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SPACE VEHICLE
DESIGN CRITERIA
(CHEMICAL PROPULSION)

NASA SP-8121

LIQUID ROCKET ENGINE TURBOPUMP ROTATING-SHAFT SEALS

FEBRUARY 1978

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

FOREWORD

NASA experience has indicated a need for uniform criteria for the design of space vehicles. Accordingly, criteria are being developed in the following areas of technology:

Environment
Structures
Guidance and Control
Chemical Propulsion

Individual components of this work will be issued as separate monographs as soon as they are completed. This document, part of the series on Chemical Propulsion, is one such monograph. A list of all monographs issued prior to this one can be found in the final pages of this document.

These monographs are to be regarded as guides to design and not as NASA requirements, except as may be specified in formal project specifications. It is expected, however, that these documents, revised as experience may indicate to be desirable, eventually will provide uniform design practices for NASA space vehicles.

This monograph, "Liquid Rocket Engine Turbopump Rotating-Shaft Seals," was prepared under the direction of Howard W. Douglass, Chief, Design Criteria Office, Lewis Research Center; project management was by Harold W. Schmidt and M. Murray Bailey. The monograph was written by R. E. Burcham of the Rocketdyne Division of Rockwell International Corporation and was edited by Russell B. Keller, Jr. of Lewis. To assure technical accuracy of this document, scientists and engineers throughout the technical community participated in interviews, consultations, and critical review of the text. In particular, Paul F. Brown, Pratt & Whitney Aircraft Division, United Technologies Corporation; Paul S. Buckmann, Aerojet Liquid Rocket Company; and Lawrence P. Ludwig and John Zuk, Lewis Research Center, individually and collectively reviewed the monograph in detail.

Comments concerning the technical content of these monographs will be welcomed by the National Aeronautics and Space Administration, Lewis Research Center (Design Criteria Office), Cleveland, Ohio 44135.

February 1978

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GUIDE TO THE USE OF THIS MONOGRAPH

The purpose of this monograph is to organize and present, for effective use in design, the significant experience and knowledge accumulated in development and operational programs to date. It reviews and assesses current design practices, and from them establishes firm guidance for achieving greater consistency in design, increased reliability in the end product, and greater efficiency in the design effort. The monograph is organized into two major sections that are preceded by a brief introduction and complemented by a set of references.

The State of the Art, section 2, reviews and discusses the total design problem, and identifies which design elements are involved in successful design. It describes succinctly the current technology pertaining to these elements. When detailed information is required, the best available references are cited. This section serves as a survey of the subject that provides background material and prepares a proper technological base for the *Design Criteria* and Recommended Practices.

The Design Criteria, shown in italics in section 3, state clearly and briefly what rule, guide, limitation, or standard must be imposed on each essential design element to assure successful design. The *Design Criteria* can serve effectively as a checklist of rules for the project manager to use in guiding a design or in assessing its adequacy.

The Recommended Practices, also in section 3, state how to satisfy each of the criteria. Whenever possible, the best procedure is described; when this cannot be done concisely, appropriate references are provided. The Recommended Practices, in conjunction with the *Design Criteria,* provide positive guidance to the practicing designer on how to achieve successful design.

Both sections have been organized into decimally numbered subsections so that the subjects within similarly numbered subsections correspond from section to section. The format for the Contents displays this continuity of subject in such a way that a particular aspect of design can be followed through both sections as a discrete subject.

The design criteria monograph is not intended to be a design handbook, a set of specifications, or a design manual. It is a summary and a systematic ordering of the large and loosely organized body of existing successful design techniques and practices. Its value and its merit should be judged on how effectively it makes that material available to and useful to the designer.

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LIQUID ROCKET ENGINE TURBOPUMP ROTATING-SHAFT SEALS

1. INTRODUCTION

Seals for the rotating shaft of a liquid-propellant rocket engine turbopump are devices that prevent or minimize the leakage of propellants or fluids between the rotating shaft and the stationary pump or turbine housing. The shaft seal must provide effective sealing during high-speed rotation at all extremes of operating conditions for the specified life of the engine. Because of the serious consequences of leakage of highly reactive oxidizers and fuels, the shaft seal is one of the most critical components on a rocket engine. A seal failure can result in mixing of incompatible fluids, and the ensuing reaction may result in total destruction of the engine and vehicle. Excessive propellant leakage can lower the engine efficiency and may result in depletion of the propellant supply prior to completion of the flight mission. This monograph has been prepared to delineate the techniques that lead to seal design in which the probability of seal failure is reduced to a practical minimum.

