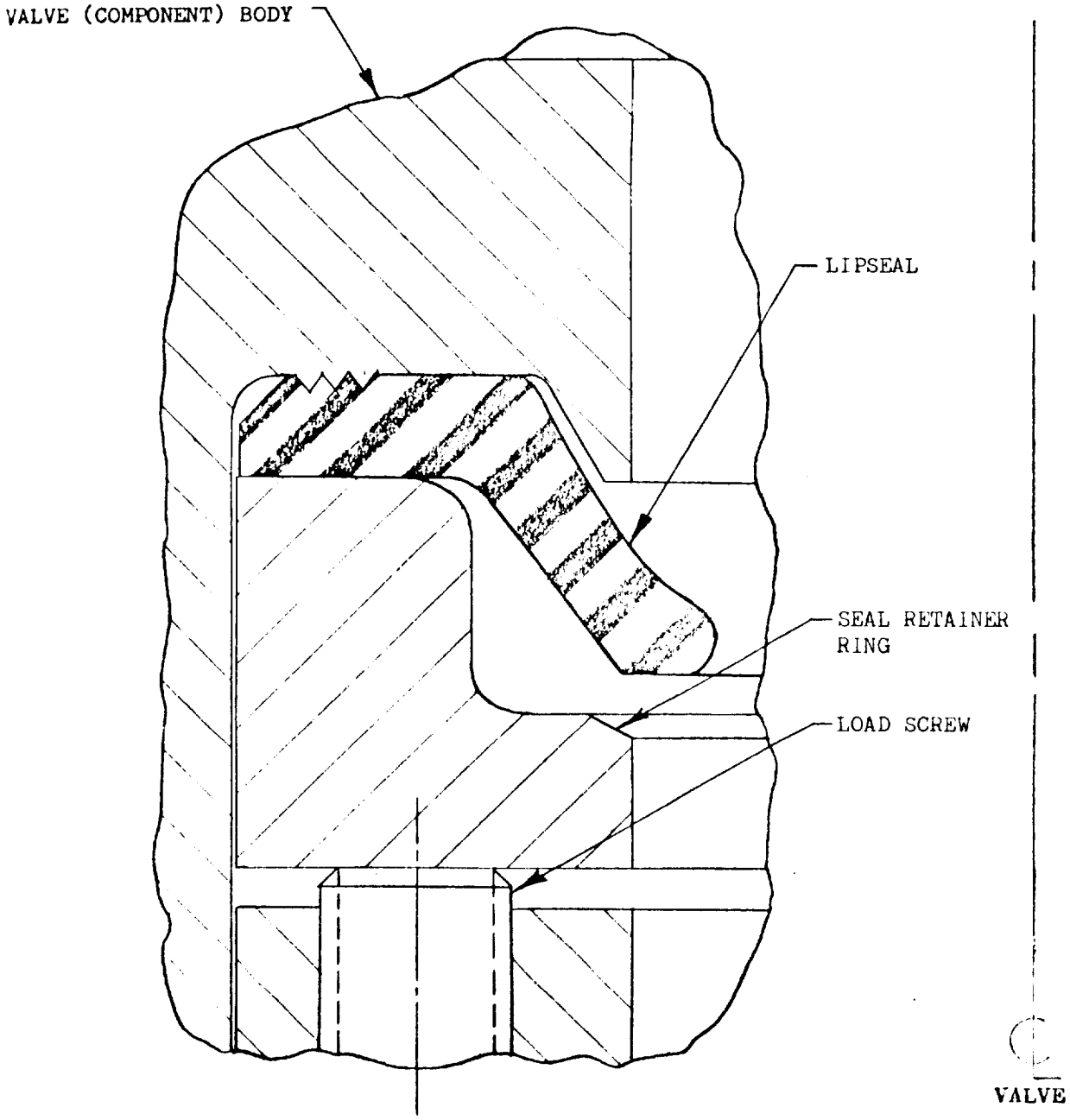


Figure 120
Unsupported Lipseal P/N 705458, M-1 Thrust Chamber Valve, Installed Position Without Sleeve

b. Leakage Tests

A total of 35 ambient and 18 cryogenic leak tests were conducted with the Omniseals. Leakage tended to stabilize (no increase in leakage with increased pressure) above 100 psig and then decrease to 1300 to 1400 psig at ambient temperature, indicating that the Omniseal is "pressure-energized" above 100 psig. The trend is shown on Figure 121. Leakage at cryogenic temperature was usually greater than leakage at ambient temperature. However, the slope of the leakage curve above 100 psig (see Figure 121) indicates that the sealing stresses still increase with increased pressure (i.e., the seal is still "pressure-energized") above 100 psig. The change in characteristics from ambient to cryogenic temperature is assumed to be a function of the change in material hardness at low temperature. The increased hardness of the jacket at low temperature would require a greater increase of sealing stresses with pressure to maintain the same leakage/pressure characteristics. These data suggest that the slope of the leakage/pressure curve would increase as the jacket material temperature decreases (jacket hardness increases); at least down to the glassy transition (Tg) of the Teflon jacket material.

c. Functional Cycling (Endurance) Test

Functional cycling tests were performed at both ambient and cryogenic temperatures. Except for the 3-3/8-in. Omniseal, leakage tended to decrease with functional cycles at ambient temperature as shown on Figures 122 through 125. Leakage tended to increase the first 20 to 30 functional cycles with the 3-3/8-in. Omniseal. The ambient temperature data (see Figure 122) above approximately 40 functional cycles was derived from limited tests, but indicates that leakage would stabilize and decrease to 100 functional cycles. Except for the 1-1/8-in. Omniseal, leakage tended to increase with function cycles at cryogenic temperature. "Rolling" failures (rotation of the seal cross-section in the groove as shown on Figure 126) occurred at both ambient and cryogenic temperature during functional cycling, but most of these failures occurred at cryogenic temperature. Rolling failures and excessive leakage at cryogenic temperature was the predominant cause of the limited data above 40 functional cycles. Rolling failures were attributed partially to the critical assembly requirements (one unit failed during the initial leakage test); however, even with improved assembly techniques failures occurred at cryogenic temperature.

d. Friction Tests

The total static (or starting) friction was calculated from actuation pressures required to start motion of the GGFV and GGOV during functional acceptance tests and from cryogenic functional tests with no pressure applied to the seals. The total friction for the GGFV seals (3-3/8-in. and 1-1/8-in. sizes) was approximately 170 lb at ambient temperature and approximately 1300 lb at cryogenic temperature. The total friction for the GGOV seals (1-7/8-in. and 11/16-in. sizes) was approximately 120 lb at ambient temperature and 400 lb at cryogenic temperature.

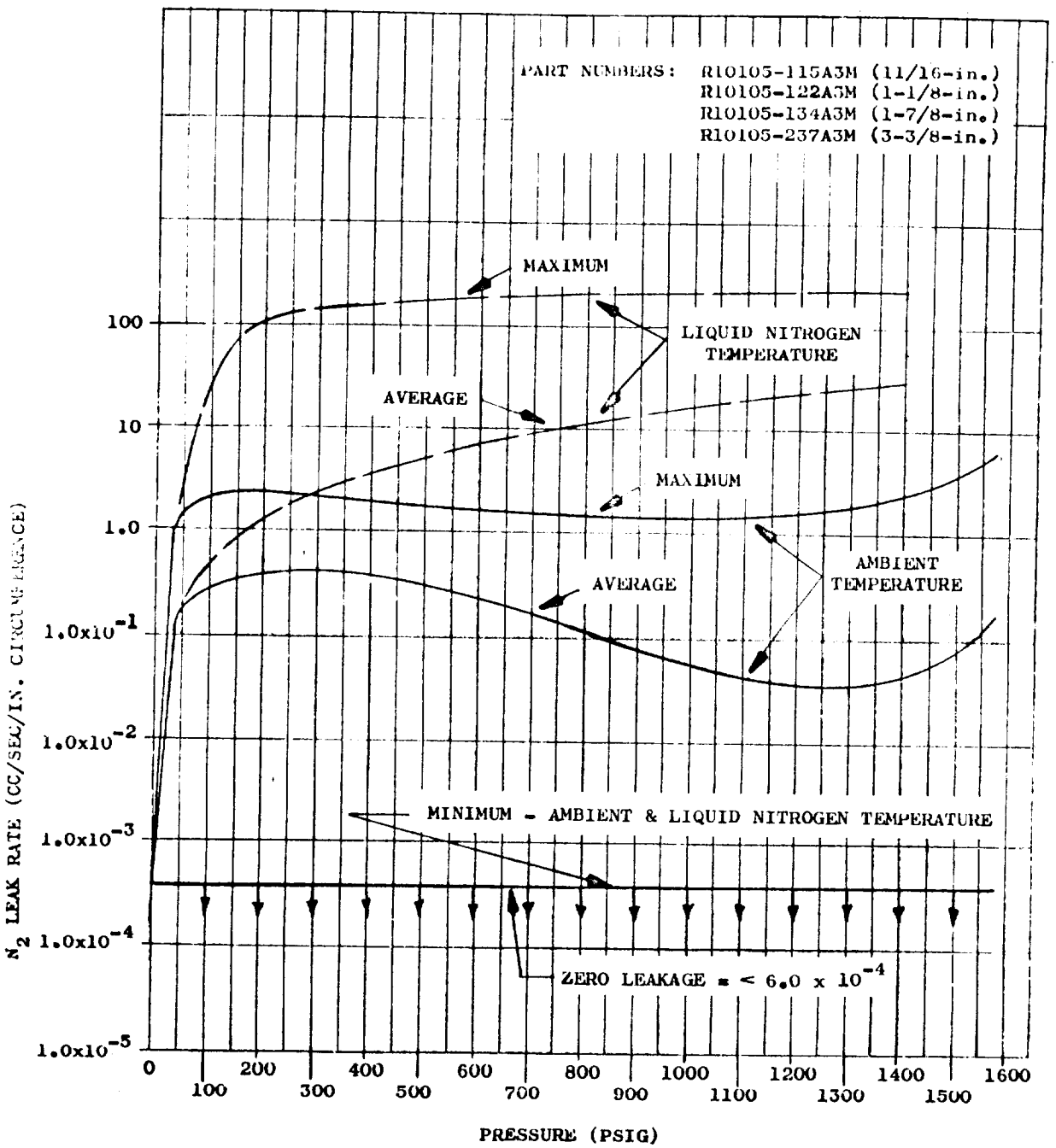


Figure 121
Leakage vs. Pressure, Ambient and Liquid Nitrogen Temperature (Reid Enterprises Omniseal)

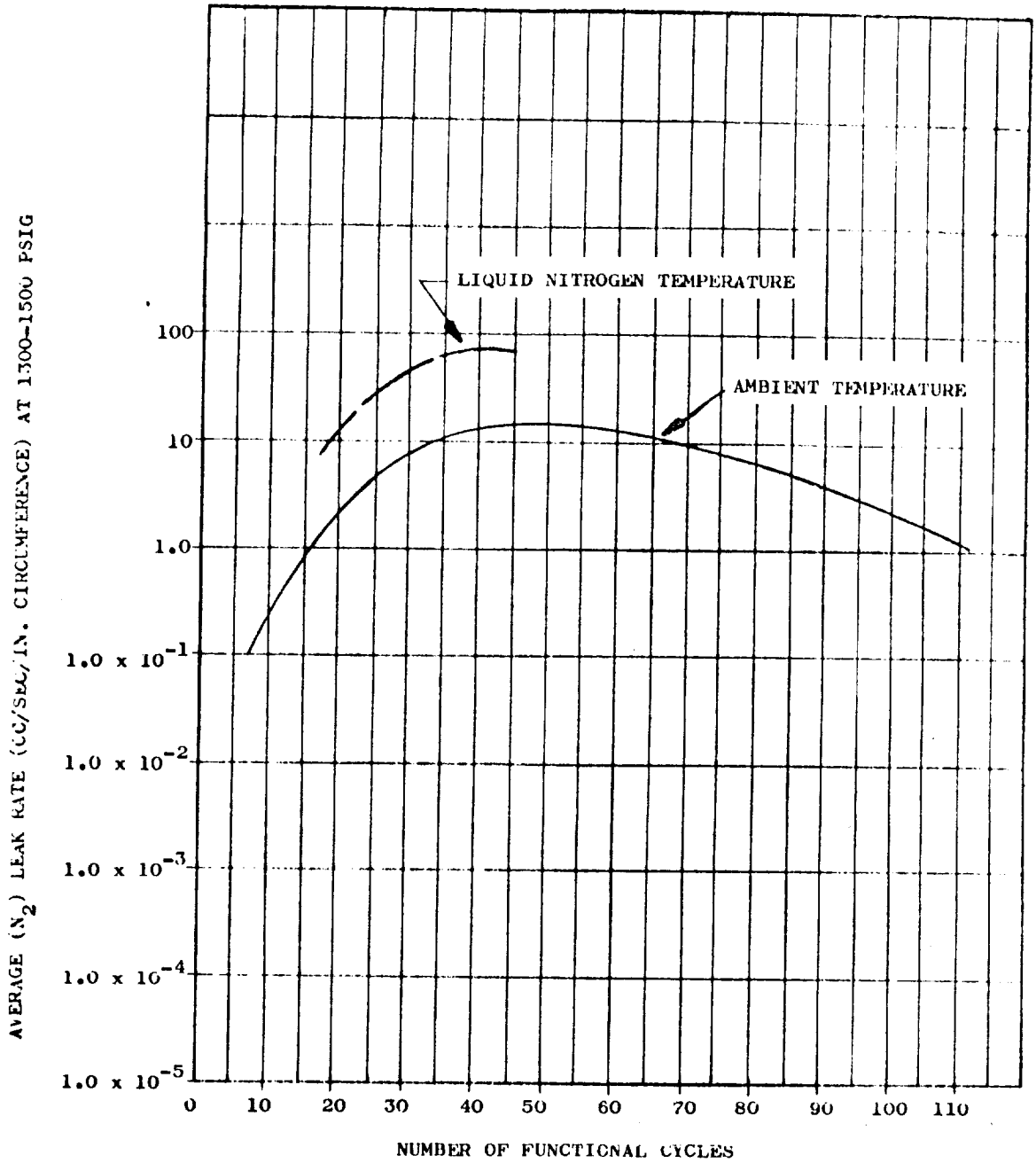


Figure 122
Leakage vs. Number of Functional Cycles, 3-3/8
"Omniseal" P/N R10105-237A3M

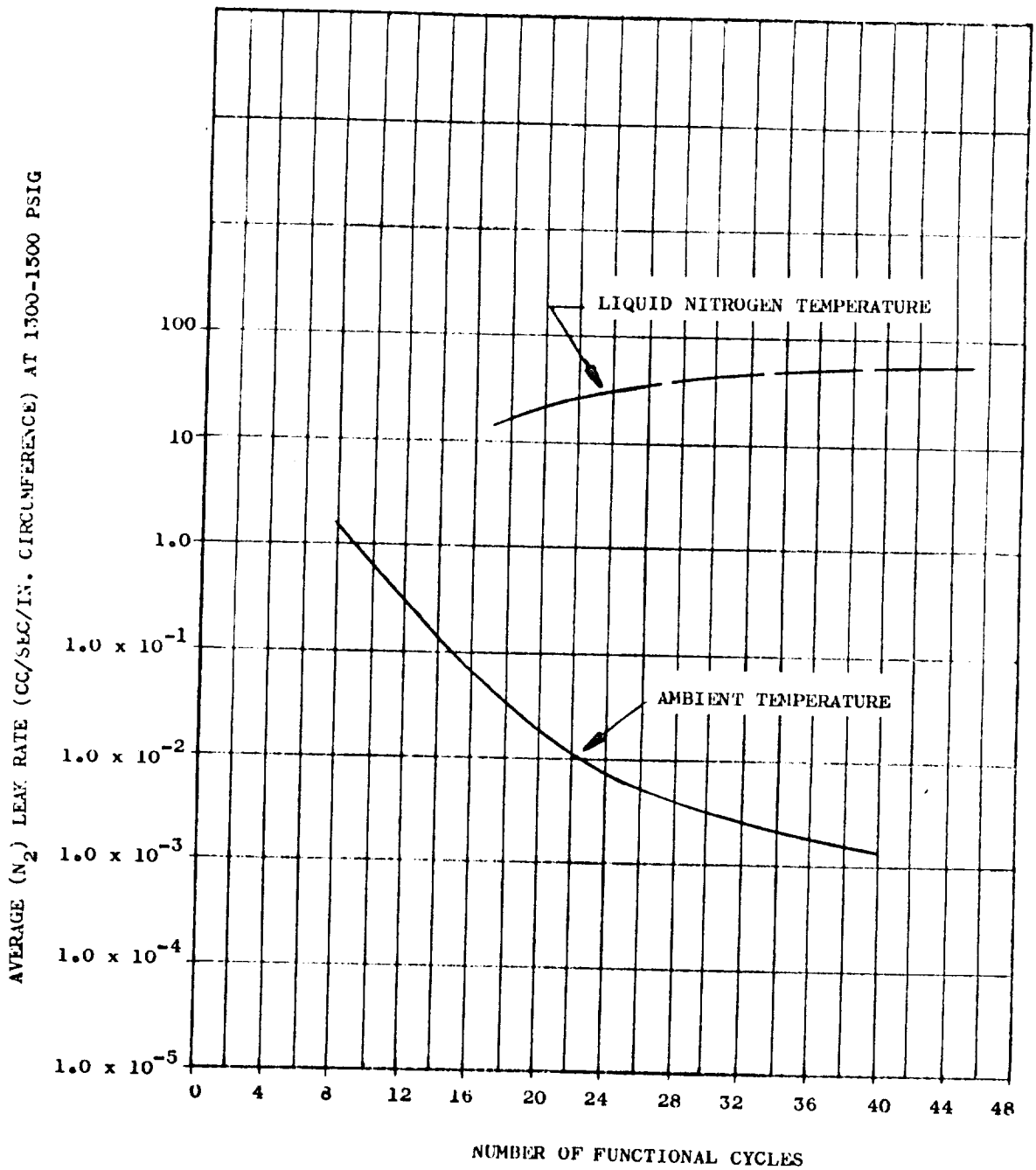


Figure 123
Leakage vs. Number of Functional Cycles,
1-7/8-in. "Omniseal" P/N R10105-134A3M

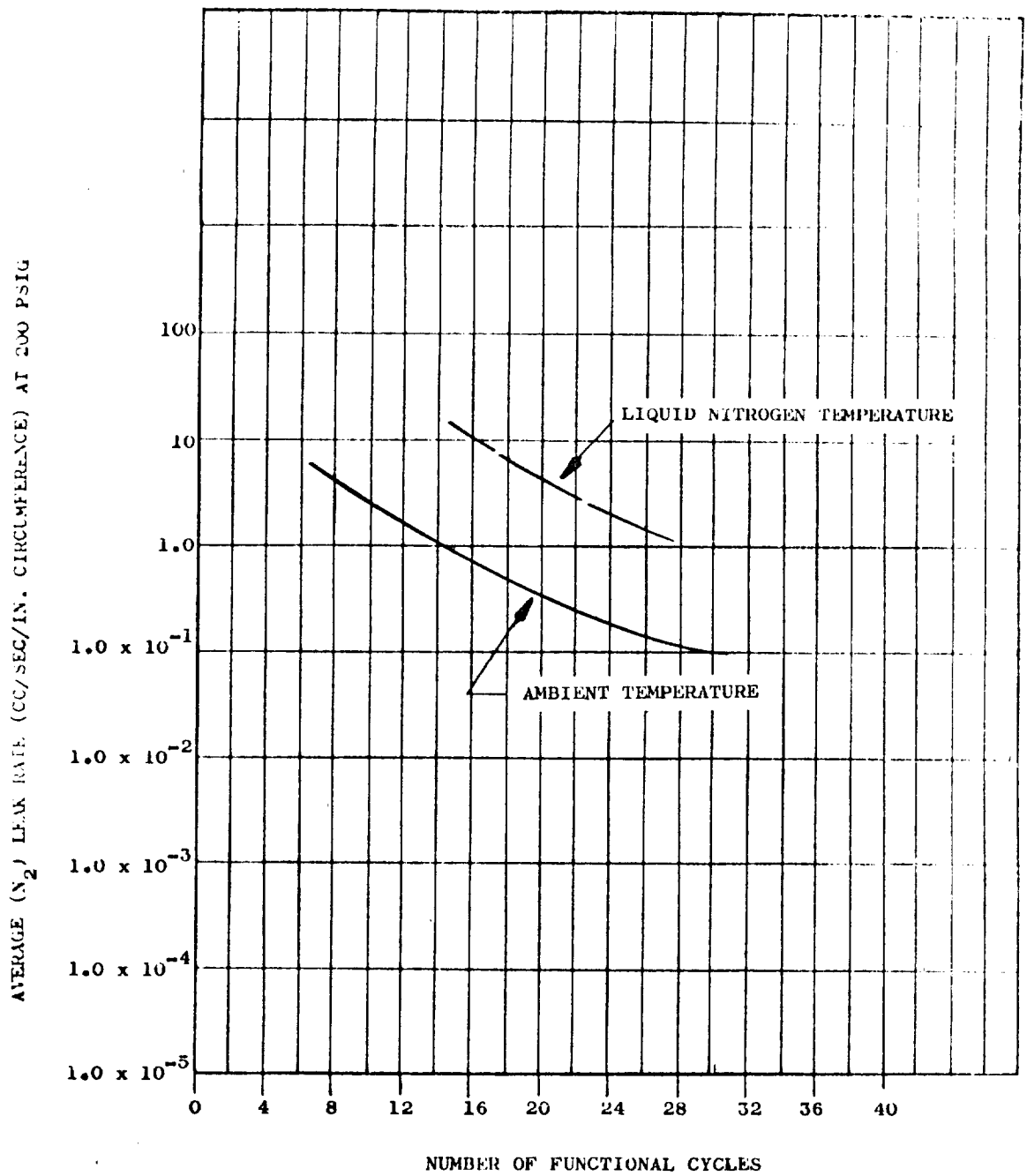


Figure 124
Leakage vs. Number of Functional Cycles,
1-1/8-in. "Omniseal" P/N R10105-122A5M

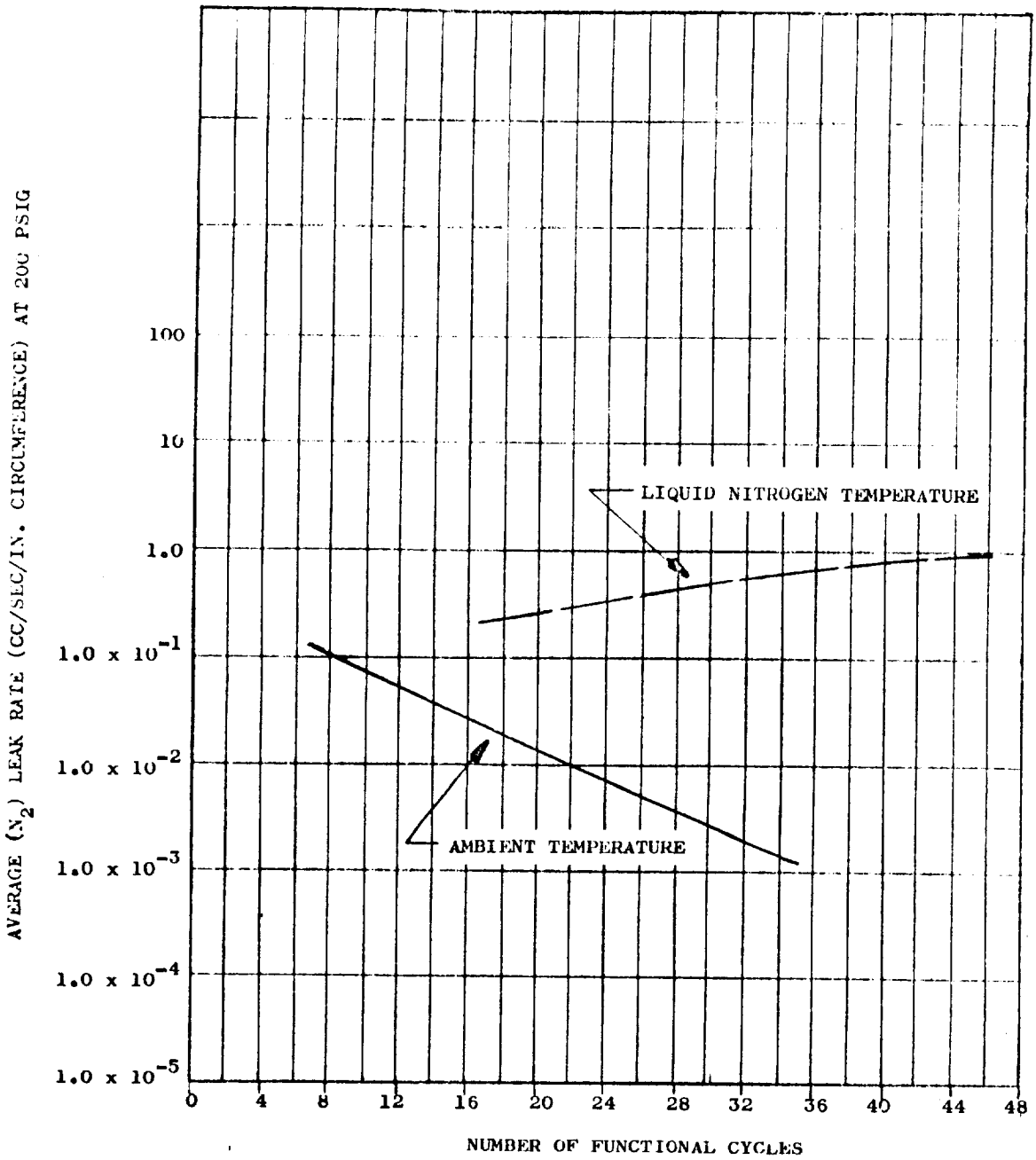


Figure 125
Leakage vs. Number of Functional Cycles,
11/16-in. "Omniseal" P/N RD109-115A3M