The design and application of rotating-shaft seals for turbopumps in rocket engine systems requires consideration of many factors that normally are not critical in more conventional applications. The extreme operating conditions combined with the high reliability and fail-safe requirements dictate a thorough analysis of each detail design factor with consideration of all possible failure modes and operating variations. Most of the problems with turbopump seals have been related to material compatibility with the rocket engine propellants and to operation at the extremely low temperatures (–297° to – 423°F)* of the cryogenic fluids. The low-temperature problem is compounded on some turbopump seals by the extreme temperature gradient that occurs when the cryogenic fluid must be sealed along a shaft that is adjacent to a high-temperature (1200°F) turbine. Additional thermal problems are created by the heat generated at the seal face by rubbing friction and viscous shear at the high rotational speeds (500 ft/sec). Rubbing-surface temperatures in excess of 1000°F have been measured on liquid-oxygen seals when the fluid environment was at –297°F.

The shaft seals currently in use on rocket engine turbopumps are primarily the face-contact and circumferential** types. In general, the face-contact seals (face seals in rubbing contact with the mating surface) are used for sealing liquid propellant, and the circumferential seals

*Factors for converting U.S. customary units to the International System of Units (SI units) are given in Appendix A.

**Seal terminology used in the text basically is that presented in ASLE SP-1 (ref. 1). Symbols, materials, and abbreviations are defined or identified in Appendix B.

are used for sealing turbine hot gases. Circumferential seals may employ either rubbing-contact seal elements or clearance elements. Circumferential clearance seals, particularly labyrinth types, are used extensively where reliability is the primary consideration and the increased leakage over that of face and circumferential clearance seals is acceptable.

The monograph deals primarily with the experience and knowledge accumulated in development and operational programs to date; therefore, most of the discussion is concerned with the face-contact, circumferential, and labyrinth seals. Because future requirements indicate the need for more advanced configurations such as the hydrostatic and hydrodynamic seals, these designs are also treated.

The monograph is organized around the sequence of tasks normally involved in seal design:

(1) Seal System. – The arrangement of seal assemblies, drains, and purges is selected to minimize the severity of seal operating conditions, provide allowances for all extremes of operation, and allow for a single seal failure without destructive failure of the turbopumps. The seal designer must be involved at the preliminary turbopump design layout to ensure that the required operating conditions are consistent with reliable seal performance.

(2) Seal Assembly. – The basic type of seal assembly is selected to satisfy the operating and performance requirements. The design problems and performance limitations that are related to the seal assembly and turbopump application are considered. The seal assembly and seal system designs are iterated until an acceptable compromise between operating conditions and performance is established.

(3) Seal Components. – The detail design analysis necessary to establish the requirements and configuration of the detail components of the seal assembly is performed and integrated with the seal system and seal assembly designs. The seal component design analysis treats the following design factors:

- Leakage
- Seal materials
- Seal interface configuration
- Secondary element
- Pressure balance
- Spring load
- Secondary friction
- Resonant frequencies
- Dynamic response
- Power loss
- Heat transfer and thermal analysis
- Stress and deflection analysis
- Dimensional tolerances

These tasks are considered in the monograph in the order and manner in which the designer must handle them. Within the task areas, the critical aspects of the requirements that the seal design must satisfy are presented.

2. STATE OF THE ART

The technology for turbopump rotating-shaft seals has evolved through approximately 20 years of rocket engine development. Because of the severe operating conditions for turbopump seals, most turbopump seal designs have required considerable development and have extended the state of the art, particularly in the areas of material compatibility, low-temperature capability, and high-speed service.

Compatibility of seal materials with rocket engine propellants has been a major problem. The highly reactive propellants are not compatible with many of the conventional seal materials. The strong oxidizers such as liquid oxygen (LO_2) or liquid fluorine (LF_2) are capable of combustible reactions that can result in total destruction of the engine and vehicle. Many of the early seal experiments resulted in explosive reactions. Much effort was expended over the years to compile data on material compatibility and to make effective use of the information in successful seal design.

The extremely low temperatures (−297°F to −423°F) of cryogenic propellants precluded the use of conventional elastomer seals and resulted in the development of the plastic-lip and metal-bellows designs. Welded metal-bellows have been extensively developed during the past 10 years and are currently the most common seal configuration in use with cryogenic fluids. Because cryogenic fluids do not provide significant lubrication, material combinations that are self-lubricating have been developed.

The low temperature of the cryogenic propellants in combination with the high temperature of the turbine hot gases results in severe temperature gradients through the seal system and across the seal assembly. The hardware is chilled to cryogenic temperature before start and must withstand the thermal stresses caused by heating from the hot gas during operation. Additional thermal gradients are caused by the heat generation at the seal interface by rubbing friction and viscous shear. Therefore, considerable development effort has been directed at providing for transient and operational thermal contraction/expansion, thermal stresses, and thermal distortion.

A summary of the chief design features of rotating-shaft seals for representative rocket engine turbopumps is presented in table I. The seal types are grouped according to the fluid being sealed. The materials for the seal nosepiece, mating ring, secondary seal or bellows element, and housing are listed for each seal. Seal diameter, pressure, and speed indicate the operating conditions. The seal face load and pressure-balance ratio show the design parameters. Face load is calculated from the sum of the spring load and the hydraulic load caused by fluid pressure differential. Hydraulic load is based on the pressure-induced closing force minus an assumed interface-pressure-induced separating force equivalent to one-half the pressure differential (linear profile). The integration of these data with the successful test experience forms the basis of the discussion under the separate design subjects.

Table I. – Summary of Chief Design Features of Representative Turbopump Rotating-Shaft Seals

Sealed *fluid*	Seal type	Materials				Seal diam, in.	Fluid pressure, psig	Shaft *speed*, rpm	Rubbing *speed*, fps	Spring *load*, lbf	Total load, lbm ①	Unit load, psi ②	PV factor, *psi x fps* ($x10^3$) ③	Balance ratio ④	Successful test experience				Engine or program
		Nose-piece	Mating material	Secondary or bellows	Housing										Total number tested	Total test, hr	Wear *life*, hr ⑤	Dynamic leakage ⑥	