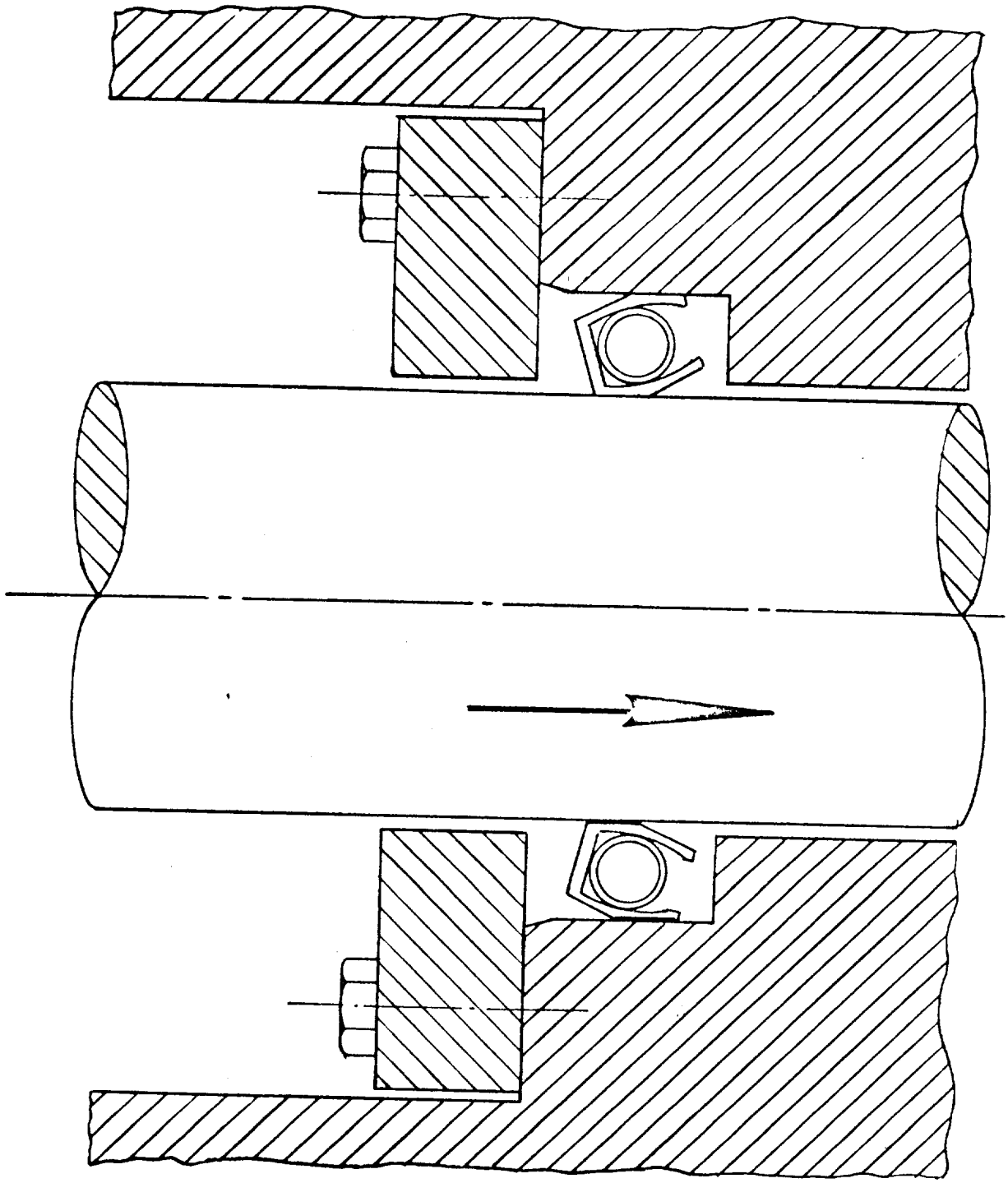


Figure 126
"Rolling" Failures on Omniseals, Basic P/N R10105

2. Small Lipseals (3-3/8-in., 1-7/8-in., 1-1/8-in., and 11/16-in.)

All tests with the small lipseals were performed in conjunction with development and evaluation tests of component hardware (M-1 gas generator valves which are shown schematically on Figures 78 and 79). No attempts were made during these tests to determine wear rates and friction, or the affect of temperature, pressure, and other factors upon these variables. Proof, leakage, functional cycling (endurance), vibration, and actual hot firing tests with mating hardware were conducted using the components with the lipseals installed. The results of these tests are discussed in the following paragraphs.

a. Proof Tests

The 3-3/8-in. and 1-7/8-in. lipseals (primary fluid seals) were subjected to static proof pressures of 2720 psig. The 1-1/8-in. and 11/16-in. lipseals (leakage collectors or secondary seals) were subjected to a proof pressure of 2000 psig. The proof tests were performed at both ambient and liquid nitrogen temperatures. No seal failures were encountered during these tests.

b. Leakage Tests

The 3-3/8-in. and 1-7/8-in. lipseals (primary seals) were subjected to incremental leakage pressures of up to 1550 psig. The 1-1/8-in. and 11/16-in. lipseals (secondary seals) were subjected to incremental leakage pressures up to 200 psig. At ambient temperatures, leakage tended to decrease with increased pressures above 50 psig as expected because the lipseal design is considered "pressure-energized." However, at cryogenic temperatures, leakage tended to increase with increased pressures. The rate of increase with increased pressure at cryogenic temperatures indicates that the lipseals are still "pressure-energized." These trends are shown on Figure 127. The reason for the difference in leakage trends at ambient and cryogenic temperatures is not completely understood. Theoretically, sealing stresses would tend to increase with lower temperatures because of the difference in contraction characteristics of the shaft and the lipseal materials. Apparently, lipseal hardness at the cryogenic temperature conditions is sufficient to require a greater increase in loading (sealing stresses) to maintain the lower leakage rates. Also, the stiffness of the section (material modulus) increases as temperatures decrease thereby causing the section to become less sensitive to pressure. The leakage rate would return to zero (less than 6.0×10^{-4} cc/sec) 75% of the time when the temperature returned to ambient. In all but one case, leakage was within acceptable limits. However, this single instance of excessive leakage was associated with galling of the shaft sealing surface after approximately 100 functional cycles.

c. Functional Cycling (Endurance) Tests

Functional cycling tests were performed with the component at both ambient and cryogenic temperatures. One test series consisted of a total of more than 100 cycles; half were conducted at ambient temperatures

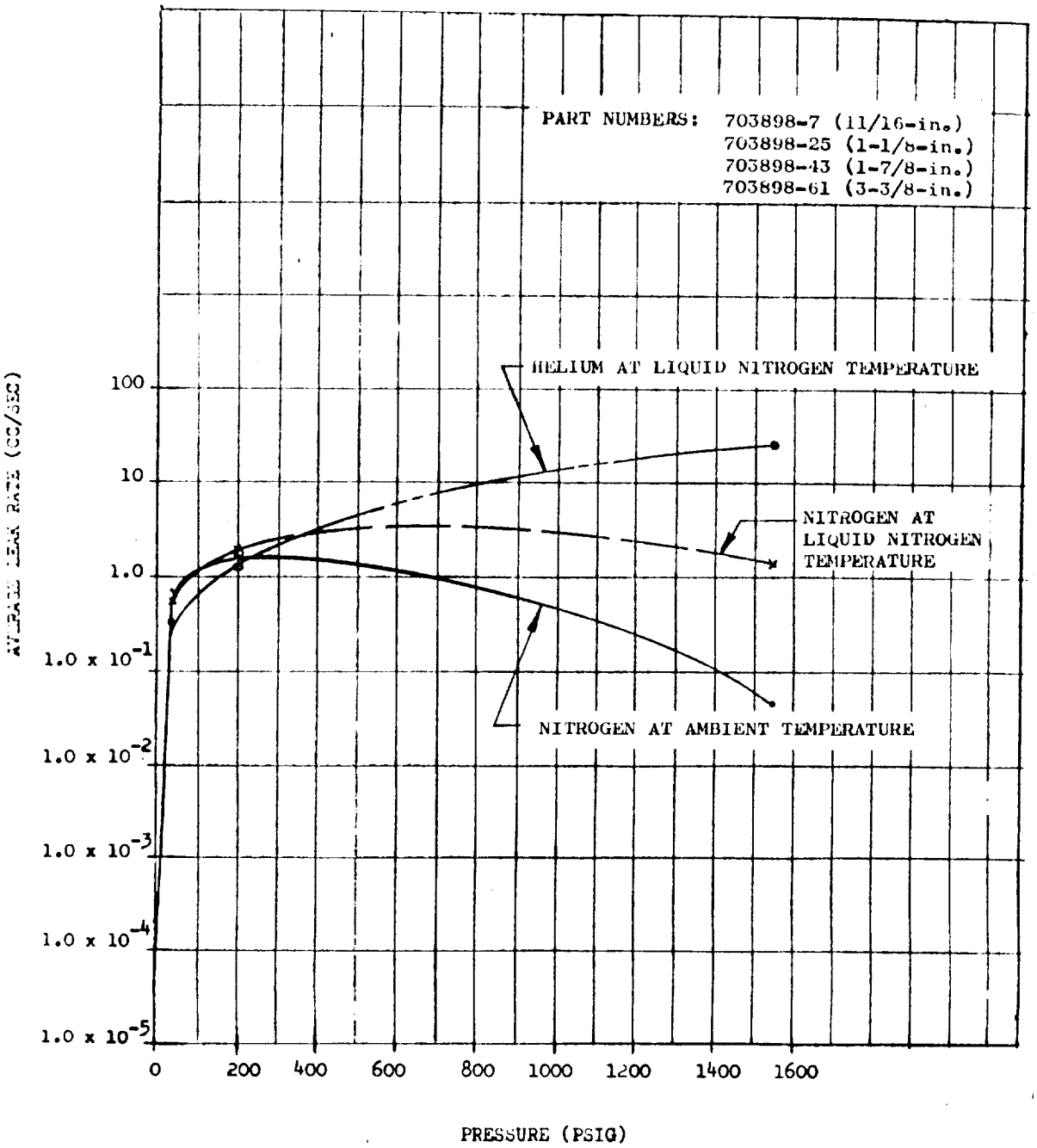


Figure 127
Leakage vs. Pressure at Various Temperatures, Aerojet-General Lipseals

and the remainder at liquid nitrogen temperatures. Excessive leakage was recorded with two seals (3-3/8-in. and 1-1/8-in.) after these tests. The excessive leakage was attributed to galling of the shaft sealing surface caused by rubbing of the shaft on the valve body clearance diameter. Although no specific leakage data was taken at various numbers of cycles during these tests, trends were established from all the functional and operating tests. These are presented on Figures 128 through 131. Except for the 1-7/8-in. size (see Figure 129), leakage rates tended to decrease initially with functional cycles at ambient temperature, then increase to 95 cycles. The inverse was true for the 1-7/8-in. size. No apparent reason could be found for this because the machining requirements for the mating surfaces were the same, the relative interference (with respect to diameters) were approximately equal, and all the lipseals were "scaled" versions of each other. Leakage was usually greater during liquid nitrogen temperature cycles and even greater still at liquid hydrogen temperature. No definite wear rate trends were established at cryogenic temperatures. In some cases (see Figures 128 and 130), leakage tended to increase with functional cycles at liquid nitrogen temperature. In other cases (see Figures 129 and 131), leakage tended to decrease. Leakage both increased (see Figure 130) and decreased (see Figure 128) with functional cycles at liquid hydrogen temperature in the GGFV. The increased interference (from differential contractions) would cause increased sealing loads and tend to increase wear rates. However, the hardness of the seal increases with decreased temperature and could improve wear rates to some extent. Also, the 3-3/8-in. seal (Figure 128) is a primary fluid seal (in contact with liquid hydrogen) and may gain improved wear characteristics by some lubrication from the fluid. The 1-1/8-in. seal (Figure 130) is a secondary seal that gains contact with the liquid hydrogen only when gross leakage occurs past the primary seal.

d. Friction Tests

The total static (or starting) friction for all of the seals combined in the mechanically-connected gas generator fuel and oxidizer valves was approximately 6000 lb. The maximum dynamic (or sliding) friction for all the seals combined in the two valves was determined to be approximately 3800 lb. No attempt was made to determine friction forces for each of the two valves or each seal, individually.

e. Vibration Test

The leakage on each of the lipseals installed in the gas generator valves (fuel and oxidizer) was less than 6.0×10^{-4} cc/sec before and after the completion of the vibration test at ambient temperature. Maximum leakage at liquid nitrogen temperature during the vibration test was less than 0.1 cc/sec. No seal failures (i.e., crazing or cracking) were encountered during these tests.

f. Hot Firing Tests

The components were assembled with the mating hardware and operating tests were conducted with the completed assembly with operating fluids (liquid hydrogen for the 3-3/8-in. and 1-1/8-in. seals and liquid oxygen

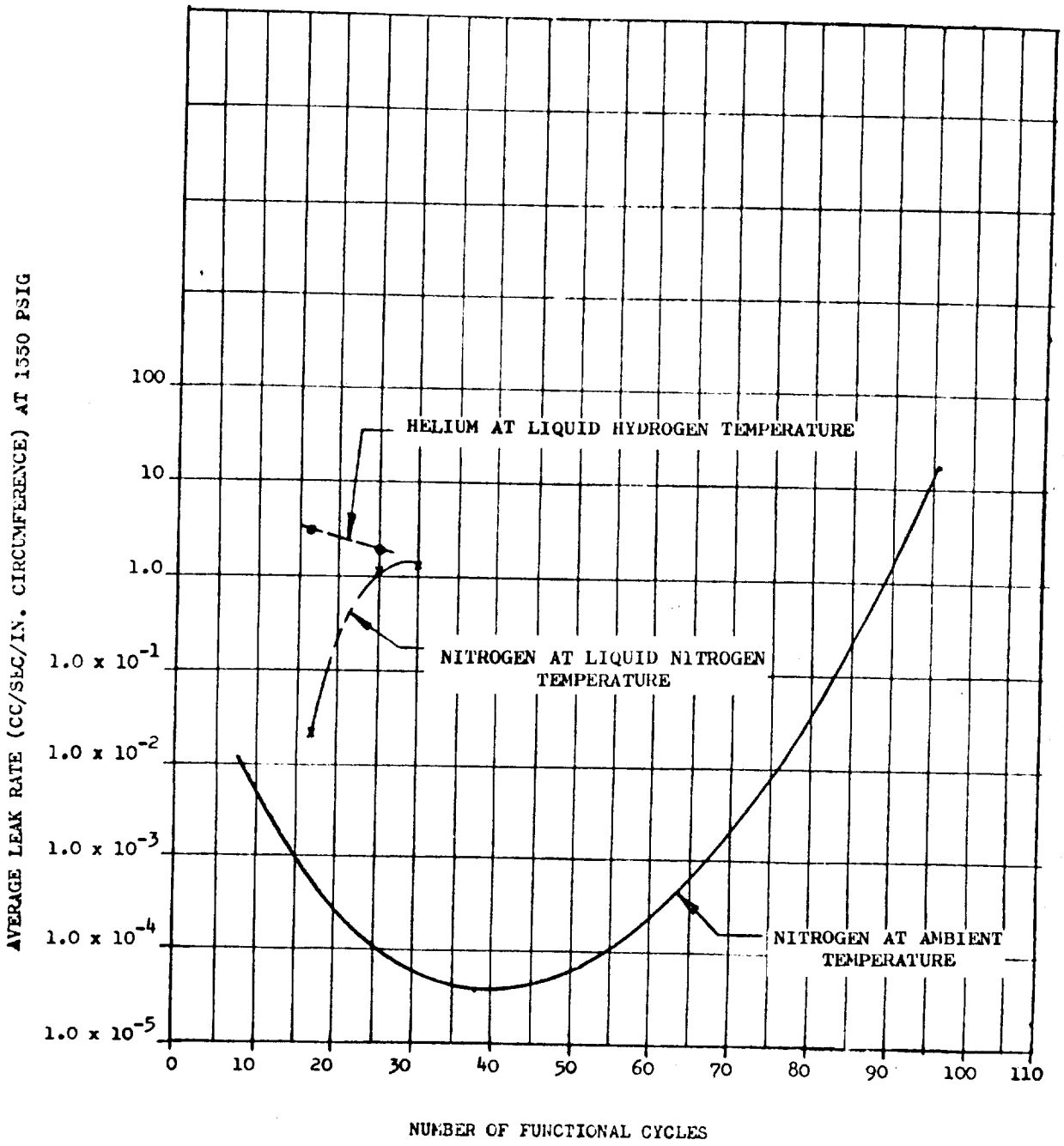


Figure 128
Leakage vs. Number of Functional Cycles,
Aerojet-General Lipseal, 3-3/8-in. Size

P/N 703898-43

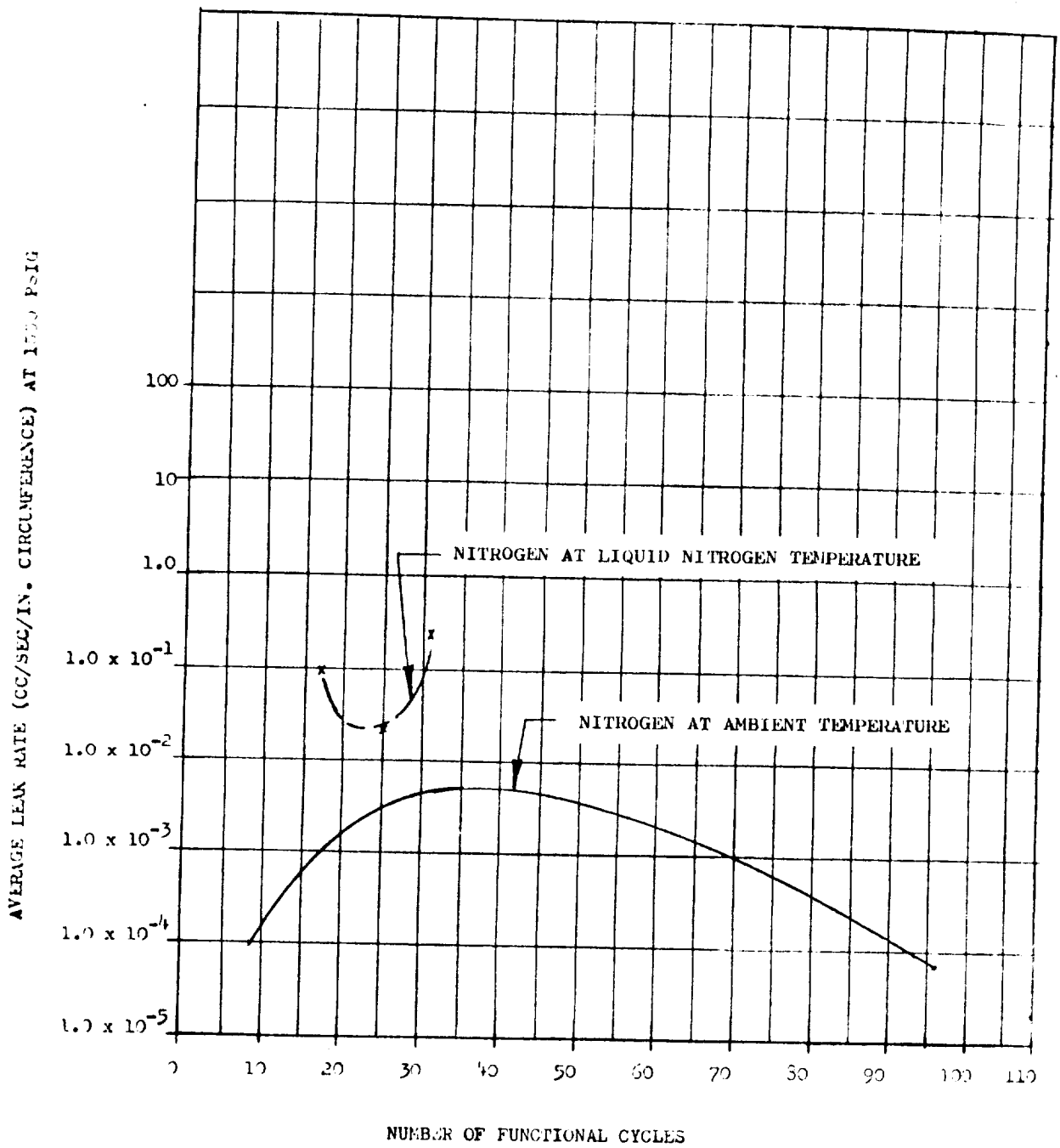


Figure 129
Leakage vs. Number of Functional Cycles,
Aerojet-General Lipseal, 1-7/8-in. Size

P/N 703898-25

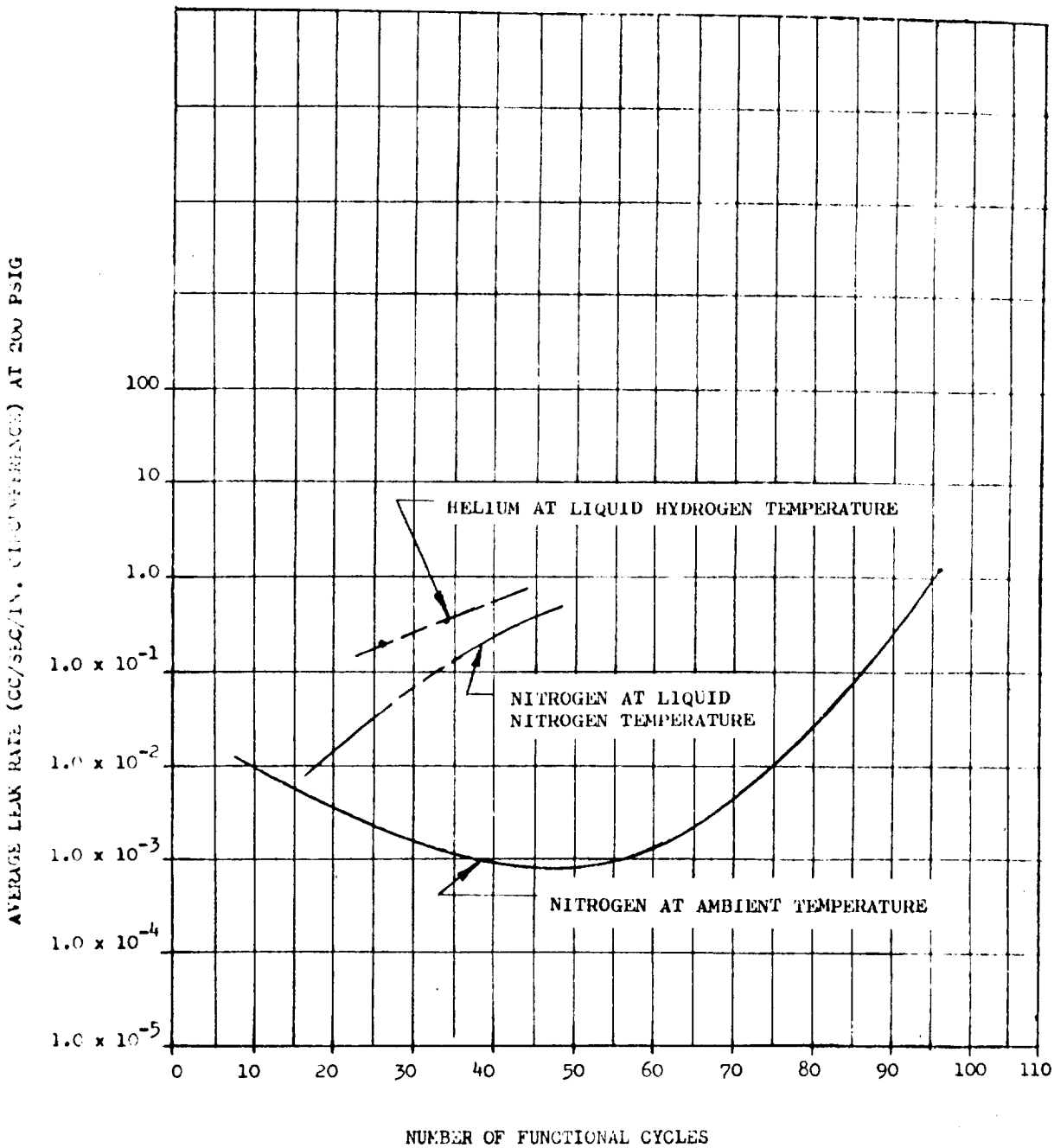


Figure 130
Leakage vs. Number of Functional Cycles,
Aerojet-General Lipseal, 1-1/8-in. Size

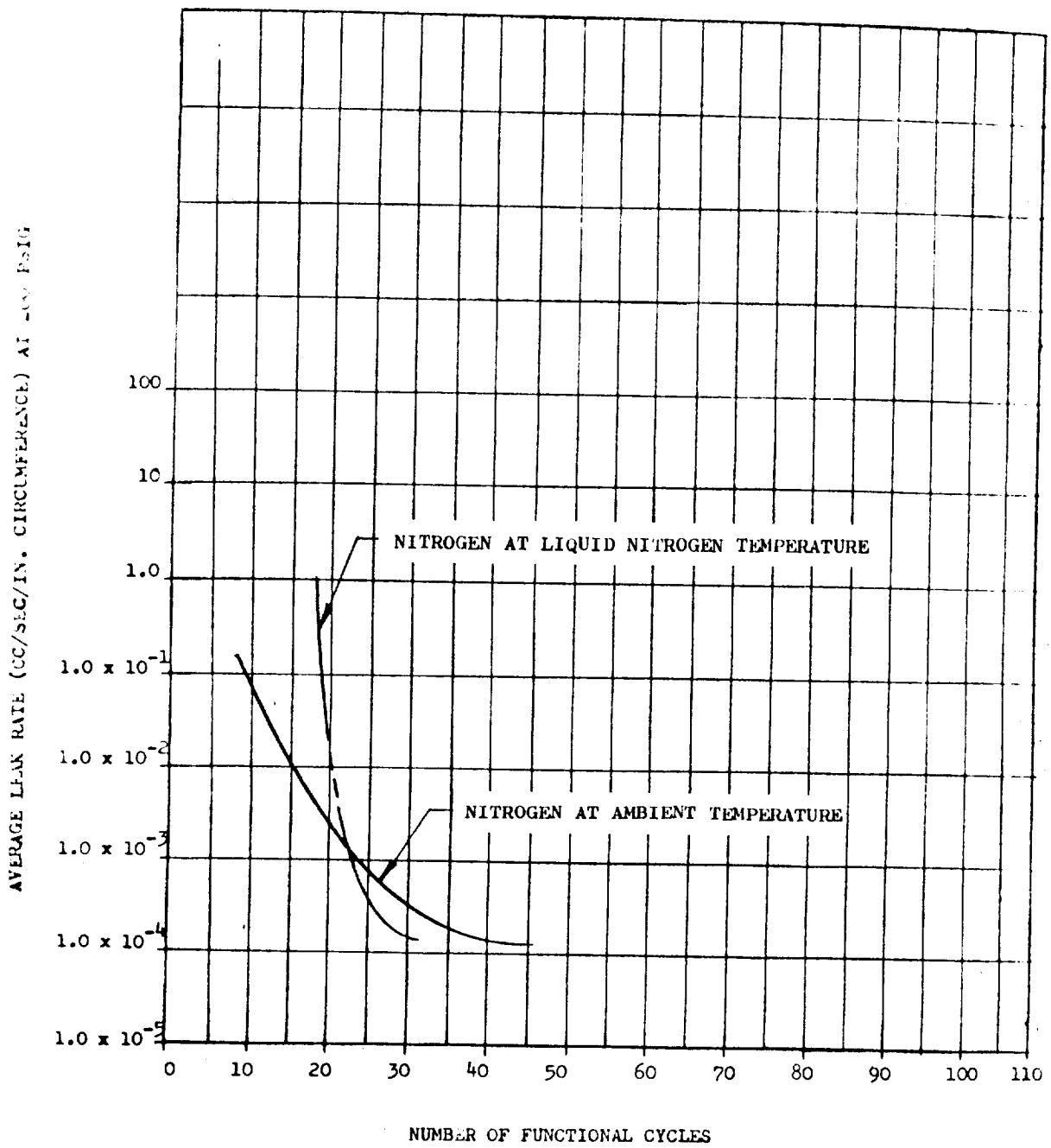


Figure 131
Leakage vs. Number of Functional Cycles,
Aerojet-General Lipseal, 11/16-in. Size

for the 1-7/8-in. and 11/16-in. seals), and rated pressures and flow rates. Eleven hot firings were conducted. There were four failures of the 1-7/8-in. (oxidizer) seal during these tests. The seal crazed or fractured near the bend (lip-to-flange intersection) radius of the seal cross-section. No fractures were encountered with any of the other three sizes and the cause of the failures with this particular (oxidizer primary) seal was never fully determined. All of these seals were essentially "scaled" versions of each other. Operating pressures of the oxidizer valve were approximately equal to those of the fuel valve. If contraction resulting from extremely low temperature was the problem, it would appear that the fuel (liquid hydrogen) seals would be more prone to failure than the oxidizer (liquid oxygen) seals. Three cases of excessive leakage were recorded, but these problems were ultimately attributed to component or seal machining deficiencies and not related to the basic design of the lipseals.

3. Large Lipseals (11.0-in.)

Tests were performed with the large lipseal designs for M-1 thrust chamber valves in both the specially-designed dynamic seal test cell as well as in conjunction with the component development and evaluation tests. The test cell was designed to simulate the basic valve configuration and most of the basic lipseal development testing was performed using the test cell. The test cell configuration was shown on Figure 117. The final refinement of the seal configuration was done in conjunction with the component hardware development and evaluation tests. Two basic lipseal designs were evaluated; the "supported" (see Figure 119) and the "unsupported" (see Figure 120). The original ("unsupported") lipseal design failed many times under various conditions of temperatures, pressures, and interference with the mating sleeve. These failures were attributed to brittleness at operating temperatures, contraction differences in the materials, and the differential pressures across the seal at the time of valve closure when the sleeve was inserted into the lipseal. The "supported" lipseal design was then introduced to overcome the problems encountered with the "unsupported" configuration. The major objective of the lipseal development program was to determine the optimum seal-to-sleeve interference and seal cross-sectional shape which would provide the minimum leakage, minimum wear rate (best life), minimum frictional drag, and withstand the forces imposed upon the seal during internal pressurization and sleeve actuation. The results of the evaluation of these seals are presented in the paragraphs which follow. Extensive details of the tests conducted have been omitted in this report, but specific findings and general trends are included.

a. Proof Tests

Proof tests were conducted using the "unsupported" lipseal design in the test cell. Initially, a 2200 psig water hydrotest was performed that caused excessive deformation of the seals at the point of contact with the sleeve. After the initial ambient hydrotest, gaseous nitrogen leakage rates were approximately 2.0×10^{-3} cc/sec/in. of circumference at ambient temperature between 1500 and 1850 psig, and approximately 2.5×10^{-3} cc/sec/in. of circumference at liquid nitrogen temperatures. Because 1850 psig

was a design differential pressure above the maximum expected at operating conditions across the seal, the 2200 psig ambient hydrotest (actually a burst test for the seal) was eliminated. No detrimental distortions were encountered with 1850 psig proof pressure on the seal.

b. Initial Leakage Tests

Numerous leakage tests with several different seal cross-sections, interference, and other variables were conducted with both the "unsupported" and "supported" lipseal design. These tests were conducted in both the test cell and component hardware (M-1 thrust chamber valves shown on Figure 77). Leakage rates less than 4.7×10^{-5} cc/sec/in. of circumference was consistently obtained at ambient temperature with nominal diametrical interference (between the seal lip and the sleeve) from 0.100-in. to 0.140-in. with both the "unsupported" and "supported" design. Leakage rates at liquid nitrogen temperature were usually greater than those at ambient temperatures. Leakage rates at liquid nitrogen temperature decreased as the diametral interference was increased as shown on Figure 132. With diametral interferences greater than 0.130-in., leakage rates at liquid nitrogen temperature were less than 4.0×10^{-5} cc/sec/in. of circumference over a pressure range of 0 psig to 1800 psig (before functional tests were conducted) with both the "supported" and "unsupported" lipseal.

The effect of pressure upon initial leakage depended upon the cross-sectional shape, interference, and other factors. Usually, maximum leakage would occur at some intermediate pressure between 0 psig and 1800 psig. That is, leakage would tend to increase up to some intermediate pressure between 0 psig and 1800 psig and then decrease as pressure was increased to 1800 psig. In some cases, leakage would reduce to less than 4.0×10^{-5} cc/sec/in. circumference at 1800 psig. Also, in some cases, leakage remained less than 4.0×10^{-5} cc/sec/in. of circumference over the full pressure range between 0 psig and 1800 psig. Temperatures also affected the "pressure-energizing" tendency of the lipseal. Again, this depended upon cross-sectional shape, interference, and other factors. For the same seal design, the pressure at which the maximum leakage rate would occur would differ at ambient and liquid nitrogen temperatures.

c. Functional Cycling (Endurance) Tests

Functional cycling tests in excess of 100 cycles were conducted with some of the more promising seal configurations using the specially-designed test cells. These tests were conducted at both ambient and cryogenic temperatures. Figure 133 shows the test cell installed in the functional cycling and cryogenic test chamber. Some seals (usually those with the greatest diametral interference) failed by crazing or breaking before completion of the desired minimum number of cycles. The craze or break generally occurred at the bend (lip-to-flange intersection) radius of the seal cross-section. In some cases, the break occurred at the sealing nub-to-lip intersection radius and others progressed from one (lip-to-flange intersection)

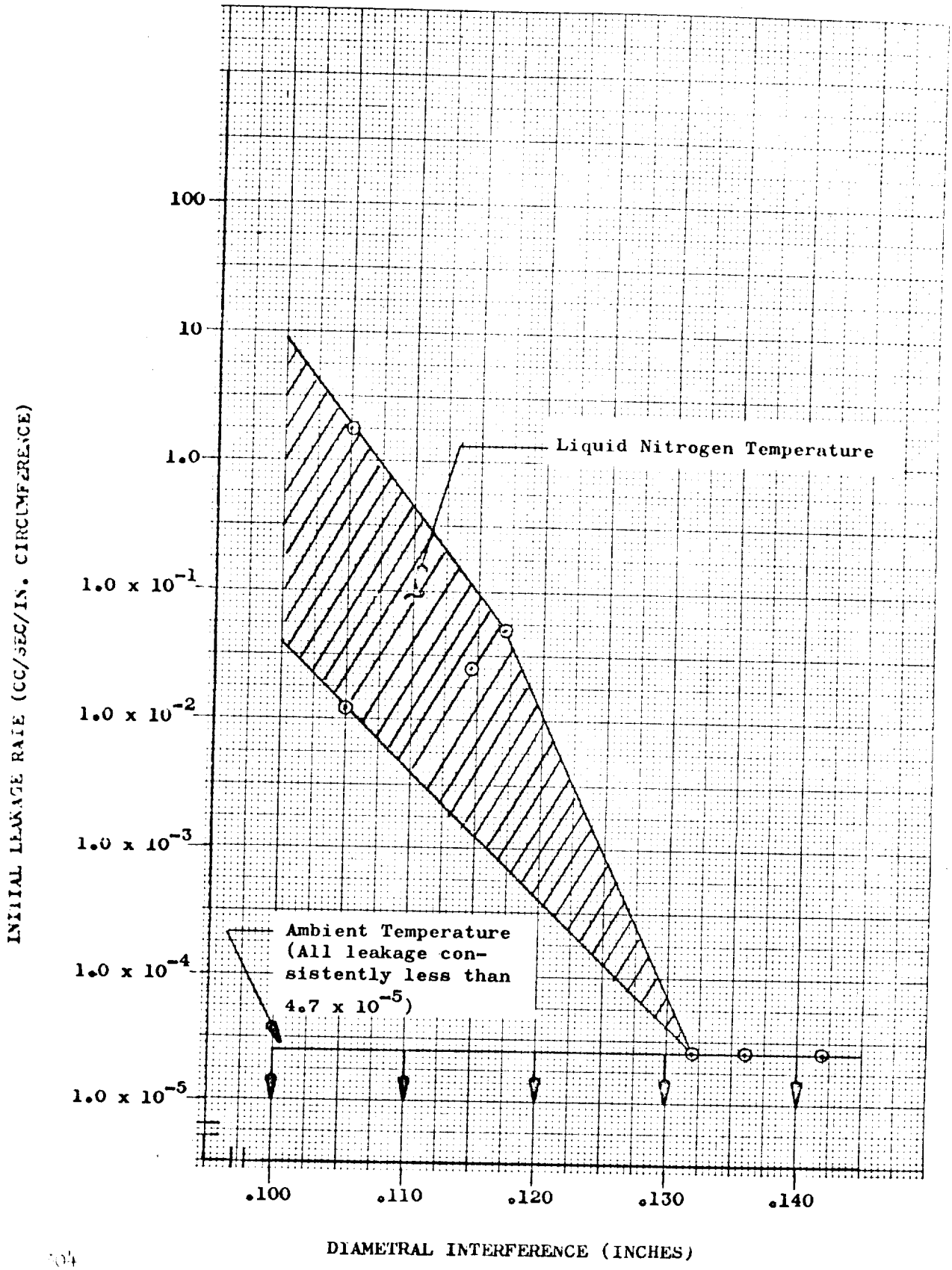


Figure 132

Initial Leak Rate vs. Diametral Interference, 11.0-in. Lipseal, "Supported" and "Unsupported"

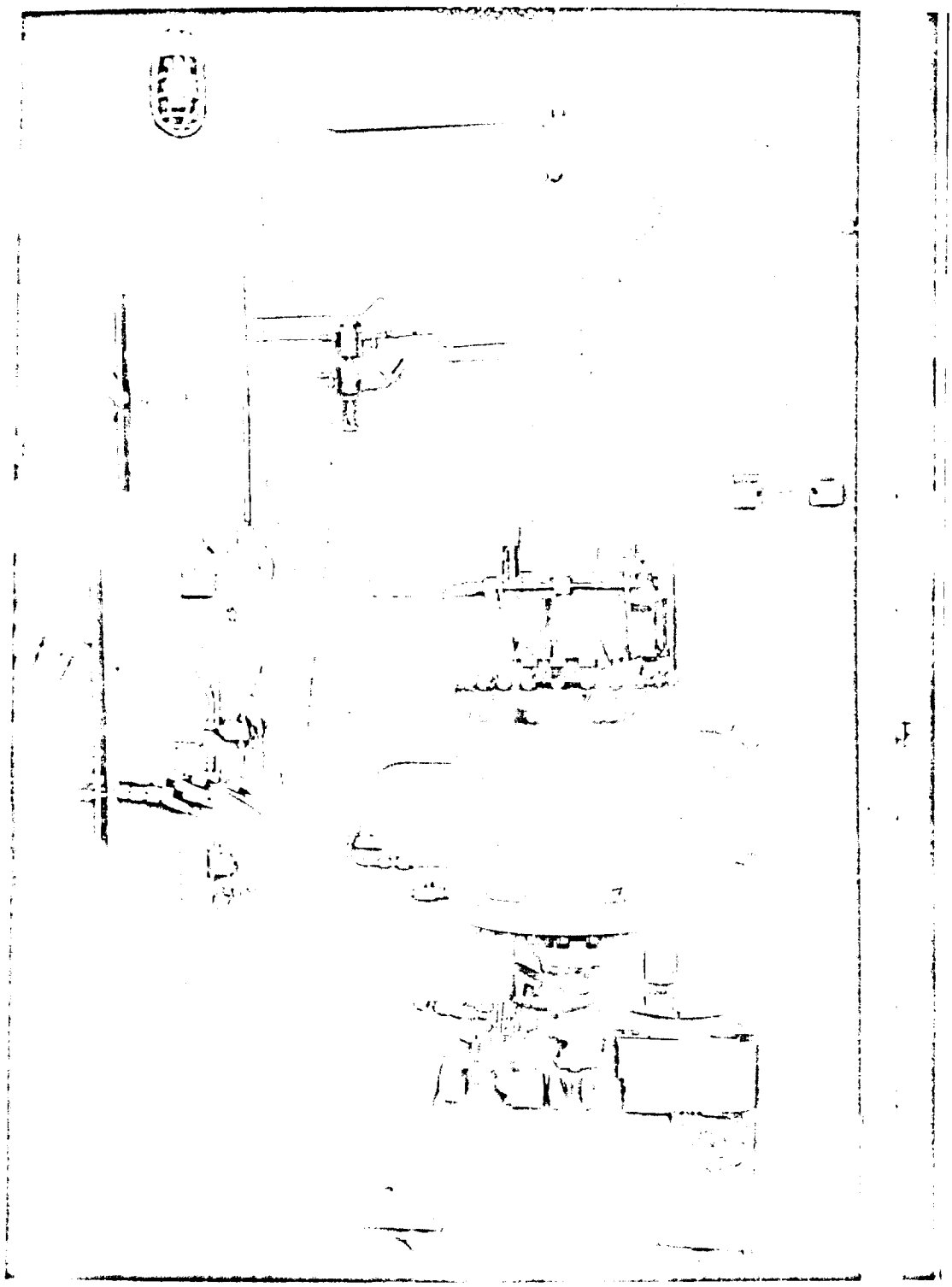


Figure 133
Typical Test Installation, M-1 Thrust
Chamber Valve Seal Tester

area to the other (sealing nub-to-lip intersection) area. Functional cycling at both ambient and cryogenic temperatures with the most promising designs was conducted well beyond the desired minimum cycle life without any crazing or breaking.

Leakage rates changed during functional cycling depending upon the cross-sectional shape of the seal, the diametral interference, and temperature. In general, for a particular cross-sectional shape, leakage tended to decrease as the valve was cycled at ambient temperature. At cryogenic temperatures, leakage decreased for a few cycles and then increased as the number of cycles increased when the seal interference was small. A typical example is shown on Figure 134. When the interference was large, leakage was less initially at low temperature but would increase more rapidly as the number of cycles increased at both ambient and cryogenic temperatures. Also, when the interference was large, the problem of crazing or breaking of the seal became more severe.

With the selected "unsupported" lipseal design, leakage was still less than 3.0×10^{-2} cc/sec/in. of circumference at liquid nitrogen temperature after 26 cycles. Also, leakage was less than 1.2×10^{-3} cc/sec/in. of circumference at liquid hydrogen temperature after 38 cycles. Although these leakage rates were somewhat greater than those of some of the other lipseal configurations, the selected design ultimately reflected a compromise between acceptable friction forces, wear rate, and initial leakage. Only the "supported" lipseal design provided consistently low leakage rates after the functional cycling.

d. Friction Tests

Friction tests were conducted with the various 11.0-in. lipseal configurations using the specially-designed test cells. A schematic diagram of the friction test set-up was shown on Figure 118. The lipseal friction forces were obtained by first obtaining total friction loads in the test cell without the particular lipseal installed (but with all other seals and equipment installed) at ambient and cryogenic temperatures. Then, the tests were repeated with the particular lipseal installed and the difference in the two loads was considered the function of the lipseal. Over-all friction forces obtained with the test cell were not of the same exact magnitude as those obtained with the thrust chamber valve component because of the minute differences in the actuation system and body design. However, consistent performance trends were obtained and reasonably accurate friction force magnitude for the lipseal was determined from these data. Total friction forces obtained from tests with the actual component (thrust chamber valve) hardware substantiated the results obtained with the test cells. Sliding (dynamic) friction forces remained constant for both the "supported" and "unsupported" lipseal designs at velocities between 2-in./sec and 10-in./sec (normal thrust chamber valve operating speed was 4-in./sec), but starting (static) friction increased very slightly as velocity increased. However, for all practical purposes, both static and dynamic friction remained constant for speeds ranging from 2-in./sec to 10-in./sec.

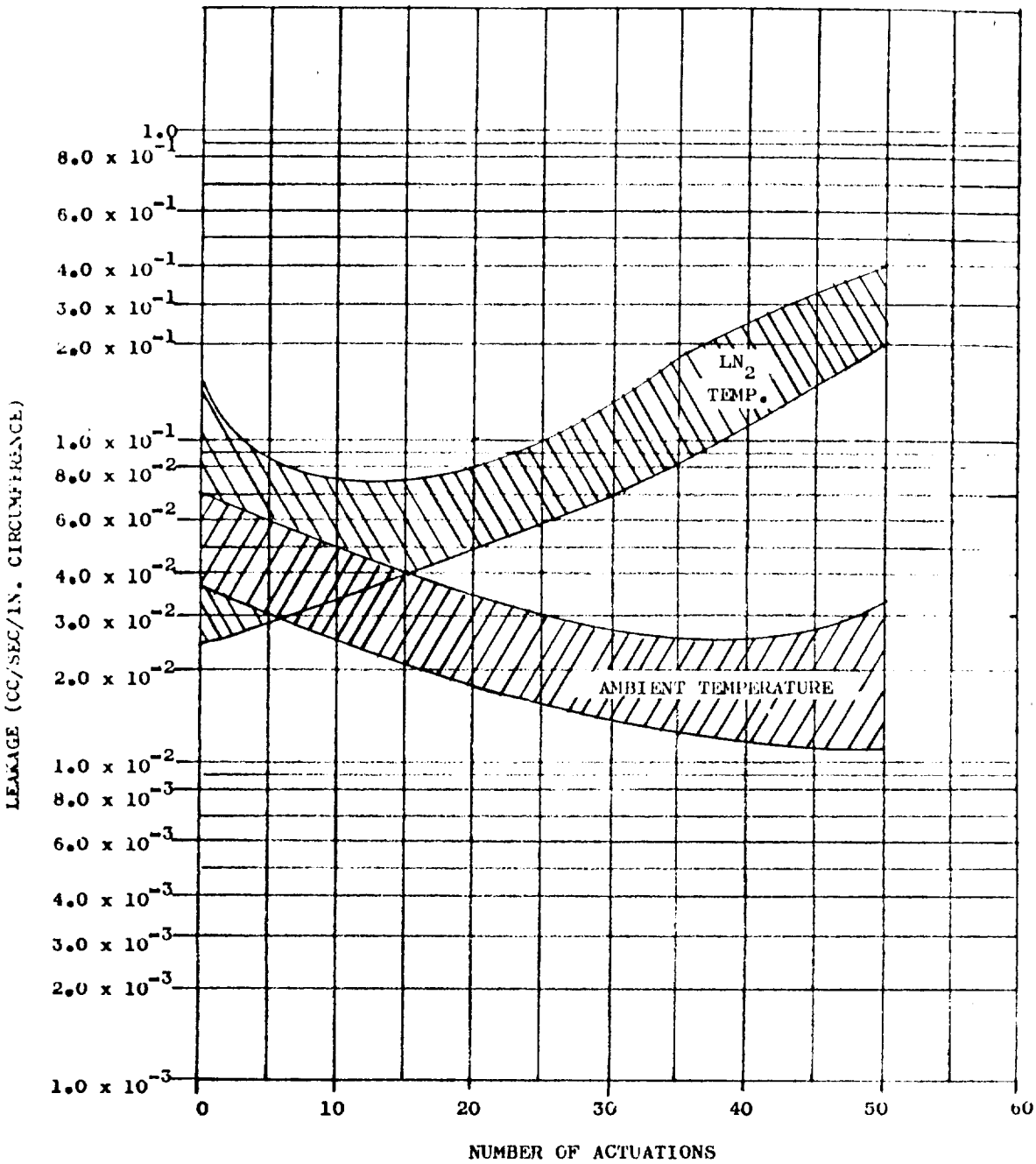


Figure 134
Leakage vs. Functional Cycles, 11.0-in. "Unsupported"
Lipseal P/N 265114-1 (.110-in. Interference)

Friction forces acting upon the "unsupported" lipseal increased with increasing pressure at both ambient and liquid nitrogen temperatures. The increase is a result of the "pressure-energizing" effect of the lipseal design. It is primarily the result of an increase in loading in the seal contact area with a possible change of the friction coefficient. Because the increase in friction force was essentially linear with pressure, it is apparent that the greatest affect upon friction forces results from the increase in normal loading of the seal lip with pressure. The rate of increase in friction with respect to pressure is a function of the lip length and section stiffness. A typical example of friction forces for the "unsupported" lipseal is presented on Figure 135. Friction forces for any particular cross-sectional configuration also increased as the diametral interference increased from 0.100-in. to 0.150-in. If friction forces obtained for a particular cross-section were considered excessive at intermediate diametral interferences between 0.100-in. and 0.150-in., no further friction tests were conducted with larger interference values. The selected lipseal configurations were a compromise between friction forces, wear rates, and sealing capability. Friction forces were also greater at liquid nitrogen temperature than at ambient temperature; that is, friction tended to increase as the temperature decreased. This was expected because the difference in contraction of the lipseal and sleeve material would tend to increase the normal loading at the seal contact area.