Liquid Hydrogen	Shaft Riding Segmented Carbon	Carbon CDJ83	No plating A286	**	304	3.5	600	22000	367	NA	NA	NA	NA	NA	NA	8	1	NA	NERVA
Liquid Hydrogen	Face Contact Metal Piston Ring	Carbon P5AG	Chrome on Inconel 5667	AMS 4530 or 4650	AMS 5735	2.09	85	30800	280	16	31	50	14	.75	NA	NA	NA	NA	RL-10
Liquid Hydrogen	Face Contact Metal Piston Ring	Carbon P5AG	Chrome on Inconel 5667	AMS 4530 or 4650	AMS 5735	1.70	400	30800	214	8	38	76	16	.55	NA	NA	NA	NA	RL-10
Liquid Hydrogen	Face Contact Metal Piston Ring	Carbon P5AG	Chrome on Inconel 5665	AMS 4530 or 4650	Inconel 5665	1.59	500	30800	235	4	17	52	12	.6	NA	NA	NA	NA	RL-10
Gaseous Hydrogen -400°F	Face Contact Welded Bellows	Carbon P5N	Chrome on 310	347	Invar 36	2.531	20	34000	375	7	9	11	4.2	.7	17	48	4	1 SCFM	Phoebus
Gaseous Hydrogen -360°F	Face Contact Welded Bellows	Carbon P5AG	Chrome on AMS 5735	347	AMS 5646 or 5512	1.23	20	12300	66	8	NA	17	1.1	.6	NA	NA	NA	NA	RL-10
Gaseous Hydrogen -110°F	Face Contact Metal Piston Ring	Carbon P5AG	Chrome on AMS 5665	AMS 4530 or 4650	Inconel 5665	2.02	450	30800	269	12	55	82	22	.6	NA	NA	NA	NA	RL-10
Gaseous Hydrogen	Shaft Riding Segmented Carbon	Carbon CDJ83	Tungsten carbide on A286	**	NA	3.0	700	24000	312	NA	NA	NA	NA	NA	NA	150+	4+	NA	NERVA
Gaseous Hydrogen -400°F	Face Contact Welded Bellows	Carbon P5N	Chrome on Inconel X	Inconel 750	321	2.950	50	28400	434	9	25	29	12.6	.7	100+	100+	2	3 SCFM	J-2
Hot Gas $H_2 + H_2O$ 1000°F	Shaft Riding Segmented Carbon	Carbon G84SC	Chrome on Inconel X	**	Inconel 600	3.768	100	28400	460	2	15	62	28.6	1.5	100+	100+	2	20 SCFM	J-2
Hot Gas $H_2 + H_2O$ 1000°F	Face Contact Welded Bellows	Carbon P5N	Chrome on Inconel X	Inconel 750	321	3.339	75	9000	130	19	41	40	5.2	.8	100+	100+	3	3 SCFM	J-2
Hot Gas 500°F GN_2	Face Contact Welded Bellows	Carbon EY105	LW-5 on Inconel X	Inconel 718	321	4.53	130	6000	120	11	51	38.6	4.6	.95	6	.38	NA	NA	M-1
Hot Gas A-50 + N_2O_4	Face Contact Welded Bellows	Carbon CDJ83	No plating 440C	AM-350	347	2.668	95	25000	277	NA	NA	40	11	.65	100+	10+	.9	NA	Titan III
Liquid Fluorine	Face Contact Welded Bellows	Al_2O_3 on Inconel 600	Al_2O_3 on Inconel 718	Inconel 718	Inconel 718	.620	50	75000	205	5	6	80	16.4	.6	2	.03	NA	1 SCFM	R&D
Liquid Fluorine	Face Contact Welded Bellows	K162B	No plating K162B	Inconel 750	Inconel 750	1.500	50	28000	170	15	25	60	10	.6	1	.03	NA	2 SCFM	R&D
Liquid Fluorine	Face Contact Machined Bellows	Al_2O_3	No plating K162B	Inconel 718	Inconel 718	1.82	600	11000	88	14	26	44	3.9	.6	21	6.3	.2	NA	Modified RL-10