Friction forces were obtained with the "supported" lipseal design at various pressures and at ambient, liquid nitrogen, and liquid hydrogen temperatures. Friction forces increased with pressure up to an intermediate pressure between 0 psig and 1200 psig and then diminished as pressure was increased to 1200 psig. That is, maximum friction forces occurred at an intermediate pressure between 0 psig and 1200 psig. The intermediate pressure at which maximum friction occurred decreased as the temperature decreased from ambient to liquid nitrogen to liquid hydrogen. Apparently, the rapidly diminishing rate of change in friction forces above certain pressures is caused by a change in the pivot point from the bend (lip-to-flange intersection) radius of the cross-section to the contact point between the support ring and the seal lip. The effective "pressurizing" area changes, with an associated change in normal loading, with pressure. Also, most of the resultant pressure load is transmitted into the support ring and at pressures above the peak friction point tends to reduce the seal-to-sleeve normal contact load. The shift in the pressure, at which peak friction occurs with lower temperature, is aided by the differential contraction of the lipseal and support ring which reduces the clearance between these two parts. Therefore, contact is made between the seal lip and support ring at lower pressures because of the reduced travel necessary to make contact between these parts. Friction forces increased at liquid nitrogen temperature and further increased at liquid hydrogen temperature as was expected because of the increased normal load resulting from differential contractions between the seal and sleeve materials. A typical example of friction forces at various pressures and temperatures for the "supported" lipseal also is shown on Figure 135.

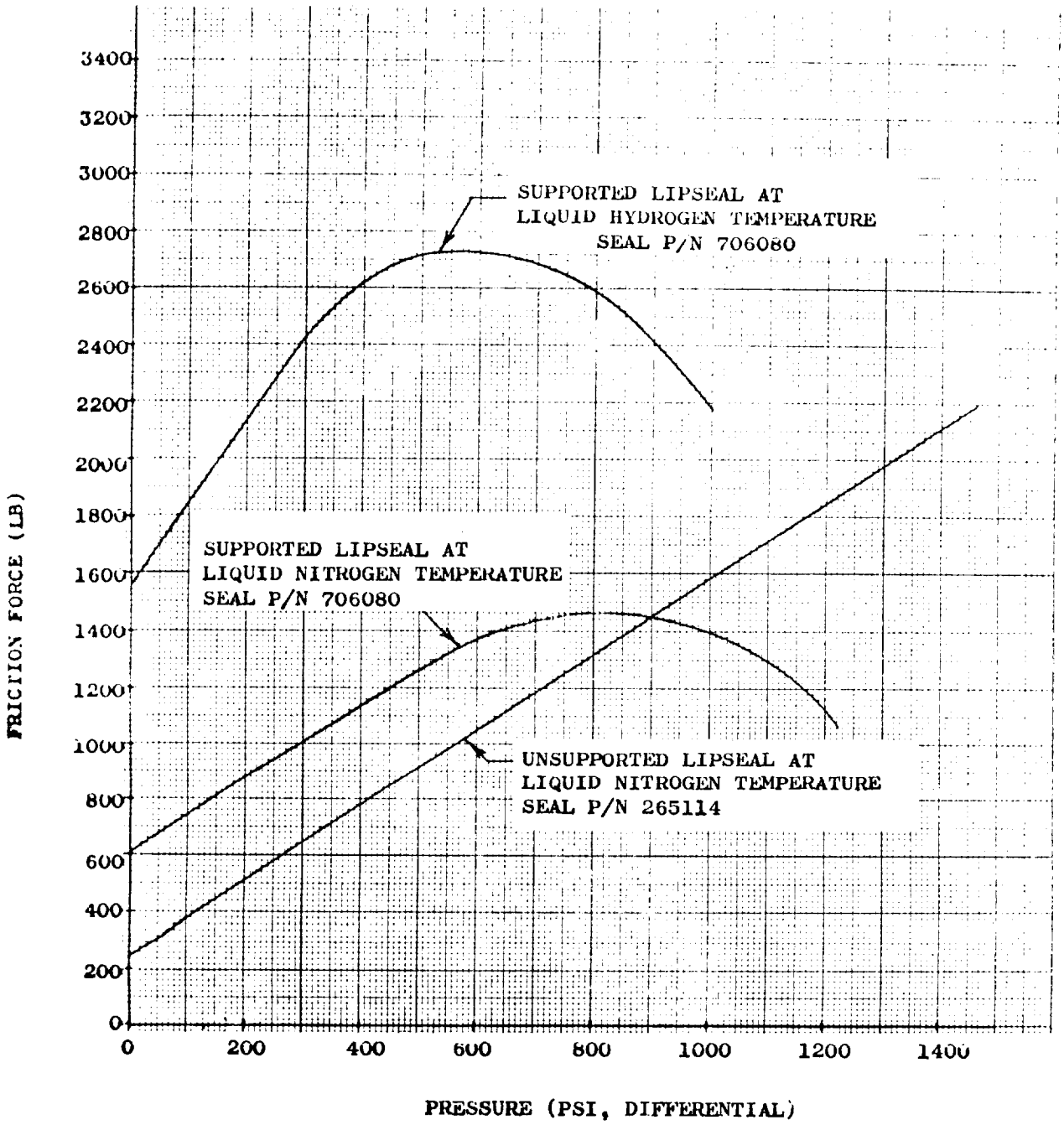


Figure 135
Friction vs. Pressure at Various Temperatures, 11-0-in. Lipseal

Friction forces for the "supported" lipseal were greater at low pressure than for the "unsupported" lipseal. However, because of the diminishing friction characteristic of the "supported" lipseal, the friction forces are considerably less for the "supported" lipseal above 1000 psig at liquid nitrogen temperature. Friction is approximately equal for both configurations at approximately 900 psig and liquid nitrogen temperature.

The increased friction and the increased rate of change in friction at liquid hydrogen conditions over those at liquid nitrogen condition also is of interest. It was theorized that friction would not increase as much as would be expected based upon differential contraction because liquid hydrogen would serve as a lubricant. The results of these tests indicate that this may not be true, at least with the material (Kel-F and hard-faced Inconel-X) involved here.

e. Burst Tests

Burst tests were performed with the large diameter lipseal designs in the test cells at liquid nitrogen temperatures. The "unsupported" lipseal failed at pressures of 2400 psig and above. The expected failure pressure (based upon calculations and previous data obtained by test) was 2300 psig. No failures occurred at design burst pressure of 2200 psig. Failures were the result of breaking at (or near) the bend radius of the lipseal cross-section. A typical example is shown on Figure 136.

Six separate attempts were made to fail the "supported" lipseal at liquid nitrogen temperature. The maximum capacity of the system was 2920 psig. No lipseal failures occurred at the maximum system pressure. After the burst test, the test cell was actuated and the system operated normally. Leakage from 0 psig to 1000 psig pressure was excessive, but was not considered abnormal after the burst test. No crazing or cracking was experienced on any of the seals tested. A typical example of the seals and support ring after these tests is shown on Figure 137. Upon the basis of these tests, the failure point of the "supported" lipseal design was expected to be well over 3000 psig at liquid nitrogen temperatures.

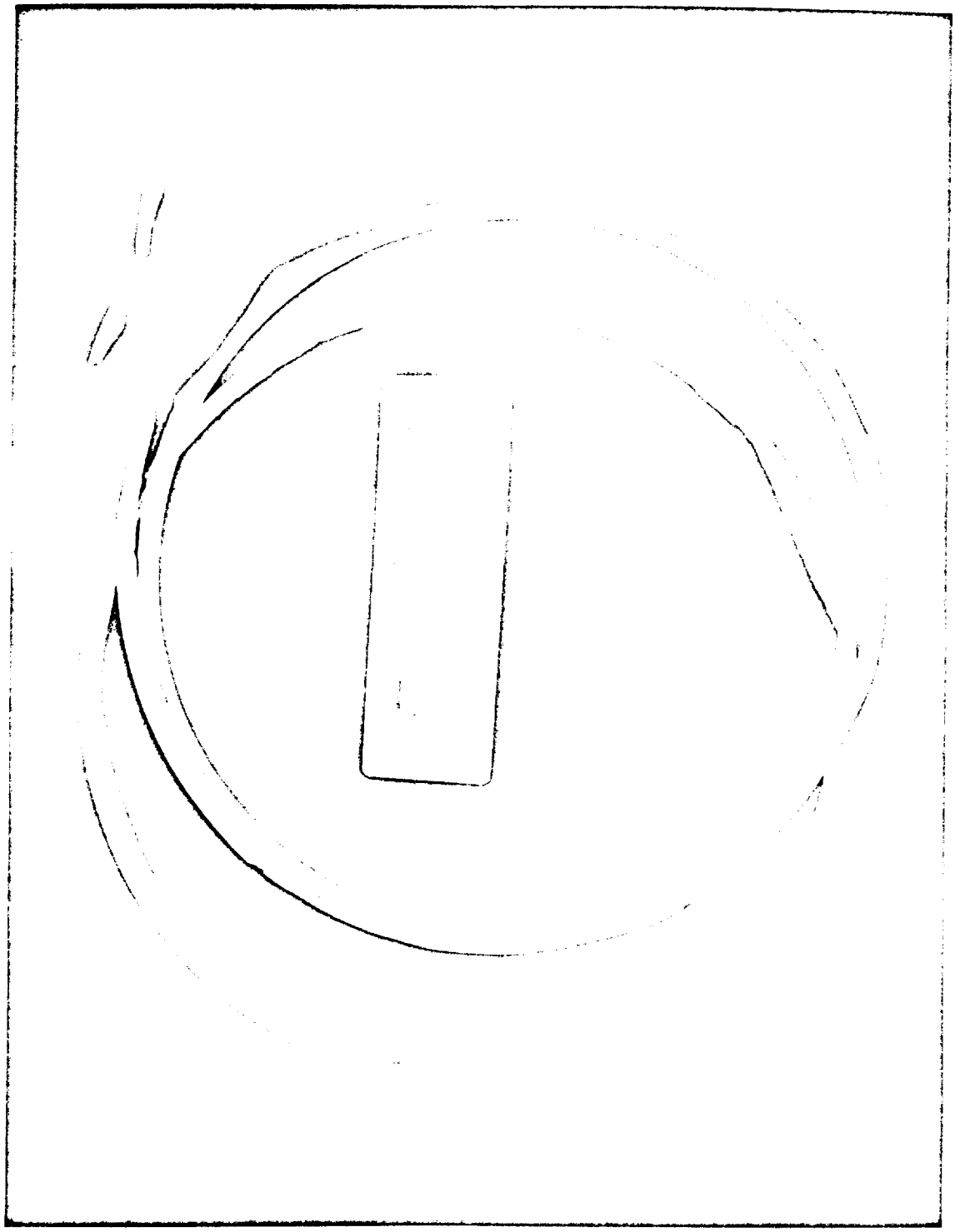


Figure 136
Typical "Unsupported" Lipseal Failure at
2400 psig Liquid Nitrogen Burst Pressure

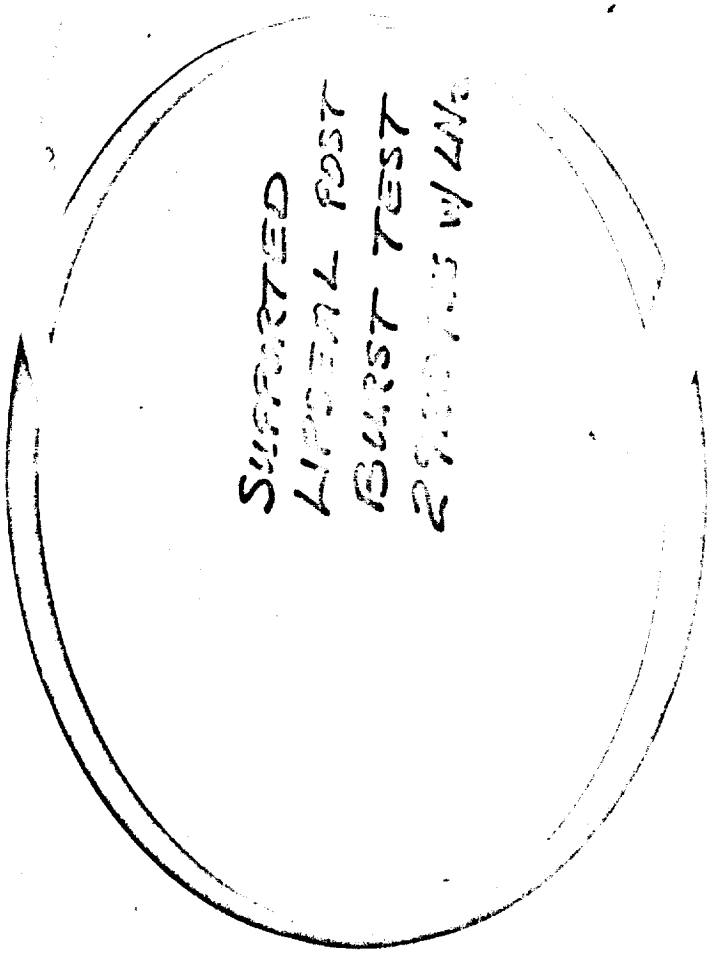


Figure 137
Typical "Supported" Lipseal After 2920 psig Burst Test

D. CONCLUSIONS

The lipseals provided more satisfactory and reliable service than the Omniseal for the liquid hydrogen and liquid oxygen applications of the M-1 gas generator valve dynamic seals. Static and dynamic friction loads were smaller for the Omniseals, but leakage and wear rates were much lower for the lipseals. Assembly problems and rolling failures were the usual difficulties encountered with the Omniseals and these contributed to leakage rates in excess of the maximum permissible values for this application. Lipseals must be designed to eliminate "cracking" failures. The "cracking" problems encountered with the lipseal in one area of the M-1 gas generator oxidizer valve were attributed to inadequate support and/or seal strength at the lip-to-flange intersection of the seal. Excessive leakages were always associated with galling of the shaft sealing surface because of contact with the mating clearance diameters, or overstroking (caused by improper settings) of the valve shaft.

Lipseals for large diameter shaft applications can be made to perform satisfactorily in liquid hydrogen and liquid oxygen systems. Careful selection of materials, seal cross-section, interference, and other machining factors are required. The selected, specially-designed lipseal for the 11.0-in. (M-1 thrust chamber valve) sleeve application performed satisfactorily, but it appears that lipseal designs need special design consideration for each application. The "supported" lipseal design performed better than the "unsupported" design. Each design ultimately is a compromise between friction, wear rate, leakage rate, and other factors depending upon the application and requirements. Considerable additional testing would be required to develop methods of analysis for other applications and requirements.

E. RECOMMENDATIONS

The lipseal design is recommended for "contact" seal applications with reciprocating shafts in liquid hydrogen and liquid oxygen systems. The lipseal design has also been successfully applied in nitrogen tetroxide and AeroZINE-50 [R] systems (Titan II and III).

Additional testing should be performed with the lipseal designs as well as the other types of seals for the primary purpose of developing methods for analysis and standardization of the various types of seals. The statistical testing plan originally proposed for dynamic seals development during the M-1 Program was devised to gain information for analysis and standardization. Because almost all of the hardware required for this program was fabricated, it could be used for such an evaluation program without the usual time necessary for procurement of long-lead hardware.

Standardization of the machining requirements (grooves) for some types of seals is also recommended. Groove standardization would help to eliminate many of the machining variables in the dynamic joints. Also, less

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work would be required for the development of the various types of "contact" seals for many applications. With the current technology, it is difficult to determine just what is the best machining requirement for each of the various types of seals. However, for the types requiring grooves similar to the O-ring grooves, machining requirements could be based upon MIL-P-5514 as a starting point. Standards for lipseals and the many other configurations could eventually be made if sufficient methods for analysis and testing were available for the many applications. Suppliers of these types could then develop seals for the many applications with known machining requirements. Specifications and part standards could then be made for the mating seals based upon pressures, operating fluids, temperature range, surface speeds, and other factors. Known values of friction, wear rates, and other factors could then be supplied to the designer for a particular application.

The seal and groove standards also suggest the necessity for improvements and possible standardization of test procedures as well as test methods. Standardized test procedures and test methods would provide a better basis for comparison while supplying additional information for the development of methods for analysis of the various applications and types of seals. A very good literature survey and some preliminary analysis methods have been started(51)(52) but considerably more work needs to be done in this area. The real problem is that no valid basis for comparison of the various types of seals for the various applications is available.

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(51) Final Report, Contract NAS 7-102, February 1962 to February 1963, op. cit.
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APPENDIX A

SYMBOLS AND NOMENCLATURE

A_e = Exposed area of gasket (in.^{2})

A_g = Gasket stress area (in.^{2})

a_b = Stress area of each bolt (in.^{2})

B = Bolt factor ($\frac{psi}{lb/in.}$)

D_{68} = Diameter at 68°F (in.)

D_{gi} = Gasket I.D. (in.)

D_{go} = Gasket O.D. (in.)

D_c = Minimum permissible (equivalent) flange diameter for bolt clearance holes (in.)

D_{bc} = Bolt circle diameter (in.)

D_F = Flange O.D. (in.)

D_i = Flange (or tube) I.D. (in.)

D_p = Mean diameter of primary gasket (in.) (or mean sealing surface diameter primary gasket)

D_s = Mean diameter of secondary gasket (in.) (or mean sealing surface diameter secondary gasket)

D_t = Tube O.D. (in.)

d_b = Bolt diameter (in.)

d_c = Bolt clearance hole diameter (in.)

ΔD_T = Change in diameter caused by temperature changes (in.)

ΔD_P = Change in diameter caused by pressure (in.)

E = Modulus of elasticity (psi)

F = Allowable stress (psi) (usually = minimum material yield strength)

F_g = Required initial gasket seating stress (psi)

F_{gm} = Minimum permissible gasket seating stress after applying initial gasket seating stress (psi)

F_b = Allowable combined stress in the bolt (psi)

f_g = Calculated gasket seating stress (psi)

f_B = Calculated bending stress (psi)

f_T = Calculated tensile stress (psi)

h_f = Female flange thickness (in.)

h_m = Male flange thickness (in.)

h_{min_F} = Minimum required flange thickness based on permissible stress (psi)

h_{min_δ} = Minimum required flange thickness based on permissible deflection (in.)

K_P = Permeation factor at operating temperature (cc/100 in.^{2}/mil/atm-day)

K_T = Coefficient of thermal expansion (in./in. @ maximum temperature excursion from 68°F)

W_A = Design load caused by externally applied axial tension (lb)

W_B = Design load caused by external bending moments (lb)

W_b = Total axial load supplied by each bolt at specified torque (lb)

W_g = Total required gasket seating load (lb)

W_P = Design load caused by internal pressure (psi)

W_T = Total design load (lb)

w_A = Design load intensity caused by externally applied axial tension (lb/in.)

w_B = Design load intensity caused by externally applied bending moment (lb/in.)

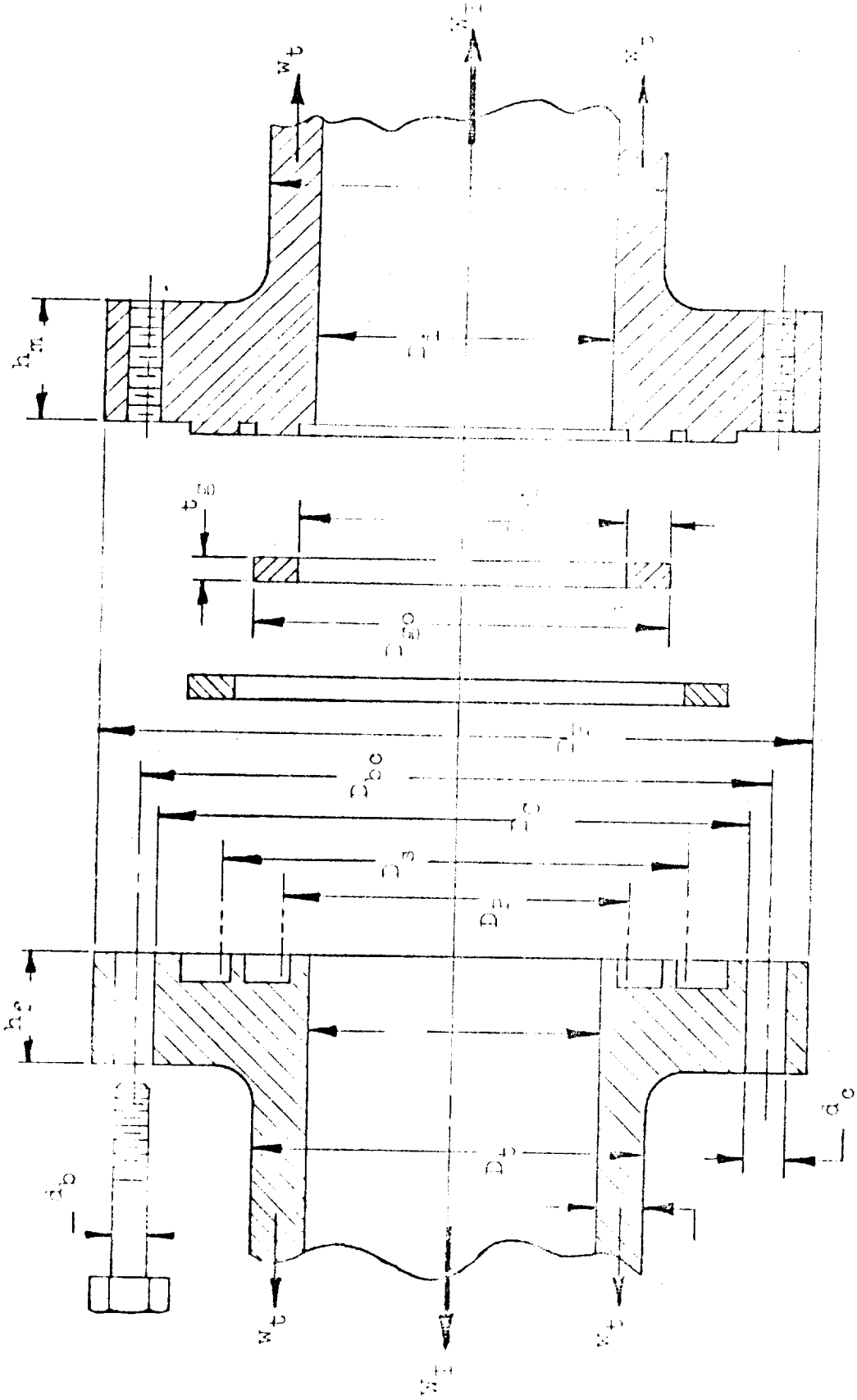
w_b = Axial load in the bolt for one in.-lb torque (lb/in.-lb)

w_g = Required gasket seating load intensity (lb/in.)

w_P = Design load intensity caused by internal pressure (lb/in.)

Appendix A

w_T	=	Total design load intensity (lb/in.)
ℓ	=	Length of leak path in direction of leakage flow = nominal thickness of gasket exposed to fluid (in.)
M	=	Externally applied bending moment (in.-lb)
m_o	=	Flange section bending moment intensity (in.-lb/in.) at the junction of the tube and flange
n	=	Number of bolts
P	=	Proof pressure
Q_P	=	Total leakage from permeation (cc/sec)
S_b	=	Calculated combined stress in the bolt at specified torque (psi)
s_b	=	Combined stress in bolt for one in.-lb torque (psi/in.-lb)
t	=	Tube wall thickness (in.)
t_g	=	Gasket thickness or free height (in.)
t_j	=	Nominal thickness of the jacket exposed to the system fluid (in.)
T_{max}	=	Maximum permissible torque on bolts (in.-lb)
T_{min}	=	Minimum specified torque on bolts (in.-lb) (assumed = .9 T_{max})
δ	=	Flange deflection (in.)
δ_a	=	Allowable flange deflection (in.)



APPENDIX B

DERIVATION AND USE OF BOLT FACTOR "B"

The maximum permissible bolt spacing is dependent upon the stiffness of flange(s) between the bolts. The flange(s) must be sufficiently stiff to prevent excessive deflections ("bowing") between bolts so that the minimum permissible gasket seating stresses are maintained when operating loads are applied. Bolt spacing for heavy duty flanges (i.e., Boiler Code Flanges with safety factor of 8 to 10) can be quite large because the flanges are stiff and flange deflections are minute. With larger spacing, a lesser number of large bolts is required and this is an advantage with regard to assembly time and effort. However, in flightweight joint designs, the optimum bolt spacing is one that permits the smallest flange thickness (stiffness) and smallest flange diameter without excessive deflection ("bowing") between bolts when operating (or design) loads are applied. In general, the flanges will be lighter with the smallest bolt spacing, but, in practice, this is limited by bolt and head and nut diameters as well as space requirements for tools used to tighten the bolts or nuts. For preliminary work, the most practical limit is approximately three times the bolt diameter; therefore, the number of bolts required for this case is a function of the bolt circle diameter (D_{bc}) and the bolt diameter (d_b) or:

$$n = \frac{\pi \; D_{b.c.}}{3 \; d_b} \qquad \text{Eq. (1)}$$

(Appendix A is a listing of the symbols and nomenclature).

The most important consideration in selecting the size and number of bolts is to assure that the required axial tension load is attained during tightening without overstressing the bolts. Table V in the text of this report provides maximum combined stress (accurate within a few percentiles) and the axial load that can be attained (80% to 90%) with various sized bolts and screws for each inch-pound of torque applied to the nut. The maximum permissible torque (T_{max}) is a function of the permissible combined stress (F_b) on the bolt or:

$$T_{max} = \frac{F_b}{s_b} \qquad \text{Eq. (2)}$$

The minimum torque (T_{min}) specified in the assembly procedures is usually approximately 90% of the maximum torque or:

$$T_{min} = .9 \; T_{max} \qquad \text{Eq. (3)}$$

If the nominal values of load per each inch-pound torque (w_b) presented in Table V is used for preliminary design work, the minimum axial load attained in each bolt would be:

$$W_b = T_{min} \times w_b \qquad \text{Eq. (4)}$$

The total restraining (or clamping) axial load supplied by all the bolts ($n \times W_b$) must be equal to or greater than the design separating loads induced in the flange:

$$n\, W_b = W_T \qquad \text{Eq. (5)}$$

Therefore, the number of bolts required is:

$$n = \frac{W_T}{W_b} \qquad \text{Eq. (6)}$$

Combining Equations (2), (3), and (4) and solving for W_b:

$$W_b = .9\, \frac{F_b}{s_b}\, w_b \qquad \text{Eq. (7)}$$

By substitution in Equation (6), the number of bolts required based upon permissible combined stress from torque on the bolts is:

$$n = \frac{W_T\ s_b}{.9\ F_b\ w_b} \qquad \text{Eq. (8)}$$

Equations (1) and (8) are approximate equalities because the number of bolts required for the best practical bolt spacing must be approximately equal to the number of bolts required, based upon permissible combined stresses from applied torque on the bolts. Therefore:

$$n = \frac{\pi\ Db.c.}{3\ d_b} \cong \frac{W_T\ s_b}{.9\ F_b\ w_b} \qquad \text{Eq. (9)}$$

By rearranging the terms, all of the known factors concerning the bolt size and threads can be equated to factors pertaining to the flange loads and configuration, and the combined stresses permissible with the selected bolt material.

$$d_b\, \frac{s_b}{w_b} = \frac{.942\ Db.c.\ \ F_b}{W_T} \qquad \text{Eq. (10)}$$

The known terms pertaining to the bolt (left side of the equation) provide a different constant for each bolt size and thread type. This constant is called the bolt factor "B":

$$B = d_b \frac{s_b}{w_b} \qquad \text{Eq. (11)}$$

The bolt factor is approximately equal to the other terms in Equation (10) pertaining to the flange (design) loads, bolt circle diameter, and the combined stresses permissible with the selected bolt material:

$$B \cong \frac{.942\ Db.c.\ F_b}{W_T} \qquad \text{Eq. (12)}$$

The bolt factor can then be estimated from Equation (12) after the total flange design loads have been determined, the bolt material selected, and the preliminary layout has progressed to the point where the bolt circle diameter can be closely estimated. From this, the size and thread type can be selected from Table V in the text of this report. This size and thread type should closely satisfy the requirement for reasonable bolt spacing and the permissible combined stresses attained during tightening of the bolt.

The minimum number of bolts required can be calculated after the bolt size and thread configuration have been selected by using the actual bolt factor and bolt diameter (d_b) given in Table V and Equation (14). The equation is derived from Equation (8) as follows:

$$\text{If} \quad B = d_b \frac{s_b}{w_b}, \text{ then } \quad s_b = \frac{B\ w_b}{d_b} \qquad \text{Eq. (13)}$$

Substituting in Equation (8) and simplifying:

$$n = \frac{W_T\ B}{.9\ F_b\ d_b} \qquad \text{Eq. (14)}$$

The number of bolts used should never be less than the number calculated from the formula. The torques to be specified can be calculated from Equations (2) and (3).

This method of bolt selection affords a reasonable starting point in the preliminary design. However, the torque-axial tension load relationships must be determined by actual tests with the flange and bolt materials as well as the lubricants to be used. These values must be used in the detailed analysis before making the joint design final. The bolt stresses calculated by this method is reasonably accurate providing there is no bending induced in the bolt during assembly or operation of the joint.

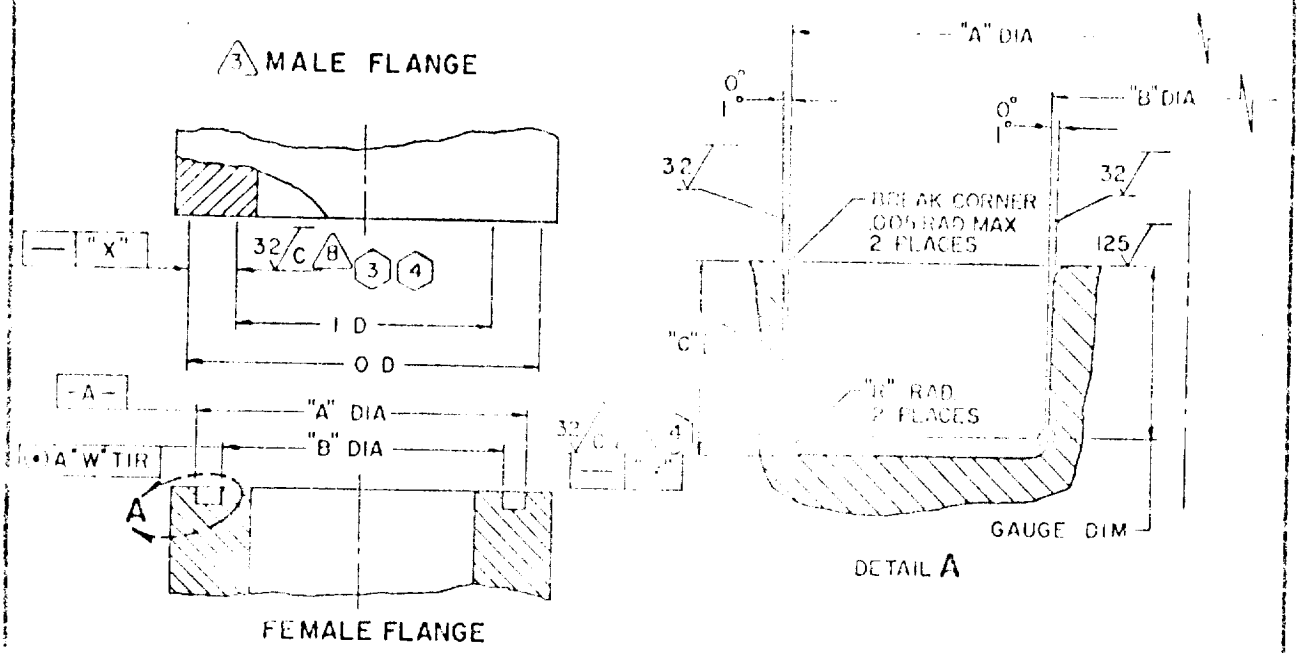
APPENDIX C

Aerojet-General Corporation

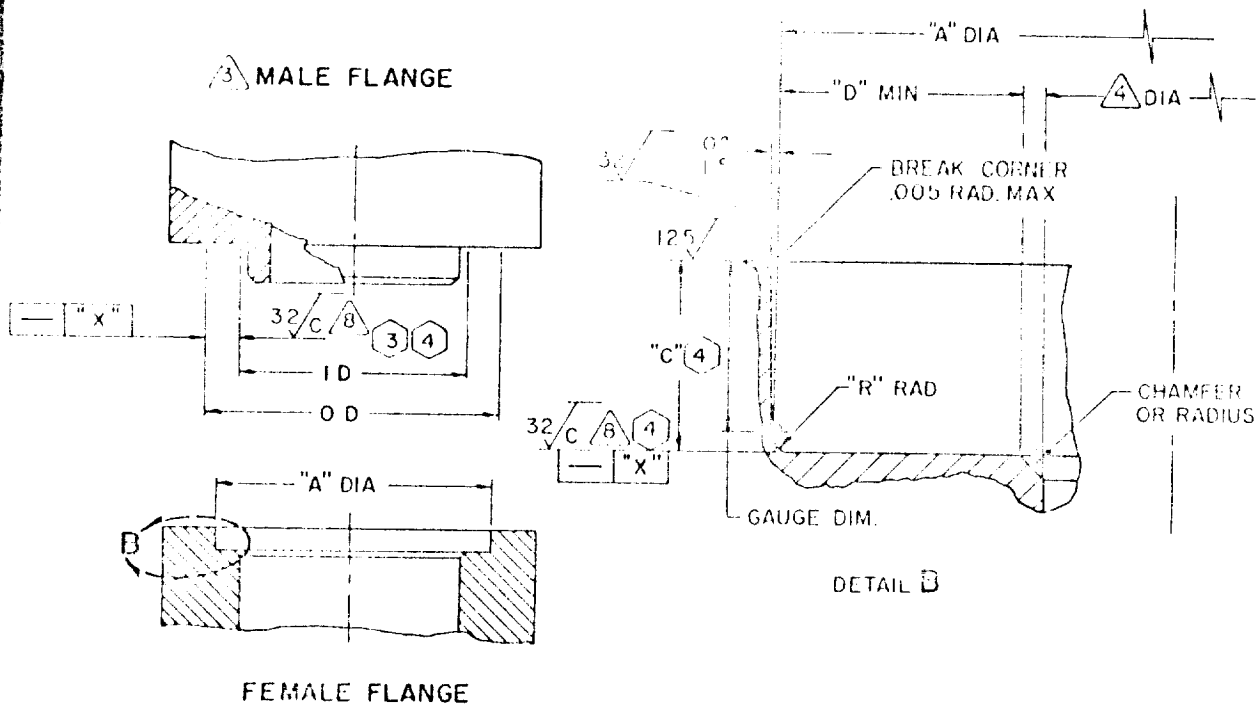
DESIGN STANDARDS FOR

STATIC FACE SEAL GROOVES

(ASD5232 and ASD5233)



STYLE A



STYLE B

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TABLE I

STATIC FACE SEAL GROOVE DIMENSIONS
NOMINAL .094 CROSS-SECTION

DASH NO.	NOM GLAND O.D.	A DIA.	A TOL.	B DIA.	B TOL.	C	D MIN	R RAD MAX.	CONC W	FLATNESS X	GAUGE DIM.
1000	.562	.562		.242							
1100	.625	.625		.305							
1110	.687	.687		.367							
1120	.750	.750		.430							
1130	.812	.812		.492							
1140	.875	.875		.555							
1150	.937	.937		.617							
1160	1.000	1.000		.680							
1170	1.062	1.062		.742							
1180	1.125	1.125		.805							
1190	1.187	1.187		.867							
1200	1.250	1.250		.930							
1210	1.312	1.312		.992							
1220	1.375	1.375		1.055							
1230	1.437	1.437		1.117							
1240	1.500	1.500		1.180							
1250	1.562	1.562	+.002	1.242	+.000	.088	.145	.015	.002	.001	.062
1260	1.625	1.625	-.000	1.305	-.005	.090					
1270	1.687	1.687		1.367							
1280	1.750	1.750		1.430							
1290	1.812	1.812		1.492							
1300	1.875	1.875		1.555							
1310	1.937	1.937		1.617							
1320	2.000	2.000		1.680							
1330	2.062	2.062		1.742							
1340	2.125	2.125		1.805							
1350	2.187	2.187		1.867							
1360	2.250	2.250		1.930							
1370	2.312	2.312		1.992							
1380	2.375	2.375		2.055							
1390	2.437	2.437		2.117							
1400	2.500	2.500		2.180							
1410	2.562	2.562		2.242							
1420	2.625	2.625		2.305							
1430	2.687	2.687		2.367							
1440	2.750	2.750		2.430							
1450	2.812	2.812		2.492							
1460	2.875	2.875		2.555							
1470	2.937	2.937		2.617							
1480	3.000	3.000		2.680							

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TABLE II

STATIC FACE SEAL GROOVE DIMENSIONS
NOMINAL .125 CROSS-SECTION

DASH NO.	NOM GLAND O.D.	A DIA.	A TOL.	B DIA.	B TOL.	C	D MIN	R RAD MAX.	CONC W	FLATNESS X	GAUGE DIM.
[illegible]	1.000	1.000		[illegible]							
[illegible]	[illegible]	1.062		[illegible]							
[illegible]	[illegible]	[illegible]		[illegible]							
[illegible]	[illegible]	1.187		[illegible]							
[illegible]	[illegible]	1.250		[illegible]							
[illegible]	[illegible]	1.312		[illegible]							
[illegible]	[illegible]	1.375		[illegible]							
[illegible]	[illegible]	1.437		1.00[illegible]							
[illegible]	[illegible]	1.500		1.06[illegible]							
[illegible]	[illegible]	1.562	+.002	1.1[illegible]							
[illegible]	[illegible]	1.625	-.000	1.1[illegible]	+.000						
[illegible]	1.687	1.687		1.25[illegible]	-.005						
2028	1.750	1.750		1.3[illegible]							
[illegible]	1.875	1.875		1.4[illegible]							
[illegible]	2.000	2.000		1.570							
[illegible]	2.125	2.125		1.6[illegible]							
[illegible]	2.250	2.250		1.8[illegible]							
[illegible]	2.375	2.375		1.24[illegible]							
[illegible]	2.500	2.500		2.070							
[illegible]	2.625	2.625		[illegible]							
[illegible]	2.750	2.750		2.320							
[illegible]	2.875	2.875		2.445							
[illegible]	3.000	3.000		2.570		.121 .123	.195	.020	.003	.002	.090
[illegible]	3.125	3.125		2.695							
[illegible]	3.250	3.250		2.820							
[illegible]	3.375	3.375		2.945							
[illegible]	3.500	3.500		3.070							
[illegible]	3.625	3.625		3.195							
[illegible]	3.750	3.750	+.004	3.320	+.000						
[illegible]	3.875	3.875	-.000	3.445	-.006						
[illegible]	4.000	4.000		3.570							
[illegible]	4.125	4.125		3.695							
[illegible]	4.250	4.250		3.820							
[illegible]	4.375	4.375		3.945							
[illegible]	4.500	4.500		4.070							
[illegible]	4.625	4.625		4.195							
[illegible]	4.750	4.750		4.320							
[illegible]	4.875	4.875		4.445							
2080	5.000	5.000		4.570							
2082	5.125	5.125		4.695							
[illegible]	5.250	5.250		4.820							
2086	5.375	5.375		4.945							
2088	5.500	5.500		5.070							
2090	5.625	5.625		5.195							
2092	5.750	5.750	+.006	5.320	+.000						
2094	5.875	5.875	-.000	5.445	-.008						
2096	6.000	6.000		5.570							
2098	6.125	6.125		5.695							
2100	6.250	6.250		5.820							
2102	6.375	6.375		5.945							
2104	6.500	6.500		6.070							
2106	6.625	6.625		6.195							

TABLE CONTINUED ON FOLLOWING SHEET

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TABLE II (CONT'D)

DASH NO.	NOM GLAND O.D.	A DIA.	A TOL.	B DIA.	B TOL.	C	D MIN	R RAD MAX.	CONC W	FLATNESS X	GAUGE DIM.
-108	6.750	6.750		6.320							
-110	6.875	6.875		6.445							
-112	7.000	7.000		6.570							
-114	7.125	7.125		6.695							
-116	7.250	7.250	+.005	6.820	+.000						
-118	7.375	7.375	-.000	6.945	-.005						
-120	7.500	7.500		7.070							
-122	7.625	7.625		7.195							
-124	7.750	7.750		7.320							
-126	7.875	7.875		7.445							
-128	8.000	8.000		7.570							
-130	8.125	8.125		7.695							
-132	8.250	8.250		7.820							
-134	8.375	8.375		7.945							
-136	8.500	8.500		8.070							
-138	8.625	8.625		8.195							
-140	8.750	8.750		8.320							
-142	8.875	8.875		8.445							
-144	9.000	9.000		8.570							
-146	9.125	9.125	+.006	8.695	+.000						
-148	9.250	9.250	-.000	8.820	-.010						
-150	9.375	9.375		8.945							
-152	9.500	9.500		9.070							
-154	9.625	9.625		9.195		.121	.195	.020	.003	.002	.090
-156	9.750	9.750		9.320		.123					
-158	9.875	9.875		9.445							
-160	10.000	10.000		9.570							
-162	10.125	10.125		9.695							
-164	10.250	10.250		9.820							
-166	10.375	10.375		9.945							
-168	10.500	10.500		10.070							
-170	10.625	10.625		10.195							
-172	10.750	10.750		10.320							
-174	10.875	10.875		10.445							
-176	11.000	11.000		10.570							
-178	11.125	11.125		10.695							
-180	11.250	11.250		10.820							
-182	11.375	11.375		10.945							
-184	11.500	11.500		11.070							
-186	11.625	11.625		11.195							
-188	11.750	11.750	+.008	11.320	+.000						
-190	11.875	11.875	-.000	11.445	-.012						
-192	12.000	12.000		11.570							
-194	12.125	12.125		11.695							
-196	12.250	12.250		11.820							
-198	12.375	12.375		11.945							
-200	12.500	12.500		12.070							
-202	12.625	12.625		12.195							
-204	12.750	12.750		12.320							
-206	12.875	12.875		12.445							
-208	13.000	13.000		12.570							
-210	13.125	13.125		12.695							
-212	13.250	13.250		12.820							

TITLE
GROOVE DIMENSIONS-STATIC FACE SEAL, INTERNAL PRESSURE

AEROJET STANDARD
ASD5232
SHEET 4 OF 15

ISSUED 23 DEC 64 REVISED

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TABLE III

STATIC FACE SEAL GROOVE DIMENSIONS
NOMINAL .250 CROSS-SECTION

DASH NO.	NOM GLAND O.D.	A DIA.	A TOL.	B DIA.	B TOL.	C	D MIN	R RAD MAX.	CONC W	FLATNESS X	GAUGE DIM.
4128	8.000	8.000		7.[illegible]							
4132	8.[illegible]	8.250		7.[illegible]							
4136	8.500	8.500		7.[illegible]							
4140	8.750	8.750		7.[illegible]							
4144	9.000	9.000		8.[illegible]							
4148	9.250	9.250		8.[illegible]							
4152	9.500	9.500		8.[illegible]							
4156	9.750	9.750		8.[illegible]							
4160	10.000	10.000		9.[illegible]							
4164	10.250	10.250		9.[illegible]							
4168	10.500	10.500		9.[illegible]							
4172	10.750	10.750	+.010	9.[illegible]							
4176	11.000	11.000	-.000	10.[illegible]							
4180	11.250	11.250		10.[illegible]							
4184	11.500	11.500		10.660							
4188	11.750	11.750		10.910							
4192	12.000	12.000		11.160							
4196	12.250	12.250		11.410							
4200	12.500	12.500		11.660							
4204	12.750	12.750		11.[illegible]							
4208	13.000	13.000		12.[illegible]							
4212	13.250	13.250		12.410							
4216	13.500	13.500		12.[illegible]	.00	.238					
4220	13.750	13.750		12.910	-.030	.240	.370	.030	.020	.065	.195
4224	14.000	14.000		13.160							
4228	14.250	14.250		13.410							
4232	14.500	14.500		13.[illegible]							
4236	14.750	14.750		13.[illegible]							
4240	15.000	15.000		13.[illegible]							
4244	15.250	15.250		14.410							
4248	15.500	15.500		14.660							
4252	15.750	15.750		14.910							
4256	16.000	16.000		15.160							
4260	16.250	16.250		15.410							
4264	16.500	16.500		15.660							
4268	16.750	16.750		15.910							
4272	17.000	17.000		16.[illegible]							
4276	17.250	17.250		16.[illegible]							
4280	17.500	17.500		17.[illegible]							
4284	17.750	17.750		17.[illegible]							
4288	18.000	18.000		17.160							
4292	18.250	18.250		17.410							
4296	18.500	18.500	+.015	17.660							
4300	18.750	18.750	-.000	17.910							
4304	19.000	19.000		18.[illegible]							
4308	19.250	19.250		18.410							
4312	19.500	19.500		18.660							
4316	19.750	19.750		18.910							
4320	20.000	20.000		19.160							
4324	20.250	20.250		19.[illegible]							
4328	20.500	20.500		19.[illegible]							
4332	20.750	20.750		19.[illegible]							
4336	21.000	21.000		20.[illegible]							

TITLE
GROOVE DIMENSIONS
INTERPLANE

ISSUED 23 DEC 64 REVISED

TABLE III (CONT'D)