(continued)

Table I. – Summary of Chief Design Features of Representative Turbopump Rotating-Shaft Seals (continued)

Sealed fluid	Seal type	Materials				Seal diam, in.	Fluid pressure, psig	Shaft speed, rpm	Rubbing speed, fps	Spring load, lbf	Total load, lbm (1)	Unit load, psi (2)	PV factor, psi x fps $(x10^3)$ (3)	Balance ratio (4)	Successful test experience				Engine or program
		Nose-piece	Mating material	Secondary or bellows	Housing										Total number tested	Total test, hr	Wear life, hr (5)	Dynamic leakage (6)	
Liquid Oxygen	Face Contact Welded Bellows	Carbon P692	Chrome on 4130	347	304	2.630	225	6750	77	31	95	158	12.2	0.97	30	35.8	3	10 SCFM	Thor
Liquid Oxygen	Face Contact Welded Bellows	Carbon P692	Chrome on 4130	Inconel 718	304	2.630	200	6800	78	82	106	176	13.7	.7	85	29	3	10 SCFM	H-1
Liquid Oxygen	Face Contact Welded Bellows	Carbon P5N	Chrome on Inconel X	Inconel 750	321	2.974	200	8650	112	41	150	210	23.5	.85	100+	100+	2	15 SCFM	J-2
Liquid Oxygen	Face Contact Welded Bellows	Carbon P692	Chrome on Inconel X	Inconel 718	321	2.935	200	9000	115	41	150	210	24	.85	25	15	2	15 SCFM	J-2S
Liquid Oxygen	Face Contact Welded Bellows	Carbon P5N	LW-5 on Inconel X	Inconel 718	321	6.902	450	4000	120	20	471	196	23.5	.92	21	2.4	1.7	2.3 SCFM	M-1
Liquid Oxygen	Face Contact Welded Bellows	Carbon P5AG	Chrome on AMS 5646	Inconel 750	AMS 5646	1.70	400	12300	92	17	67	41	3.8	.65	NA	NA	NA	NA	RL-10
Liquid Oxygen	Face Contact Lip Secondary	Carbon P692	Chrome on 440C	Mylar	303	6.463	140	6000	170	58	127	51	8.7	.7	100+	100+	3	25 SCFM	F-1
Liquid Oxygen	Face Contact Lip Secondary	Carbon P692	Chrome on 4130	Kel-F	302	2.650	225	6800	78	32	82	130	10	.85	100+	100+	3	10 SCFM	Thor
Liquid Oxygen	Face Contact Lip Secondary	Carbon P692	Chrome on 4130	Kel-F	302	2.630	250	10000	115	32	86	137	15.8	.85	100+	100+	3	10 SCFM	Atlas
Liquid Oxygen	Face Hybrid Welded Bellows (7)	Carbon P692	Chrome on Inconel X	Inconel 718	Inconel 718	2.65	50	25000	290	34	NA	NA	NA	.6	1	14	10	15 SCFM	ADP
Liquid Oxygen	Shaft Riding Segmented Carbon	Carbon P5N	LW-5 on Inconel X	**	321	4.33	385	6000	113	NA	NA	NA	NA	NA	2	.66	NA	3.5 GPM	M-1
Liquid Hydrogen	Face Contact Welded Bellows	Carbon P5N	Chrome on Inconel X	Inconel 750	321	2.950	200	28000	360	15	64	89	32	.7	100+	100+	2	.01 lbm/sec	J-2
Liquid Hydrogen	Face Contact Welded Bellows	Carbon P5N	Chrome on Inconel X	Inconel 718	Inconel 600	3.515	350	28000	430	20	95	101	43.5	.7	25	15	2	.02 lbm/sec	J-2S
Liquid Hydrogen	Face Contact Welded Bellows	Carbon P5N	Chrome on - -	347	Invar 36	2.531	150	34000	375	7	25	31	11.6	.7	17	48	4	.006 lbm/sec	Phoebus
Liquid Hydrogen	Face Contact Welded Bellows	Carbon PO3N	LW-5 on Inconel X	Inconel 750	Inconel	4.53	300	18000	356	11	133	100	35.6	.8	1	.13	.15	NA	M-1
Liquid Hydrogen	Shaft Clearance Arch Bound Segmented Carbon	Carbon P5N	LW-5 on Inconel X	**	321	5.24	210	15500	385	NA	NA	NA	NA	NA	NA	.56	NA	50 GPM	M-1
Liquid Hydrogen	Shaft Riding Segmented Carbon	Carbon CDJ83	Tungsten carbide on A286	**	304	3.0	150	24000	312	NA	NA	NA	NA	NA	NA	200+	10+	NA	NERVA

(continued)

Table I. – Summary of Chief Design Features of Representative Turbopump Rotating-Shaft Seals (concluded)