DASH NO.	NOM GLAND O.D.	A DIA.	A TOL.	B DIA.	B TOL.	C	D MIN.	R RAD MAX.	CONC W	FLATNESS X	GAPO DIM.
[illegible]	21.250	21.250		[illegible]							
[illegible]	21.500	21.500		[illegible]							
[illegible]	21.750	21.750		[illegible]							
[illegible]	22.000	22.000		21.160							
[illegible]	22.250	22.250		21.410							
[illegible]	22.500	22.500		[illegible]							
[illegible]	22.750	22.750		[illegible]							
[illegible]	23.000	23.000	+.015	[illegible]							
[illegible]	23.250	23.250	-.000	22.410							
[illegible]	23.500	23.500		[illegible]							
[illegible]	23.750	23.750		22.910							
[illegible]	24.000	24.000		23.160							
[illegible]	24.250	24.250		23.410							
[illegible]	24.500	24.500		[illegible]							
[illegible]	24.750	24.750		23.910							
[illegible]	25.000	25.000		24.160							
[illegible]	25.250	25.250		24.410							
[illegible]	25.500	25.500		24.660							
[illegible]	25.750	25.750		24.910	+.000	.238	.390	.030	.020	.005	.195
[illegible]	26.000	26.000		25.160	-.030	.240					
[illegible]	26.250	26.250		25.410							
[illegible]	26.500	26.500		25.660							
[illegible]	26.750	26.750		25.910							
[illegible]	27.000	27.000		26.160							
[illegible]	27.250	27.250		26.410							
[illegible]	27.500	27.500		26.660							
[illegible]	27.750	27.750		26.910							
[illegible]	28.000	28.000		27.160							
[illegible]	28.250	28.250		27.410							
[illegible]	28.500	28.500		27.660							
[illegible]	28.750	28.750	+.020	27.910							
[illegible]	29.000	29.000	-.000	28.160							
[illegible]	29.250	29.250		28.410							
[illegible]	29.500	29.500		28.660							
[illegible]	29.750	29.750		28.910							
[illegible]	30.000	30.000		29.160							
[illegible]	30.250	30.250		29.410							
[illegible]	30.500	30.500		29.660							
[illegible]	30.750	30.750		29.910							
[illegible]	31.000	31.000		30.160							
[illegible]	31.250	31.250		30.410							
[illegible]	31.500	31.500		30.660							
[illegible]	31.750	31.750		30.910							
[illegible]	32.000	32.000		31.160							
[illegible]	32.250	32.250		31.410							
[illegible]	32.500	32.500		31.660							
[illegible]	32.750	32.750		31.910							
[illegible]	33.000	33.000		32.160							
[illegible]	33.250	33.250		32.410							
[illegible]	33.500	33.500		32.660							

TABLE CONTINUED ON FOLLOWING SHEET

TITLE

GROOVE DIMENSIONS-STATIC FACE SEAL, INTERNAL PRESSURE

AEROJET STANDARD

ASD5232

SHEET [illegible] OF [illegible]

REVISED 28 DEC 64

TABLE III (CONT'D)

DASH NO.	NOM GLAND O.D.	A DIA.	A TOL.	B DIA.	B TOL.	C	D MIN.	R RAD MAX.	CONC W	FLATNESS X	GAUGE DIM.
[illegible]	33.750	33.750		32.910							
[illegible]	34.000	34.000		33.160							
[illegible]	34.250	34.250		33.410							
[illegible]	34.500	34.500		33.660							
[illegible]	34.750	34.750		33.910							
[illegible]	35.000	35.000		[illegible]							
[illegible]	35.250	35.250		[illegible]							
[illegible]	35.500	35.500		[illegible]							
[illegible]	35.750	35.750		[illegible]							
[illegible]	36.000	36.000		[illegible]							
[illegible]	36.250	36.250		35.410							
[illegible]	36.500	36.500		35.660							
[illegible]	36.750	36.750		35.910							
[illegible]	37.000	37.000		36.160							
[illegible]	37.250	37.250		36.410							
[illegible]	37.500	37.500		[illegible]							
4604	37.750	37.750		36.910							
4608	38.000	38.000		37.160							
4612	38.250	38.250		37.410							
4616	38.500	38.500		[illegible]							
4620	38.750	38.750		37.910							
[illegible]	39.000	39.000	+.020	[illegible]	+.000	.238	.390	.030	.020	.005	.195
[illegible]	39.250	39.250	-.000	[illegible]	-.030	.240					
[illegible]	39.500	39.500		[illegible]							
4636	39.750	39.750		38.910							
[illegible]	40.000	40.000		39.160							
[illegible]	40.250	40.250		39.410							
[illegible]	40.500	40.500		39.660							
[illegible]	40.750	40.750		39.910							
[illegible]	41.000	41.000		40.160							
[illegible]	41.250	41.250		40.410							
[illegible]	41.500	41.500		40.660							
[illegible]	41.750	41.750		40.910							
[illegible]	42.000	42.000		41.160							
[illegible]	42.250	42.250		41.410							
[illegible]	42.500	42.500		41.660							
[illegible]	42.750	42.750		41.910							
[illegible]	43.000	43.000		42.160							
[illegible]	43.250	43.250		42.410							
[illegible]	43.500	43.500		42.660							
4700	43.750	43.750		42.910							
4704	44.000	44.000		43.160							
[illegible]	44.250	44.250		43.410							
4712	44.500	44.500		43.660							
4716	44.750	44.750		43.910							
4720	45.000	45.000		44.160							
4724	45.250	45.250		44.410							
[illegible]	45.500	45.500		44.660							
[illegible]	45.750	45.750		44.910							
4736	46.000	46.000		45.160							
4740	46.250	46.250		45.410							
4744	46.500	46.500		45.660							

TABLE CONTINUED ON FOLLOWING SHEET

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ISSUED 2 DEC 64 REVISED

CODE IDENT NUMBER 05824

TITLE
GROOVE DIMENSIONS-STATIC FACE SEAL, INTERNAL PRESSURE

AEROJET STANDARD
ASD5232
SHEET 7 OF 15

TABLE III (CONT'D)

BASIC NO.	NOM GLAND O.D.	A DIA.	A TOL.	B DIA.	B TOL.	C	D MIN	R RAD MAX.	CONC W	FLATNESS X	GAUGE DIM
4713	46.750	46.750		45.910							
[illegible]	47.000	[illegible]		46.160							
[illegible]	47.250	47.250		46.410							
[illegible]	[illegible]	47.500		[illegible]							
[illegible]	[illegible]	[illegible]		[illegible]							
[illegible]	[illegible]	[illegible]		[illegible]							
[illegible]	[illegible]	[illegible]	+.020	[illegible]	[illegible]	.236	.370	.030	.020	.005	.195
[illegible]	[illegible]	[illegible]	-.000	[illegible]	.030	[illegible]					
[illegible]	[illegible]	[illegible]		[illegible]							
[illegible]	[illegible]	[illegible]		[illegible]							
[illegible]	[illegible]	[illegible]		[illegible]							
[illegible]	[illegible]	[illegible]		[illegible]							
[illegible]	[illegible]	[illegible]		[illegible]							
[illegible]	[illegible]	49.750		48.910							
4900	50.000	50.000		49.160							

TITLE

GROOVE DIMENSIONS STAT FACE SEAL
INTERNAL PRESSUR.

[illegible]OJET STAND
[illegible]523

SHEET [illegible] OF [illegible]

DOCUMENT NUMBER [illegible]

ISSUED 20 DEC 64 REVISED

00 100-015 (REV 7/63

TABLE IV

STATIC FACE SEAL GROOVE DIMENSIONS
NOMINAL .378 CROSS-SECTION

DASH NO.	NOM GLAND O.D.	A DIA.	A TOL.	B DIA.	B TOL.	C	D MIN	R RAD MAX.	CONC W	FLATNESS X	GAUGE DIM.
[illegible]	[illegible]	25.000		23.770							
[illegible]	[illegible]	25.250		24.020							
[illegible]	[illegible]	25.500		[illegible]							
[illegible]	[illegible]	25.750		[illegible]							
[illegible]	26.000	26.000		[illegible]							
[illegible]	[illegible]	26.250		[illegible]							
[illegible]	[illegible]	26.500		[illegible]							
[illegible]	[illegible]	26.750		[illegible]							
[illegible]	[illegible]	27.000		[illegible]							
[illegible]	[illegible]	27.250		[illegible]							
[illegible]	[illegible]	27.500		[illegible]							
[illegible]	[illegible]	27.750		[illegible]							
[illegible]	[illegible]	28.000		[illegible]							
[illegible]	[illegible]	28.250		[illegible]							
[illegible]	[illegible]	28.500		[illegible]							
[illegible]	[illegible]	28.750		[illegible]							
[illegible]	[illegible]	29.000		[illegible]							
[illegible]	[illegible]	29.250		[illegible]							
[illegible]	[illegible]	29.500		[illegible]							
[illegible]	[illegible]	29.750		[illegible]							
[illegible]	30.000	30.000	+.020	[illegible]	+.000	.[illegible]	.585	.030	.020	.005	.310
[illegible]	[illegible]	30.250	-.000	[illegible]	-.030	.560					
[illegible]	[illegible]	30.500		29.270							
[illegible]	[illegible]	30.750		29.520							
[illegible]	[illegible]	31.000		29.770							
[illegible]	[illegible]	31.250		30.020							
[illegible]	[illegible]	31.500		30.270							
[illegible]	[illegible]	31.750		30.520							
[illegible]	[illegible]	32.000		30.770							
[illegible]	[illegible]	32.250		31.020							
[illegible]	[illegible]	32.500		31.270							
[illegible]	[illegible]	32.750		31.520							
[illegible]	[illegible]	33.000		31.770							
[illegible]	[illegible]	33.250		32.020							
[illegible]	[illegible]	33.500		32.270							
[illegible]	[illegible]	33.750		32.520							
[illegible]	[illegible]	34.000		32.770							
[illegible]	[illegible]	34.250		33.020							
[illegible]	[illegible]	34.500		33.270							
[illegible]	[illegible]	34.750		33.520							
[illegible]	[illegible]	35.000		33.770							
[illegible]	[illegible]	35.250		34.020							
[illegible]	[illegible]	35.500		34.270							
[illegible]	[illegible]	35.750		34.520							
[illegible]	[illegible]	36.000		34.770							
[illegible]	[illegible]	36.250		35.020							
[illegible]	[illegible]	36.500		35.270							
[illegible]	[illegible]	36.750		35.520							
[illegible]	37.000	37.000		35.770							
[illegible]	37.250	37.250		36.020							
[illegible]	37.500	37.500		36.270							
[illegible]	37.750	37.750		36.520							

TABLE CONTINUED ON FOLLOWING SHEET

TABLE IV (CONT'D)

DASH NO.	NOM GLAND O.D.	A DIA.	A TOL.	B DIA.	B TOL.	C	D MIN	R RAD MAX.	CONC W	FLATNESS X	GAUGE DIM.
[illegible]	33.000	[illegible]		36.770							
[illegible]	33.250	[illegible]		[illegible]							
[illegible]	33.500	[illegible]		[illegible]							
[illegible]	33.750	[illegible]		37.[illegible]							
[illegible]	[illegible]	[illegible]		[illegible]							
[illegible]	[illegible]	[illegible]		[illegible]							
[illegible]	[illegible]	37.500		[illegible]							
[illegible]	[illegible]	37.750		[illegible]							
[illegible]	38.[illegible]	38.000		[illegible]							
[illegible]	38.[illegible]	38.250		[illegible]							
[illegible]	38.[illegible]	38.500		[illegible]							
[illegible]	38.[illegible]	38.750		[illegible]							
[illegible]	39.[illegible]	39.000		[illegible]							
[illegible]	39.[illegible]	39.250		[illegible]							
[illegible]	39.[illegible]	39.500		[illegible]							
[illegible]	39.[illegible]	39.750		[illegible]							
[illegible]	40.[illegible]	40.000		[illegible]							
[illegible]	40.[illegible]	40.250		[illegible]							
[illegible]	40.[illegible]	40.500		[illegible]							
[illegible]	40.[illegible]	40.750		[illegible]							
[illegible]	41.[illegible]	41.000		39.770							
[illegible]	41.250	41.250		40.020							
[illegible]	41.[illegible]00	41.500		40.270							
[illegible]	41.750	41.750		40.520							
[illegible]	[illegible]	42.000		40.770							
[illegible]	42.[illegible]	42.250		41.020							
[illegible]	42.[illegible]	42.500		41.270							
[illegible]	42.[illegible]	42.750		41.520							
[illegible]	43.[illegible]	43.000	+.020	41.770	+.000	.[illegible]	.585	.030	.020	.005	.310
[illegible]	43.[illegible]	43.250	-.000	42.020	-.030	.[illegible]					
[illegible]	43.	43.500		42.270							
[illegible]	43.[illegible]	43.750		[illegible]							
[illegible]	44.[illegible]	44.000		[illegible]							
[illegible]	44.[illegible]	44.250		43.0[illegible]							
[illegible]	44.[illegible]	44.500		[illegible]							
[illegible]	44.	44.750		[illegible]							
[illegible]	[illegible]	45.000		43.770							
[illegible]	45.[illegible]	45.250		44.020							
[illegible]	45.[illegible]	45.500		44.270							
[illegible]	45.	45.750		44.520							
[illegible]	46.[illegible]	46.000		44.770							
[illegible]	46.	46.250		45.020							
[illegible]	46.[illegible]	46.500		45.270							
[illegible]	46.[illegible]	46.750		45.520							
[illegible]	47.[illegible]	47.000		45.770							
[illegible]	47.[illegible]	47.250		46.020							
[illegible]	47.500	47.500		46.270							
[illegible]	47.750	47.750		46.520							
[illegible]	48.000	48.000		46.770							
[illegible]	48.250	48.250		47.020							
[illegible]	48.500	48.500		47.270							
[illegible]	48.750	48.750		47.520							
[illegible]	49.000	49.000		47.770							
[illegible]	49.250	49.250		48.020							
[illegible]	49.500	49.500		48.270							
[illegible]	49.750	49.750		48.520							
[illegible]	50.000	50.000		48.770							

TABLE CONTINUED ON FOLLOWING SHEET

TABLE IV (CONT'D)

DASH NO.	NOM GLAND O.D.	A DIA.	A TOL.	B DIA.	B TOL.	C	D MIN	R RAD MAX	CONC W	FLATNESS X	GAUGE DIM.
5402	50.250	50.250		49.020							
5404	50.500	50.500		49.270							
5406	50.750	50.750		49.520							
5408	51.000	51.000		49.770							
5410	51.250	51.250		50.020							
5412	51.500	51.500		50.270							
5414	51.750	51.750		50.520							
5416	52.000	52.000		50.770							
5418	52.250	52.250		51.020							
5420	52.500	52.500		51.270							
5422	52.750	52.750		51.520							
5424	53.000	53.000		51.770							
5426	53.250	53.250		52.020							
5428	53.500	53.500		52.270							
5430	53.750	53.750		52.520							
5432	54.000	54.000	+.020	52.770							
5434	54.250	54.250	-.000	53.020							
5436	54.500	54.500		53.270							
5438	54.750	54.750		53.520							
5440	55.000	55.000		53.770							
5442	55.250	55.250		54.020							
5444	55.500	55.500		54.270							
5446	55.750	55.750		54.520	+.000	.356	.585	.030	.020	.005	.310
5448	56.000	56.000		54.770	-.030	.360					
5450	56.250	56.250		55.020							
5452	56.500	56.500		55.270							
5454	56.750	56.750		55.520							
5456	57.000	57.000		55.770							
5458	57.250	57.250		56.020							
5460	57.500	57.500		56.270							
5462	57.750	57.750		56.520							
5464	58.000	58.000		56.770							
5466	58.250	58.250		57.020							
5468	58.500	58.500		57.270							
5470	58.750	58.750		57.520							
5472	59.000	59.000		57.770							
5474	59.250	59.250		58.020							
5476	59.500	59.500		58.270							
5478	59.750	59.750		58.520							
5480	60.000	60.000		58.770							
5482	60.250	60.250		59.020							
5484	60.500	60.500		59.270							
5486	60.750	60.750		59.520							
5488	61.000	61.000		59.770							
5490	61.250	61.250		60.020							
5492	61.500	61.500	+.030	60.270							
5494	61.750	61.750	-.000	60.520							
5496	62.000	62.000		60.770							
5498	62.250	62.250		61.020							
5500	62.500	62.500		61.270							
5502	62.750	62.750		61.520							
5504	63.000	63.000		61.770							

TABLE CONTINUED ON FOLLOWING SHEET

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CODE IDENT NUMBER 05824

TITLE
GROOVE DIMENSIONS-STATIC FACE SEAL, INTERNAL PRESSURE

AEROJET STANDARD
ASD5232
SHEET 11 OF 15

ISSUED 29 DEC 64 REVISED

00-100-015 (REV 7/63)

TABLE IV (CONT'D)

DASH NO.	NOM GLAND O.D.	A DIA.	A TOL.	B	B TOL.	C	D MIN	R RAD MAX	CONC W	FLATNESS X	GAUG DIM
5506	63.250	63.250		[illegible]							
[illegible]	63.500	63.500		[illegible]							
[illegible]10	[illegible]	63.750		[illegible]							
[illegible]12	[illegible]	64.000		[illegible]							
[illegible]14	[illegible]	64.250		[illegible]							
[illegible]16	[illegible]	64.500		[illegible]							
[illegible]	[illegible]	64.750		[illegible]							
[illegible]	[illegible]	65.000		[illegible]							
[illegible]	[illegible]	65.250		[illegible]							
[illegible]	[illegible]	65.500		[illegible]							
[illegible]	[illegible]	65.750		[illegible]							
[illegible]	[illegible]	66.000		[illegible]							
[illegible]	66.250	66.250		[illegible]							
[illegible]	66.500	66.500		[illegible]							
[illegible]	66.750	66.750		[illegible]							
[illegible]	67.000	67.000		[illegible]							
[illegible]	67.250	67.250		66.020							
[illegible]	67.500	67.500		66.270							
[illegible]	67.750	67.750		66.520							
[illegible]	[illegible]	68.000		66.770							
[illegible]	[illegible]	68.250		67.020							
[illegible]	[illegible]	68.500		67.270							
[illegible]	[illegible]	68.750	+.030	67.520	+.000	.356	.585	.030	.020	.005	.310
[illegible]	[illegible]	69.000	-.000	67.770	-.030	.360					
[illegible]	[illegible]	69.250		68.020							
[illegible]	[illegible]	69.500		68.270							
[illegible]	[illegible]	69.750		68.520							
[illegible]	[illegible]	70.000		68.770							
[illegible]	70.250	70.250		69.020							
[illegible]	70.500	70.500		69.270							
[illegible]	70.750	70.750		69.520							
[illegible]	71.000	71.000		69.770							
[illegible]70	71.250	71.250		70.020							
[illegible]72	71.500	71.500		70.270							
[illegible]	71.750	71.750		70.520							
[illegible]76	72.000	72.000		70.770							
[illegible]	72.250	72.250		71.020							
[illegible]	72.500	72.500		71.270							
[illegible]	72.750	72.750		71.520							
[illegible]	73.000	73.000		71.770							
[illegible]86	73.250	73.250		72.020							
[illegible]	73.500	73.500		72.270							
[illegible]90	73.750	73.750		72.520							
[illegible]92	74.000	74.000		72.770							
[illegible]	74.250	74.250		73.020							
[illegible]96	74.500	74.500		73.270							
[illegible]98	74.750	74.750		73.520							
5600	75.000	75.000		73.770							
5602	75.250	75.250		74.020							
5604	75.500	75.500		74.270							

TABLE CONTINUED ON FOLLOWING SHEET

TABLE IV (CONT'D)

DASH NO.	NOM GLAND O.D.	A DIA.	A TOL.	B DIA.	B TOL.	C	D MIN	R RAD MAX	CONC W	FLATNESS X	GAUGE DIM.
[illegible]	75.750	75.750		[illegible]							
[illegible]	76.000	76.000		[illegible]							
[illegible]	76.250	76.250		[illegible]							
[illegible]	76.500	76.500		[illegible]							
[illegible]	76.750	76.750		[illegible]							
[illegible]	77.000	77.000		[illegible]							
[illegible]	77.250	77.250		[illegible]							
[illegible]	77.500	77.500		[illegible]							
[illegible]	77.750	77.750		[illegible]							
[illegible]	78.000	78.000		[illegible]							
[illegible]	78.250	78.250		[illegible]							
[illegible]	78.500	78.500		[illegible]							
[illegible]	78.750	78.750		[illegible]							
[illegible]	79.000	79.000		[illegible]							
[illegible]	79.250	79.250		[illegible]							
[illegible]	79.500	79.500		[illegible]							
[illegible]	79.750	79.750		[illegible]							
[illegible]	80.000	80.000		[illegible]							
[illegible]	80.250	80.250		[illegible]							
[illegible]	80.500	80.500		[illegible]							
[illegible]	80.750	80.750		[illegible]							
[illegible]	81.000	81.000		[illegible]							
[illegible]	81.250	81.250		[illegible]							
[illegible]	81.500	81.500	+.030	[illegible]	+.000	.356	.585	.030	.020	.005	.310
[illegible]	81.750	81.750	-.000	[illegible]	-.030	.360					
[illegible]	82.000	82.000		[illegible]							
[illegible]	82.250	82.250		[illegible]							
[illegible]	82.500	82.500		[illegible]							
[illegible]	82.750	82.750		[illegible]							
[illegible]	83.000	83.000		[illegible]							
[illegible]	83.250	83.250		[illegible]							
[illegible]	83.500	83.500		[illegible]							
[illegible]	83.750	83.750		[illegible]							
[illegible]	84.000	84.000		[illegible]							
[illegible]	84.250	84.250		[illegible]							
[illegible]	84.500	84.500		[illegible]							
[illegible]	84.750	84.750		[illegible]							
[illegible]	85.000	85.000		[illegible]							
[illegible]	85.250	85.250		[illegible]							
5684	85.500	85.500		84.270							
5686	85.750	85.750		84.520							
5688	86.000	86.000		84.770							
5690	86.250	86.250		85.020							
5692	86.500	86.500		85.270							
5694	86.750	86.750		85.520							
5696	87.000	87.000		85.770							
5698	87.250	87.250		86.020							
5700	87.500	87.500		86.270							
5702	87.750	87.750		86.520							
5704	88.000	88.000		86.770							
5706	88.250	88.250		87.020							
5708	88.500	88.500		87.270							

TABLE CONTINUED ON FOLLOWING SHEET

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TABLE IV (CONT'D)

DASH NO.	NOM GLAND O.D.	A DIA.	A TOL.	B DIA.	B TOL.	C	D MIN	R RAD MAX	CONC W	FLATNESS X	GAUGE DIM.
5710	88.750	88.750		87.520							
5712	89.000	89.000		87.770							
5714	89.250	89.250		88.020							
5716	89.500	89.500		88.270							
5718	89.750	89.750		88.520							
5720	90.000	90.000		88.770							
5722	90.250	90.250		89.020							
5724	90.500	90.500		89.270							
5726	90.750	90.750		89.520							
5728	91.000	91.000		89.770							
5730	91.250	91.250		90.020							
5732	91.500	91.500		90.270							
5734	91.750	91.750		90.520							
5736	92.000	92.000		90.770							
5738	92.250	92.250		91.020							
5740	92.500	92.500		91.270							
5742	92.750	92.750		91.520							
5744	93.000	93.000		91.770							
5746	93.250	93.250		92.020							
	93.500	93.500	+.030	92.270	+.000	.556	.585	.030	.020	.005	.310
	93.750	93.750	-.000	92.520	-.030	.560					
	94.000	94.000		92.770							
	94.250	94.250		93.020							
	94.500	94.500		93.270							
	94.750	94.750		93.520							
	95.000	95.000		93.770							
	95.250	95.250		94.020							
	95.500	95.500		94.270							
	95.750	95.750		94.520							
	96.000	96.000		94.770							
5750	96.250	96.250		95.020							
5752	96.500	96.500		95.270							
5754	96.750	96.750		95.520							
5756	97.000	97.000		95.770							
	97.250	97.250		96.020							
	97.500	97.500		96.270							
	97.750	97.750		96.520							
	98.000	98.000		96.770							
	98.250	98.250		97.020							
	98.500	98.500		97.270							
	98.750	98.750		97.520							
	99.000	99.000		97.770							
5774	99.250	99.250		98.020							
5776	99.500	99.500		98.270							
5778	99.750	99.750		98.520							
5800	100.000	100.000		98.770							

TITLE

GROOVE DIMENSIONS-STATIC FACE SEAL, INTERNAL PRESSURE

AEROJET STANDARD

ASD5232

SHEET 14 OF 15

ISSUED 29 DEC 64 REVISED

00-100-015 (REV 7/63)

NOTES:

1. DIMENSIONS ARE IN INCHES.

2. INTERPRET DRAWING PER STANDARDS PRESCRIBED IN MIL-D-70327.

3. THE DIMENSIONS AND NOTES SHOWN FOR THE MALE FLANGE SHALL BE SPECIFIED ON ENGINEERING DRAWING.

4. TO BE SPECIFIED ON ENGINEERING DRAWING. THIS DIAMETER SHALL BE EQUAL TO OR LESS THAN "B" DIAMETERS SPECIFIED IN TABLE I THRU IV.

5. REMOVE ALL BURRS AND SHARP EDGES EQUIVALENT TO .005/.015 RADIUS UNLESS OTHERWISE SPECIFIED.

6. AFTER MACHINING, THE MACHINED AREAS OF THE GROOVE MUST BE PROTECTED WITH APPROPRIATE PLASTIC COVERS OR PLUGS IN ORDER TO PREVENT DAMAGE TO THE MACHINED AREAS OF THE GROOVE. PROTECTIVE COVERS OR PLUG MUST BE INSTALLED AT ALL TIMES WHILE NOT BEING ACTUALLY WORKED ON.

7. AFTER MACHINING, THE MACHINED AREAS OF THE GROOVES MUST BE INSPECTED TO THE APPLICABLE DASH NUMBER DIMENSIONS IN ACCORDANCE WITH AGC-STD-4869.

8. THIS SURFACE SHALL BE FREE OF STEPS, CHATTER MARKS, AND FLAWS; AND NO RADIAL OR SPIRAL SCRATCHES, SCORES, DENTS, ETC., ARE PERMITTED EXCEEDING SURFACE ROUGHNESS HEIGHT RATING INDICATED.

ENGINEERING INFORMATION:

1. TYPICAL DRAWING CALLOUT:

MACHINE PER ASD 5232 A 1280

- 1280 — NOMINAL .094 CROSS-SECTION; NOMINAL 1.750 DIAMETER
- A — STYLE A
- ASD 5232 — GROOVE DIMENSIONS-STATIC FACE SEAL, INTERNAL PRESSURE

2. DO NOT USE UNASSIGNED DASH NUMBERS.

3. THIS AREA ON MALE FLANGE OPPOSITE THE GROOVE MUST BE $32\sqrt{C}$ FINISH AND MUST OVERLAP ON BOTH SIDES OF ACTUAL GROOVE WIDTH TO ALLOW FOR MAXIMUM MISMATCH OF FLANGES.

4. IT IS IMPERATIVE THAT THESE DIMENSIONS BE MET FOR PROPER SEALING OF THE JOINT. ANY DEVIATION FROM THESE DIMENSIONS AND TOLERANCES AS SPECIFIED WILL CAUSE LEAKAGE AND/OR ULTIMATE JOINT FAILURE.

5. IF ALLOWABLE LEAKAGE RATES WITH HELIUM OR HYDROGEN IS LESS THAN 1.0×10^{-2} STANDARD CC/SEC, ALUMINUM GROOVES CANNOT BE ANODIZED AND PASSIVATION OF THE VARIOUS 300 SERIES CRES GROOVES MUST BE CAREFULLY CONTROLLED TO PREVENT MINUTE SURFACE PITS AND POROSITY.

6. THIS AGC DESIGN STANDARD TO BE USED IN CONJUNCTION WITH AS4061, AS4062, AS4063, AS4064, AS4065 AND AS4066 SERIES OF SEALS.

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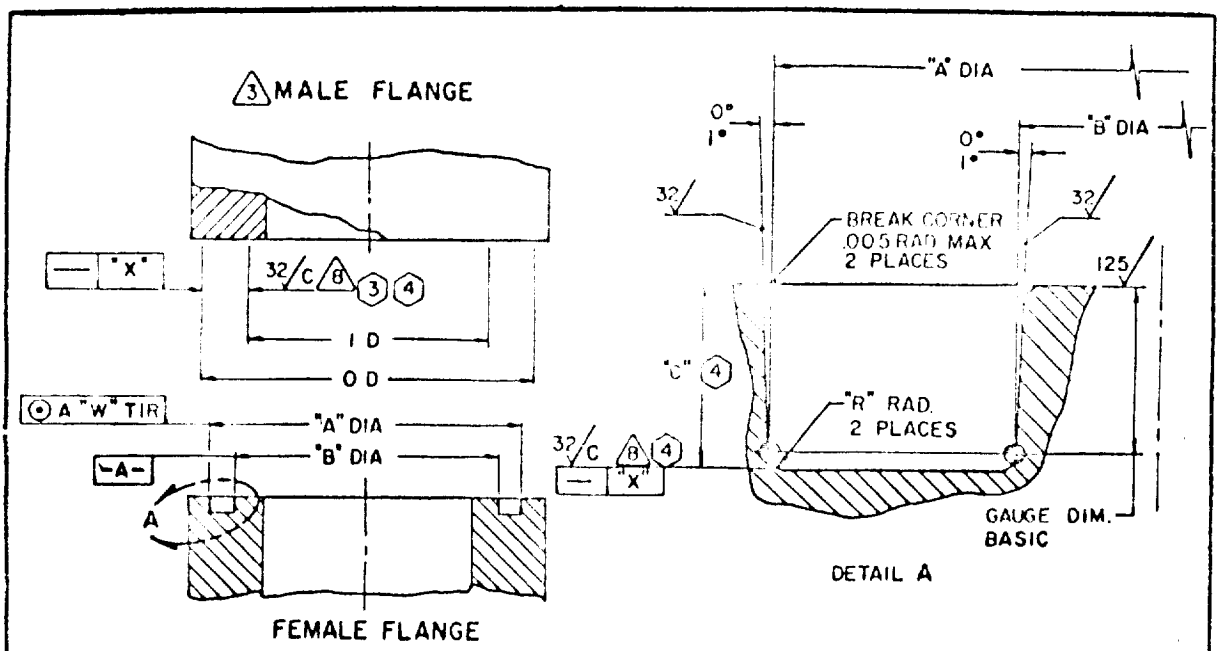
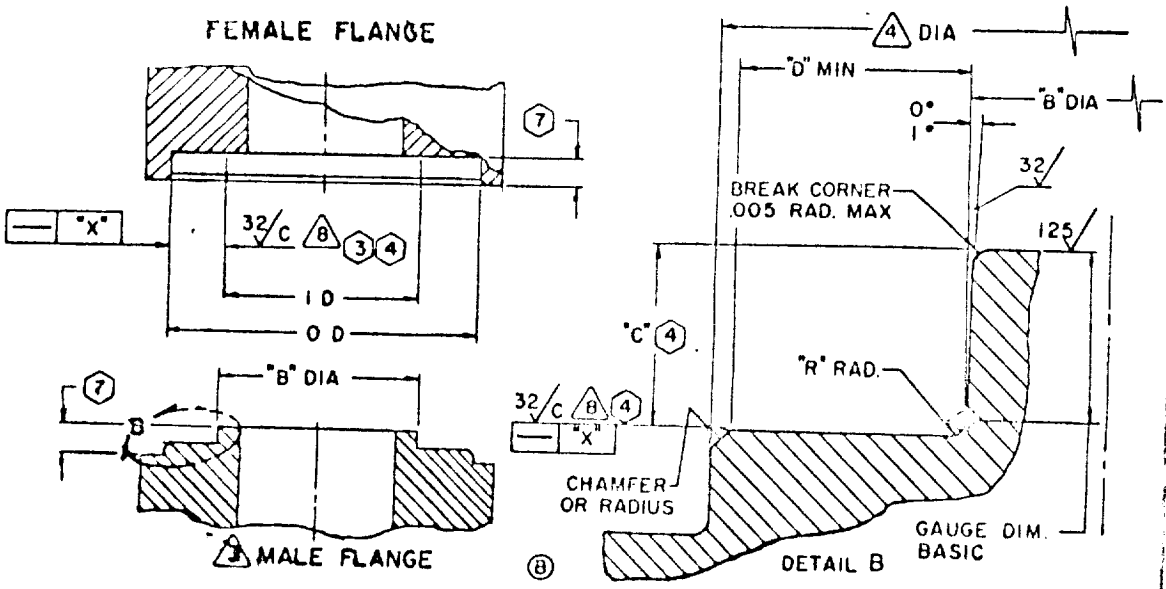
	TITLE	AEROJET STANDARD
	GROOVE DIMENSIONS-STATIC FACE SEAL, INTERNAL PRESSURE	ASD5232
		SHEET 15 OF 17

REVISED

ISSUED



STYLE A



STYLE B

05824	TITLE GROOVE DIMENSIONS - STATIC FACE SEAL EXTERNAL PRESSURE	AEROJET STANDARD ASD5233 SHEET 1 OF 15

00-100-015 (REV 7/63)

ISSUED 17 FEB 65 REVISED NOV (A) 28 JUL 65 (B) 26 JUL 66

TABLE I

STATIC FACE SEAL GROOVE DIMENSIONS
NOMINAL .094 CROSS-SECTION

DASH NO.	NOM GLAND ID	A DIA.	A TOL	B DIA.	B TOL.	C	D MIN	R RAD MAX	CONC W	FLATNESS X	GAUGE DIM
1090	.375	.695		.375							
1100	.437	.757		.437							
1110	.500	.820		.500							
1120	.562	.882		.562							
1130	.625	.945		.625							
1140	.687	1.007		.687							
1150	.750	1.070		.750							
1160	.812	1.132		.812							
1170	.875	1.195		.875							
1180	.937	1.257		.937							
1190	1.000	1.320		1.000							
[illegible]	1.062	1.382		1.062							
1210	1.125	1.445		1.125							
1220	1.187	1.507		1.187							
1230	1.250	1.570		1.250							
1240	1.312	1.632		1.312							
1250	1.375	1.695		1.375							
1260	1.437	1.757	+.005	1.437	+.000	.088	.145	.015	.002	.001	.062
1270	1.500	1.820	-.000	1.500	-.002	.090					
1280	1.562	1.882		1.562							
1290	1.625	1.945		1.625							
1300	1.687	2.007		1.687							
1310	1.750	2.070		1.750							
1320	1.812	2.132		1.812							
1330	1.875	2.195		1.875							
[illegible]	1.937	2.257		1.937							
[illegible]	2.00	2.320		2.000							
1360	2.062	2.382		2.062							
1370	2.125	2.445		2.125							
1380	2.187	2.507		2.187							
1390	2.250	2.570		2.250							
1400	2.312	2.632		2.312							
1410	2.375	2.695		2.375							
1420	2.437	2.757		2.437							
1430	2.500	2.820		2.500							
1440	2.562	2.882		2.562							
1450	2.625	2.945		2.625							
1460	2.687	3.007		2.687							
1470	2.750	3.070		2.750							
1480	2.812	3.132		2.812							

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ISSUED 17 FEB 65 REVISED N/C (A) 28 JUL 65 N/C (B) 26JUL 66

TITLE
GROOVE DIMENSIONS-STATIC FACE SEAL, EXTERNAL PRESSURE

AEROJET STANDARD
ASD5233
SHEET 2 OF 15

CODE IDENT NUMBER 05824

00-100-015 (REV 7/63)

TABLE II

STATIC FACE SEAL GROOVE DIMENSIONS
NOMINAL .125 CROSS SECTION

DASH NO.	NOM GLAND ID	A (B) DIA.	A TOL.	B DIA.	B TOL.	C	D MIN	R RAD MAX	CONC W	FLAT-NESS X	GAUGE DIM
[illegible]	[illegible]	1.180		[illegible]							
[illegible]	[illegible]	1.242		.812							
[illegible]	[illegible]	1.305		.875							
[illegible]	[illegible]	1.367		.937							
[illegible]	1.000	1.430		1.000							
2021	1.062	1.492		1.062							
2022	1.125	1.555		1.125							
2023	1.187	1.617		1.187							
2024	1.250	1.680		1.250							
[illegible]	1.312	1.742		1.312							
[illegible]	1.375	1.805	+.005	1.375							
[illegible]	[illegible]	1.867	-.000	[illegible]	+.000						
[illegible]	[illegible]	1.930		[illegible]	-.002						
[illegible]	[illegible]	2.055		1.625							
[illegible]	[illegible]	2.180		1.750							
[illegible]	[illegible]	2.305		1.875							
[illegible]	[illegible]	2.430		2.000							
[illegible]	[illegible]	2.555		2.125							
[illegible]	[illegible]	2.680		2.250							
[illegible]	[illegible]	2.805		2.375							
[illegible]	[illegible]	2.930		2.500							
[illegible]	[illegible]	3.055		2.625							
[illegible]	[illegible]	3.180		2.750							
[illegible]	[illegible]	3.305		2.875							
[illegible]	[illegible]	3.430		3.000							
[illegible]	[illegible]	3.555		3.125							
[illegible]	[illegible]	3.680		3.250		.121	.195	.020	.003	.002	.090
[illegible]	[illegible]	3.805		3.375		.123					
[illegible]	[illegible]	3.930		3.500							
[illegible]	[illegible]	4.055	+.006	3.625							
[illegible]	[illegible]	4.180	-.000	3.750	+.000						
[illegible]	3.875	4.305		3.875	-.004						
[illegible]	4.000	4.430		4.000							
2070	4.125	4.555		4.125							
2072	4.250	4.680		4.250							
2074	4.375	4.805		4.375							
2076	4.500	4.930		4.500							
2078	4.625	5.055		4.625							
2080	4.750	5.180		4.750							
2082	4.875	5.305		4.875							
2084	5.000	5.430		5.000							
2086	5.125	5.555		5.125							
2088	5.250	5.680		5.250							
2090	5.375	5.805		5.375							
2092	5.500	5.930	+.008	5.500							
2094	5.625	6.055	-.000	5.625	+.000						
2096	5.750	6.180		5.750	-.005						
2098	5.875	6.305		5.875							
2100	6.000	6.430		6.000							
2102	6.125	6.555		6.125							
2104	6.250	6.680		6.250							
2106	6.375	6.805		6.375							
2108	6.500	6.930		6.500							

TABLE CONTINUED ON FOLLOWING SHEET

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AEROJET GENERAL

COST CENTER NUMBER 05824

TITLE

GROOVE DIMENSIONS-STATIC FACE SEAL, EXTERNAL PRESSURE

AEROJET STANDARD

ASD5235

SHEET 3 OF 15

ISSUED 17 FEB 65 REVISED N/C (A) 28 JUL 65 (B) 26 JUL 66

00-100-015 (REV 7/63)

Ⓑ TABLE II (Cont'd)

DASH NO.	NOM GLAND ID	A DIA.	A TOL.	B DIA.	B TOL.	C	D MIN	R RAD MAX	CONC W	FLATNESS X	GAUGE DIM
2110	6.625	7.055		6.625							
2112	6.750	7.180		6.750							
2114	6.875	7.305		6.875							
2116	7.000	7.430		7.000							
2118	7.125	7.555	+.008	7.125	+.000						
2120	7.250	7.680	-.000	7.250	-.005						
2122	7.375	7.805		7.375							
2124	7.500	7.930		7.500							
2126	7.625	8.055		7.625							
2128	7.750	8.180		7.750							
2130	7.875	8.305		7.875							
2132	8.000	8.430		8.000							
2134	8.125	8.555		8.125							
2136	8.250	8.680		8.250							
2138	8.375	8.805		8.375							
2140	8.500	8.930		8.500							
2142	8.625	9.055		8.625							
2144	8.750	9.180		8.750							
2146	8.875	9.305	+.010	8.875	+.000						
2148	9.000	9.430	-.000	9.000	-.006						
2150	9.125	9.555		9.125							
2152	9.250	9.680		9.250							
2154	9.375	9.805		9.375							
2156	9.500	9.930		9.500							
2158	9.625	10.055		9.625							
2160	9.750	10.180		9.750		.121	.195	.020	.003	.002	.090
2162	9.875	10.305		9.875		.123					
2164	10.000	10.430		10.000							
2166	10.125	10.555		10.125							
2168	10.250	10.680		10.250							
2170	10.375	10.805		10.375							
2172	10.500	10.930		10.500							
2174	10.625	11.055		10.625							
2176	10.750	11.180		10.750							
2178	10.875	11.305		10.875							
2180	11.000	11.430		11.000							
2182	11.125	11.555		11.125							
2184	11.250	11.680		11.250							
2186	11.375	11.805		11.375							
2188	11.500	11.930	+.012	11.500	+.000						
2190	11.625	12.055	-.000	11.625	-.008						
2192	11.750	12.180		11.750							
2194	11.875	12.305		11.875							
2196	12.000	12.430		12.000							
2198	12.125	12.555		12.125							
2200	12.250	12.680		12.250							
2202	12.375	12.805		12.375							
2204	12.500	12.930		12.500							
2206	12.625	13.055		12.625							
2208	12.750	13.180		12.750							
2210	12.875	13.305		12.875							
2212	13.000	13.430		13.000							

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AEROJET GENERAL

CODE IDENT NUMBER 05824

TITLE

GROOVE DIMENSIONS-STATIC FACE SEAL, EXTERNAL PRESSURE

AEROJET STANDARD

ASD5233

SHEET 4 OF 15

ISSUED 17 FEB 65 REVISED N/C Ⓐ 28 JUL 65 Ⓑ 26 JUL 66

00-100-015 (REV 7/63)

TABLE III

STATIC FACE SEAL GROOVE DIMENSIONS
NOMINAL .250 CROSS SECTION

DASH NO.	NOM GLAND ID	A (B) DIA.	A (B) TOL.	[illegible] DIA.	[illegible] TOL.	C	D MIN	R RAD MAX	CONC W	FLAT-NESS X	GAUGE DIM
4128	7.500	8.340		[illegible]							
4132	7.750	8.590		[illegible]							
4136	8.000	8.840		[illegible]							
[illegible]	8.250	9.090		8.250							
[illegible]	8.500	9.340		8.500							
4144	8.750	9.590		8.750							
4152	9.000	9.840		9.000							
4156	9.250	10.090		9.250							
4160	9.500	10.340		9.500							
4164	9.750	10.590		9.750							
[illegible]	10.000	10.840		10.000							
4172	10.250	11.090		10.250							
[illegible]	10.500	11.340		10.500							
[illegible]	10.750	11.590		10.750							
[illegible]	11.000	11.840		11.000	+.000						
[illegible]	11.250	12.090		11.250	-.010						
[illegible]	11.500	12.340		11.500							
4196	11.750	12.590		11.750							
4200	12.000	12.840		12.000							
[illegible]	12.250	13.090		12.250							
[illegible]	12.500	13.340		12.500							
[illegible]	12.750	13.590		12.750							
[illegible]	13.000	13.840		13.000							
[illegible]	13.250	14.090	+.030	13.250		.238	.390	.030	.020	.005	.195
[illegible]	[illegible]	14.340	-.000	13.500		.240					
[illegible]	[illegible]	14.590		13.750							
[illegible]	[illegible]	14.840		14.000							
[illegible]	14.250	15.090		14.250							
[illegible]	14.500	15.340		14.500							
[illegible]	14.750	15.590		14.750							
4248	15.000	15.840		15.000							
4252	15.250	16.090		15.250							
[illegible]	15.500	16.340		15.500							
[illegible]	15.750	16.590		15.750							
4264	16.000	16.840		16.000							
4268	16.250	17.090		16.250							
4272	16.500	17.340		16.500							
4276	16.750	17.590		16.750							
4280	17.000	17.840		17.000							
4284	17.250	18.090		17.250							
4288	17.500	18.340		17.500	+.000						
4292	17.750	18.590		17.750	-.015						
4296	18.000	18.840		18.000							
[illegible]	18.250	19.090		18.250							
[illegible]	18.500	19.340		18.500							
[illegible]	18.750	19.590		18.750							
[illegible]	19.000	19.840		19.000							
[illegible]	19.250	20.090		19.250							
[illegible]	19.500	20.340		19.500							
[illegible]	19.750	20.590		19.750							
[illegible]	20.000	20.840		20.000							
[illegible]	20.250	21.090		20.250							
[illegible]	20.500	21.340		20.500							

TABLE CONTINUED ON FOLLOWING SHEET

THESE SPECIFICATIONS [illegible] PROPRIETARY RIGHTS

REVISED N/C (4) 28 JUL 65 (5) 26 JUL 66

ISSUED 17 FEB 65

AEROJET	TITLE GROOVE DIMENSIONS-STATIC FACE SEAL, EXTERNAL PRESSURE	AEROJET STANDARD ASD5233
CODE IDENT NUMBER 05824		SHEET 5 OF 15

00-100-015 (REV 7/63)

TABLE III (Cont'd)

DASH NO.	NOM GLAND ID	A DIA.	A TOL.	B DIA.	B TOL.	C	D MIN	R RAD MAX	CONC W	FLAT-NESS X	GAUGE DIM
4340	20.750	21.590		20.750							
4344	21.000	21.840		21.000							
4348	21.250	22.090		21.250							
4352	21.500	22.340		21.500							
4356	21.750	22.590		21.750							
4360	22.000	22.840		22.000							
4364	22.250	23.090		22.250							
4368	22.500	23.340		22.500	+.000						
4372	22.750	23.590		22.750	-.015						
4376	23.000	23.840		23.000							
4380	23.250	24.090		23.250							
4384	23.500	24.340		23.500							
4388	23.750	24.590		23.750							
4392	24.000	24.840		24.000							
4396	24.250	25.090		24.250							
4400	24.500	25.340		24.500							
4404	24.750	25.590		[illegible]							
4408	25.000	25.840		[illegible]							
4412	25.250	26.090		25.250							
4416	25.500	26.340		25.500							
4420	25.750	26.590		25.750							
4424	26.000	26.840		26.000							
4428	26.250	27.090	+.030	26.250		.238	.390	.030	.020	.005	.195
4432	26.500	27.340	-.000	26.500		.240					
4436	26.750	27.590		26.750							
4440	27.000	27.840		27.000							
4444	27.250	28.090		27.250							
4448	27.500	28.340		27.500							
4452	27.750	28.590		27.750							
4456	28.000	28.840		28.000							
4460	28.250	29.090		28.250							
4464	28.500	29.340		28.500							
4468	28.750	29.590		28.750	+.000						
4472	29.000	29.840		29.000	-.020						
4476	29.250	30.090		29.250							
4480	29.500	30.340		29.500							
4484	29.750	30.590		29.750							
4488	30.000	30.840		30.000							
4492	30.250	31.090		30.250							
4496	30.500	31.340		30.500							
4500	30.750	31.590		30.750							
4504	31.000	31.840		31.000							
4508	31.250	32.090		31.250							
4512	31.500	32.340		31.500							
4516	31.750	32.590		31.750							
4520	32.000	32.840		32.000							
4524	32.250	33.090		32.250							
4528	32.500	33.340		32.500							
4532	32.750	33.590		32.750							

TABLE CONTINUED ON FOLLOWING SHEET

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TABLE III (Cont'd)

DASH NO	NOM GLAND ID	A DIA.	A TOL.	B DIA.	B TOL.	C	D MIN	R RAD MAX	CONC W	FLAT-NESS X	GAUGE DIM
4536	33.000	33.840		33.000							
4540	33.250	34.090		[illegible]							
4544	33.500	34.340		[illegible]							
4548	33.750	34.590		[illegible]							
4552	34.000	34.840		34.0[illegible]							
4556	34.250	35.090		34.[illegible]							
4560	34.500	35.340		34.5[illegible]							
4564	34.750	35.590		34.7[illegible]							
4568	35.000	35.840		35.000							
4572	35.250	36.090		35.250							
4576	35.500	36.340		35.500							
4580	35.750	36.590		35.750							
4584	36.000	36.840		36.000							
4588	36.250	37.090		36.250							
4592	36.500	37.340		36.500							
4596	36.750	37.590		36.750							
4600	37.000	37.840		37.000							
4604	37.250	38.090		37.[illegible]							
[illegible]	37.500	38.340		[illegible]							
[illegible]	37.750	38.590		37.[illegible]							
[illegible]	[illegible].000	38.840		38.000							
[illegible]	[illegible].250	39.090		38.250							
[illegible]	[illegible].500	39.340	+.030	38.[illegible]	+.000	.238	.390	.030	.020	.005	.195
[illegible]	38.750	39.590	-.000	38.[illegible]	-.020	.240					
4632	39.000	39.840		39.000							
4636	39.250	40.090		39.[illegible]							
4640	39.500	40.340		39.5[illegible]							
4644	39.750	40.590		39.7[illegible]							
4648	40.000	40.840		40.000							
[illegible]52	40.250	41.090		40.250							
[illegible]6	40.500	41.340		40.500							
[illegible]0	[illegible]0.750	41.590		40.7[illegible]							
[illegible]	41.000	41.840		41.00[illegible]							
[illegible]	41.250	42.090		41.250							
[illegible]72	41.500	42.340		41.500							
4676	41.750	42.590		41.750							
4680	42.000	42.840		42.000							
4684	42.250	43.090		42.250							
[illegible]	42.500	43.340		42.500							
[illegible]	42.750	43.590		42.750							
4696	43.000	43.840		43.000							
4700	43.250	44.090		43.250							
4704	43.500	44.340		43.500							
4708	43.750	44.590		43.750							
4712	44.000	44.840		44.000							
4716	44.250	45.090		44.250							
4720	44.500	45.340		44.500							
4724	44.750	45.590		44.750							
4728	45.000	45.840		45.000							
4732	45.250	46.090		45.250							
4736	45.500	46.340		45.500							
4740	45.750	46.590		45.750							
4744	46.000	46.840		46.000							

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AEROJET	TITLE GROOVE DIMENSIONS-STATIC FACE SEAL, EXTERNAL PRESSURE	AEROJET STANDARD ASD5233
COMPONENT NUMBER 65824		SHEET 7 OF 15

00-100-015 (REV 7/63)

ISSUED 17 FEB 65 REVISED N/C (A) 28JUL 65 N/C (B) 26JUL66

TABLE III(Cont'd)

DASH NO.	NOM GLAND ID	A DIA	A TOL.	B DIA.	B TOL.	C	D MIN	R RAD MAX	(B) CONC W	FLAT-NESS X	GAIGE DIM
4748	46.250	47.090		46.250							
4752	46.500	47.340		46.500							
4756	46.750	47.590		46.750							
4760	47.000	47.840		47.000							
4764	47.250	48.090		47.250							
4768	47.500	48.340		47.500							
4772	47.750	48.590	+.030	47.750	+.000	.238	.390	.030	.020	.005	.195
4776	48.000	48.840	-.000	48.000	-.020	.240					
4780	48.250	49.090		48.250							
4784	48.500	49.340		48.500							
4788	48.750	49.590		48.750							
4792	49.000	49.840		49.000							
4796	49.250	50.090		49.250							
4800	49.500	50.340		49.500							

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TITLE

GROOVE DIMENSIONS-STATIC FACE SEAL, EXTERNAL PRESSURE

AEROJET STANDARD

ASD5233

SHEET 8 OF 15

CODE IDENT NUMBER 05924

ISSUED 17 FEB 65 REVISED N/C (A) 28JUL 65 (B) 26 JUL 66

00-100-015 (REV 7/63)

TABLE IV

STATIC FACE SEAL GROOVE DIMENSION
NOMINAL .375 CROSS SECTION

DASH NO.	NOM GLAND ID	A DIA.	A TOL.	B DIA.	B TOL.	C	D MIN	R RAD MAX	CONC W	FLAT-NESS X	GAUGE DIM
5200	24.[illegible]	25.480		24.[illegible]							
5202	24.[illegible]	25.730		25.[illegible]							
5204	24.[illegible]	25.980		25.[illegible]							
5206	25.[illegible]	26.230		25.000							
5208	25.[illegible]	26.480		25.250							
5210	25.[illegible]	26.730		25.500							
5212	25.[illegible]	26.980		25.750							
5214	26.000	27.230		26.000							
5216	26.250	27.480		26.250							
5218	26.500	27.730		26.500							
5220	26.[illegible]	27.980		26.750							
5222	27.[illegible]	28.230		27.000							
5224	27.[illegible]	28.480		27.250							
5226	27.500	28.730		27.500							
[illegible]	27.[illegible]	28.980		27.750							
[illegible]	28.[illegible]	29.230		28.000							
[illegible]	28.[illegible]	29.480		28.250							
[illegible]	28.[illegible]	29.730		28.500							
[illegible]	28.[illegible]	29.980		28.750							
[illegible]	29.[illegible]	30.230		29.000							
[illegible]	29.[illegible]	30.480		29.250							
[illegible]	29.500	30.730		29.500							
[illegible]	29.[illegible]	30.980		29.750							
[illegible]	30.[illegible]	31.230		30.000							
[illegible]	30.[illegible]	31.480		30.250							
[illegible]	30.[illegible]	31.730		30.500							
[illegible]	30.[illegible]	31.980		30.750							
[illegible]	31.[illegible]	32.230		31.000							
[illegible]	31.250	32.480		31.250							
5258	31.500	32.730	+.030	31.500	+.000	.356	.585	.030	.020	.005	.310
5260	31.750	32.980	-.000	31.750	-.020	.360					
5262	32.000	33.230		32.000							
5264	32.250	33.480		32.250							
5266	32.500	33.730		32.500							
5268	32.750	33.980		32.750							
5270	33.000	34.230		33.000							
5272	33.250	34.480		33.250							
5274	33.500	34.730		33.500							
5276	33.750	34.980		33.750							
5278	34.000	35.230		34.000							
5280	34.250	35.480		34.250							
5282	34.500	35.730		34.500							
5284	34.750	35.980		34.750							
5286	35.000	36.230		35.000							
5288	35.250	36.480		35.250							
5290	35.500	36.730		35.500							
5292	35.750	36.980		35.750							
5294	36.000	37.230		36.000							
5296	36.250	37.480		36.250							
5298	36.500	37.730		36.500							
5300	36.750	37.980		36.750							
5302	37.000	38.230		37.000							
5304	37.250	38.480		37.250							

CONTINUED ON FOLLOWING SHEET

TITLE

GROOVE DIMENSIONS-STATIC FACE SEAL, EXTERNAL PRESSURE

AEROJET STANDARD

ASD5233

SHEET 9 OF 15

CODE IDENT NUMBER 05824

00-100-015 (REV 7/63)

REVISED N/C Ⓐ 28 JUL 65 N/C Ⓑ 26 JUL 66

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TABLE IV (Cont'd)

DASH NO.	NOM GLAND ID (B)	A DIA (B)	A TOL	B DIA	B TOL	C	D MIN	R RAD MAX	CONC W	FLAT-NESS X	GAUGE DIM.
5306	37.500	38.730		37.500							
5308	37.750	38.980		37.750							
5310	38.000	39.230		38.000							
5312	38.250	39.480		38.250							
5314	38.500	39.730		38.500							
5316	38.750	39.980		38.750							
5318	39.000	40.230		39.000							
5320	39.250	40.480		39.250							
5322	39.500	40.730		39.500							
5324	39.750	40.980		39.750							
5326	40.000	40.230		40.000							
5328	40.250	41.480		40.250							
5330	40.500	41.730		40.500							
5332	40.750	41.980		40.750							
5334	41.000	42.230		41.000							
5335	41.250	42.480		41.250							
5338	41.500	42.730		41.500							
5340	41.750	42.980		41.750							
5342	42.000	43.230		42.000							
5344	42.250	43.480		42.250							
5346	42.500	43.730		42.500							
5348	42.750	43.980		42.750							
5350	43.000	44.230		43.000							
5352	43.250	44.480		43.250							
5354	43.500	44.730		43.500							
5356	43.750	44.980		43.750							
5358	44.000	45.230	+.030	44.000	+.000	.356	.585	.030	.020	.005	.310
5360	44.250	45.480	-.000	44.250	-.020	.360					
5362	44.500	45.730		44.500							
5364	44.750	45.980		44.750							
5366	45.000	46.230		45.000							
5368	45.250	46.480		45.250							
5370	45.500	46.730		45.500							
5372	45.750	46.980		45.750							
5374	46.000	47.230		46.000							
5376	46.250	47.480		46.250							
5378	46.500	47.730		46.500							
5380	46.750	47.980		46.750							
5382	47.000	48.230		47.000							
5384	47.250	48.480		47.250							
5386	47.500	48.730		47.500							
5388	47.750	48.980		47.750							
5390	48.000	49.230		48.000							
5392	48.250	49.480		48.250							
5394	48.500	49.730		48.500							
5396	48.750	49.980		48.750							
5398	49.000	50.230		49.000							
5400	49.250	50.480		49.250							
5402	49.500	50.730		49.500							
5404	49.750	50.980		49.750							
5406	50.000	51.230		50.000							
5408	50.250	51.480		50.250							
5410	50.500	51.730		50.500							

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ISSUED 17 FEB 65 REVISED (A) 28 JUL 65 (B) 26 JUL 65

TITLE

GROOVE DIMENSIONS-STATIC FACE SEAL, EXTERNAL PRESSURE

AEROJET STANDARD

ASD5233

SHEET 10 OF 15

CODE IDENT NUMBER 05824

00-100-015 (REV 7/63)

TABLE IV (Cont'd)

DASH NO.	NOM GLAND ID	A DIA	A TOL	B DIA	B TOL	C	D MIN	R RAD MAX	CONC W	FLAT-NESS X	GAUGE DIM.
5412	50.750	51.980		[illegible]							
5414	51.000	52.230		[illegible]							
5416	51.250	52.480		[illegible]							
5418	51.500	52.730		[illegible]							
5420	51.750	52.980		[illegible]							
5422	52.000	53.230		[illegible]							
5424	52.250	53.480		52.250							
5426	52.500	53.730		52.500							
5428	52.750	53.980		52.750							
5430	53.000	54.230		53.000							
5432	53.250	54.480		53.250							
5434	53.500	54.730		53.500							
[illegible]	[illegible]	54.980		[illegible]							
[illegible]	54.000	55.230		[illegible]							
[illegible]	54.250	55.480		[illegible]							
[illegible]	[illegible]	55.730		[illegible]							
[illegible]	[illegible]	55.980		[illegible]							
[illegible]	[illegible]	56.230		[illegible]							
[illegible]	[illegible]	56.480		[illegible]							
[illegible]	[illegible]	56.730		[illegible]							
[illegible]	[illegible]	56.980		55.750							
[illegible]	[illegible]	57.230	+.030	56.000							
[illegible]	[illegible]	57.480	-.000	56.250	+.000	.356	.585	.030	.020	.005	.310
[illegible]	[illegible]	57.730		56.500	-.020	.360					
[illegible]	[illegible]	57.980		56.750							
[illegible]	[illegible]	58.230		57.000							
[illegible]	[illegible]	58.480		57.250							
[illegible]	[illegible]	58.730		57.500							
[illegible]	[illegible]	58.980		57.750							
[illegible]	[illegible]	59.230		58.000							
[illegible]	[illegible]	59.480		58.250							
[illegible]	[illegible]	59.730		58.500							
[illegible]	58.750	59.980		58.750							
[illegible]	59.000	60.230		59.000							
[illegible]	59.250	60.480		59.250							
[illegible]	59.500	60.730		59.500							
[illegible]	59.750	60.980		59.750							
[illegible]	60.000	61.230		60.000							
[illegible]	60.250	61.480		60.250							
5490	60.500	61.730		60.500							
5492	60.750	61.980		60.750							
5494	61.000	62.230		61.000							
5496	61.250	62.480		61.250							
5498	61.500	62.730		61.500	+.000						
5500	61.750	62.980		61.750	-.030						
5502	62.000	63.230		62.000							
5504	62.250	63.480		62.250							
5506	62.500	63.730		62.500							
5508	62.750	63.980		62.750							
5510	63.000	64.230		63.000							
5512	63.250	64.480		63.250							
5514	63.500	64.730		63.500							
5516	63.750	64.980		63.750							

TABLE CONTINUED ON FOLLOWING SHEET

SPECIFICATIONS MAY CONTAIN INFORMATION RELATING TO ... MATERIALS, OR METHODS ... ALL OF WHICH ARE PROPRIETARY ... NOT FOR USE OR DISCLOSURE ... WITHOUT PRIOR WRITTEN PERMISSION OF AEROJET GENERAL

AEROJET GENERAL

CODE IDENT NUMBER 05824

TITLE

GROOVE DIMENSIONS-STATIC FACE SEAL, EXTERNAL PRESSURE

AEROJET STANDARD

ASD5233

SHEET 11 OF 15

ISSUED 17 FEB 65 REVISED N/C (A) 28 JUL 65 (B) 26 JUL 66

DO-100-015 (REV 7/63)

TABLE IV (Cont'd)

DASH NO.	NOM GLAND ID	A DIA.	A TOL.	B DIA.	B TOL.	C	D MIN	R RAD MAX	CONC W	FLAT-NESS X	GAUGE DIM
5518	64.000	65.230		64.000							
5520	64.250	65.480		[illegible]							
5522	64.500	65.730		[illegible]							
5524	64.750	65.980		64.750							
5526	65.000	66.230		65.000							
5528	65.250	66.480		65.250							
5530	65.500	66.730		65.500							
5532	65.750	66.980		65.750							
5534	66.000	67.230		66.000							
5536	66.250	67.480		66.250							
5538	66.500	67.730		66.500							
5540	66.750	67.980		66.750							
[illegible]	67.000	68.230		67.000							
[illegible]	67.250	68.480		67.250							
[illegible]	67.500	68.730		67.500							
[illegible]	67.750	68.980		67.750							
[illegible]	68.000	69.230		68.000							
[illegible]	68.250	69.480		68.250							
[illegible]	68.500	69.730		68.500							
[illegible]	68.750	69.980		68.750							
[illegible]	69.000	70.230	+.030	69.000	+.000	.356	.585	.030	.020	.005	.310
[illegible]	69.250	70.480	-.000	69.250	-.030	.360					
[illegible]	69.500	70.730		69.500							
[illegible]	69.750	70.980		69.750							
[illegible]	70.000	71.230		70.000							
[illegible]	70.250	71.480		70.250							
[illegible]	70.500	71.730		70.500							
[illegible]	70.750	71.980		70.750							
5574	71.000	72.230		71.000							
5576	71.250	72.480		71.250							
5578	71.500	72.730		71.500							
5580	71.750	72.980		71.750							
5582	72.000	73.230		72.000							
5584	72.250	73.480		72.250							
5586	72.500	73.730		72.500							
5588	72.750	73.980		72.750							
5590	73.000	74.230		73.000							
5592	73.250	74.480		73.250							
5594	73.500	74.730		73.500							
5596	73.750	74.980		73.750							
5598	74.000	75.230		74.000							
5600	74.250	75.480		74.250							
5602	74.500	75.730		74.500							
5604	74.750	75.980		74.750							
5606	75.000	76.230		75.000							
5608	75.250	76.480		75.250							
5610	75.500	76.730		75.500							
5612	75.750	76.980		75.750							
5614	76.000	77.230		76.000							
5616	76.250	77.480		76.250							
5618	76.500	77.730		76.500							
5620	76.750	77.980		76.750							
5622	77.000	78.230		77.000							

TABLE CONTINUED ON FOLLOWING SHEET

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TABLE IV (Cont'd)

DASH NO.	NOM GLAND ID	A DIA	A TOL	B DIA	B TOL	C	D MIN	R RAD MAX	CONC W	FLAT-NESS X	GAUGE DIM
5624	77.250	78.480		77.250							
5626	77.500	78.730		77.500							
5628	77.750	78.980		77.750							
5630	78.000	79.230		78.000							
5632	78.250	79.480		78.250							
5634	78.500	79.730		78.500							
5636	78.750	79.980		78.750							
5638	79.000	80.230		79.000							
5640	79.250	80.480		79.250							
5642	79.500	80.730		79.500							
5644	79.750	80.980		79.750							
5646	80.000	81.230		80.000							
5648	80.250	81.480		80.250							
5650	80.500	81.730		80.500							
5652	80.750	81.980		80.750							
5654	81.000	82.230		81.000							
5656	81.250	82.480		81.250							
5658	81.500	82.730		81.500							
5660	81.750	82.980		81.750							
5662	82.000	83.230		82.000							
5664	82.250	83.480		82.250							
5666	82.500	83.730		82.500							
5668	82.750	83.980		82.750							
5670	83.000	84.230	+.030	83.000	+.000	.356	.585	.030	.020	.005	.310
5672	83.250	84.480	-.000	83.250	-.030	.360					
5674	83.500	84.730		83.500							
5676	83.750	84.980		83.750							
5678	84.000	85.230		84.000							
5680	84.250	85.480		84.250							
5682	84.500	85.730		84.500							
5684	84.750	85.980		84.750							
5686	85.000	86.230		85.000							
5688	85.250	86.480		85.250							
5690	85.500	86.730		85.500							
5692	85.750	86.980		85.750							
5694	86.000	87.230		86.000							
5696	86.250	87.480		86.250							
5698	86.500	87.730		86.500							
5700	86.750	87.980		86.750							
5702	87.000	88.230		87.000							
5704	87.250	88.480		87.250							
5706	87.500	88.730		87.500							
5708	87.750	88.980		87.750							
5710	88.000	89.230		88.000							
5712	88.250	89.480		88.250							
5714	88.500	89.730		88.500							
5716	88.750	89.980		88.750							
5718	89.000	90.230		89.000							
5720	89.250	90.480		89.250							
5722	89.500	90.730		89.500							
5724	89.750	90.980		89.750							
5726	90.000	91.230		90.000							
5728	90.250	91.480		90.250							

TABLE CONTINUED ON FOLLOWING SHEET

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REVISED (A) 28 JUL 65 N/C (B) 26 JUL 66
ISSUED 17 FEB 65

TITLE

GROOVE DIMENSIONS-STATIC FACE SEAL, EXTERNAL PRESSURE

AEROJET STANDARD
ASD5233
SHEET 13 OF 15

CODE IDENT NUMBER 05824

00-100-015 (REV 7/63)

TABLE IV (Cont'd)

DASH NO.	NOM GLAND ID	A DIA	A TOL	B DIA	B TOL	C	D MIN	R RAD MAX	CONC W	FLAT-NESS X	GAUGE DIM
5730	90.500	91.730		90.500							
5732	90.750	91.980		90.750							
5734	91.000	92.230		91.000							
5736	91.250	92.480		91.250							
5738	91.500	92.730		91.500							
5740	91.750	92.980		91.750							
5742	92.000	93.230		92.000							
5744	92.250	93.480		92.250							
5746	92.500	93.730		92.500							
5748	92.750	93.980		92.750							
5750	93.000	94.230		93.000							
5752	93.250	94.480		93.250							
5754	93.500	94.730		93.500							
5756	93.750	94.980		93.750							
5758	94.000	95.230		94.000							
[illegible]	94.250	95.480		94.250							
[illegible]	94.500	95.730		94.500							
[illegible]	94.7[illegible]	95.980	+.030	94.750	+.000	.356	.585	.030	.020	.005	.310
[illegible]	95.0[illegible]	96.230	-.000	95.000	-.030	.360					
[illegible]	95.250	96.480		95.250							
[illegible]	95.500	96.730		95.500							
[illegible]	95.750	96.980		95.750							
[illegible]	96.000	97.230		96.000							
5776	96.250	97.480		96.250							
5778	96.500	97.730		96.500							
5780	96.750	97.980		96.750							
5782	97.000	98.230		97.000							
5784	97.250	98.480		97.250							
5786	97.500	98.730		97.500							
5788	97.750	98.980		97.750							
5790	98.000	99.230		98.000							
5792	98.250	99.480		98.250							
5794	98.500	99.730		98.500							
5796	98.750	99.980		98.750							
5798	99.000	100.230		99.000							
5800	99.250	100.480		99.250							

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ISSUED 17 FEB 65 REVISED N/C (A) 28 JUL 65 N/C (B) 26 JUL 66

AEROJET GENERAL

TITLE

GROOVE DIMENSIONS-STATIC FACE SEAL, EXTERNAL PRESSURE

AEROJET STANDARD

ASD5233

GO 100 003 (REV 7/63)

NOTES:

1. DIMENSIONS ARE IN INCHES.

2. INTERPRET DRAWING PER STANDARDS PRESCRIBED IN MIL-D-70327.

3. THE DIMENSIONS AND NOTES SHOWN FOR THE MALE FLANGE SHALL BE SPECIFIED ON ENGINEERING DRAWING.

4. TO BE SPECIFIED ON ENGIN[illegible] DRAWING. THE DIAMETER SHALL BE EQUAL TO OR GREATER THAN "A" DIAM[illegible] SPECIFI[illegible] IN TABLE [illegible] GRU IV.

5. REMOVE ALL BURRS AND SHARP EDGES EQUIVALENT TO .005/.015 RADIUS UNLESS OTHERWISE SPECIFIED.

6. AFTER MACHINING, THE MACHINED AREAS OF THE GROOVE MUST BE PROTECTED WITH APPROPRIATE PLASTIC COVERS OR PLUGS IN ORDER TO PREVENT DAMAGE TO THE MACHINED AREAS OF THE GROOVE. PROTECTIVE COVERS OR PLUG MUST BE INSTALLED AT ALL TIMES WHILE NOT BEING PHYSICALLY WORKED ON.

7. AFTER MACHINING, THE MACHINED AREAS OF THE GROOVES MUST BE INSPECTED TO THE APPLICABLE DASH NUMBER DIMENSIONS IN ACCORDANCE WITH AGC-STD-4869.

8. THIS SURFACE SHALL BE FREE OF STEPS, CHATTER MARKS, AND FLAWS; AND NO RADIAL OR SPIRAL SCRATCHES, SCORES, DENTS, ETC., ARE PERMITTED EXCEEDING SURFACE ROUGHNESS HEIGHT RATING INDICATED.

ENGINEERING INFORMATION:

1. TYPICAL DRAWING CALLOUT:

MACHINE PER ASD 5233 A 1280

- 1280 — NOMINAL .094 CROSS-SECTION; NOMINAL 1.562 DIAMETER
- A — STYLE A
- ASD 5233 — GROOVE DIMENSIONS-STATIC FACE SEAL, EXTERNAL PRESSURE

2. DO NOT USE UNASSIGNED DASH NUMBERS.

3. THIS AREA ON MALE FLANGE OPPOSITE THE GROOVE MUST BE 32/C FINISH AND MUST OVERLAP ON BOTH SIDES OF ACTUAL GROOVE WIDTH TO ALLOW FOR MAXIMUM MISMATCH OF FLANGES.

4. IT IS IMPERATIVE THAT THESE DIMENSIONS BE MET FOR PROPER SEALING OF THE JOINT. ANY DEVIATION FROM THESE DIMENSIONS AND TOLERANCES AS SPECIFIED WILL CAUSE LEAKAGE AND/OR ULTIMATE JOINT FAILURE.

5. IF ALLOWABLE LEAKAGE RATES WITH HELIUM OR HYDROGEN IS LESS THAN 1.0×10^{-2} STANDARD CC/SEC, ALUMINUM GROOVES CANNOT BE ANODIZED AND PASSIVATION OF THE VARIOUS 300 SERIES CRES GROOVES MUST BE CAREFULLY CONTROLLED TO PREVENT MINUTE SURFACE PITS AND POROSITY.

6. THIS AGC DESIGN STANDARD TO BE USED IN CONJUNCTION WITH AS4073, AS4074, AS4075, AS4076, AS4077, AND AS4078 SERIES OF SEALS.

7. DEPTH ON THE MALE FLANGE MUST BE EQUAL, OR GREATER THAN THE DEPTH ON THE FEMALE FLANGE. (MUST "BOTTOM OUT" AT FLANGE I.D. FACES.)

AEROJET GENERAL	TITLE GROOVE DIMENSIONS-STATIC FACE SEAL, EXTERNAL PRESSURE	AEROJET STANDARD ASD5233 SHEET 15 OF 15

DOCUMENT NUMBER 05824

00-100-015 (REV 7/63)

ISSUED 17 FEB 65 REVISED N/C (A) 23 JUL 65 (B) 26 JUL [illegible]