Sealed fluid	Seal type	Materials				Seal diam, in.	Fluid pressure, psig	Shaft speed, rpm	Rubbing speed, fps	Spring load, lbf	Total load, lbm (1)	Unit load, psi (2)	PV factor, psi x fps ($x10^3$) (3)	Balance ratio (4)	Successful test experience				Engine or program
		Nose-piece	Mating material	Secondary or bellows	Housing										Total number tested	Total test, hr	Wear life, hr (5)	Dynamic leakage (6)	
FLOX (8)	Face Contact Lip Secondary	K162B	No plating K162B	Kel-F	302	2.626	225	10000	114	30	61	97	11	.7	4	3.1	3	10 SCFM	R&D
N_2O_4	Face Contact Welded Bellows	Carbon EY-105	LW-5 on 321	AM-350	300	2.5	50	8360	91	NA	NA	31	2.8	.7	40+	3+	1+	NA	Titan III
A-50 (9)	Face Contact Welded Bellows	Carbon EY-105	LW-5 on 321	AM-350	347	1.887	50	23500	195	10	NA	33	6.4	.7	40+	3+	1+	NA	Titan III
RP-1	Face Contact V-Packing Secondary	Carbon G39	Chrome on 4130	Buna-N	300	2.364	250	10000	107	10	64	103	10.6	.85	100+	100+	4+	5 CC/HR	Atlas
RP-1	Face Contact V-Packing Secondary	Carbon G39	Chrome on 4130	Buna-N	300	3.177	200	6500	90	22	75	99	8.9	.85	100+	100+	4+	5 CC/HR	Thor
RP-1	Face Contact Welded Bellows	Carbon CCA-72	Chrome on 4130	AM-350	302	3.174	200	7000	97	18	91	117	11.4	.97	36	46	4+	2 CC/HR	H-1
RP-1	Face Contact O-Ring Secondary	Carbon G39	Chrome on 440C	Viton A	Al 2024	6.497	300	6500	181	47	304	126	22.9	.8	100+	100+	3	100 CC/HR	F-1
RP-1/ Hot Gas 800^oF	Face Contact Welded Bellows	Carbon P2003	Chrome on Hastelloy	AM-350	Hastelloy	10.142	120	6500	280	300	414	81	22.8	.65	12	4.8	NA	NA	F-1
Hot Gas LOX + RP-1 1000^oF	Shaft Riding Segmented Carbon	Carbon G84	Chrome on 4340	**	416	9.500	50	6500	270	NA	NA	NA	NA	NA	100+	100+	4+	NA	F-1
Hot Gas LOX + RP-1	Shaft Riding Segmented Carbon	Carbon G84	Tungsten carbide on 4340	**	416	2.000	125	40000	350	NA	NA	NA	NA	NA	100+	100+	4+	NA	Atlas
Hot Gas LOX + RP-1	Shaft Riding Floating Ring	Carbon P33	Tungsten carbide on 4340	**	416	3.000	160	29000	380	**	**	**	**	**	100+	100+	4+	NA	H-1

(1) Spring Load + Hydraulic Load (Based on Linear Profile)

(2) Total Nose Load ÷ Nose Contact Area

(3) Nose Unit Load x Rubbing Speed, psi x fps

(4) Effective Closing Area ÷ Sealing Dam Area

(5) Estimated Minimum Effective Sealing Life

(6) Measured or Estimated Maximum

(7) Annular Grooved Face

(8) Mixture 30% Liquid Fluorine + 70% Liquid Oxygen

(9) Aerozine 50 (50% hydrazine + 50% UDMH)

NA - data not available

** - does not apply

Anticipated operating conditions for turbopump seals in the future will require the current state of the art to be extended in the areas of long life, high speed, high pressure, and low leakage. For example, the operating conditions for the seals in the turbopumps on the SSME* include liquid-oxygen pressures up to 650 psia, hot (900°F) gas pressures up to 4000 psia, and rotational speeds up to 400 ft/sec, all combined with a wear-life requirement of 10 hr. Current designs generally are capable of satisfactory operation at one of these conditions; however, new concepts will be required to satisfy all the conditions concurrently. The current designs for rubbing-contact seals provide low leakage at pressures up to approximately 500 psi or speeds up to approximately 500 ft/sec; however, the wear life is limited to approximately 4 hours. The current designs for circumferential floating-ring controlled-gap or labyrinth seals are capable of meeting the higher pressure-speed-life requirements, but the leakage rate is excessive for many applications. The seal concepts that appear to have the best potential for satisfying all of the future requirements are the hydrostatic or hydrodynamic designs. These seals maintain a small ($\approx$ 0.0001 to 0.0004 in.) clearance gap that essentially eliminates rubbing contact and minimizes leakage. The test experience in cryogenic fluids is limited; however, the feasibility of the design concepts has been demonstrated in several programs.

2.1 SEAL SYSTEM

The seal system consists of the arrangement of seal assemblies, drains, and purges required to satisfy the overall design objectives. The first and one of the most important steps in the design of turbopump seals is the tradeoff analysis for selection of the total seal system. Reliable operation of the turbopump requires a seal system designed to minimize the severity of seal operating conditions, provide allowances for all possible extremes of operation, and allow for seal failure without destructive failure of the turbopump. Many methods to accomplish these goals exist; however, it is usually necessary that the design features be incorporated as part of the preliminary turbopump layout, since many of the seal system design requirements are interrelated with other turbopump design features and may be difficult to incorporate later.

The seal system design generally is iterated with consideration of the individual seal assembly capabilities until an acceptable compromise between operating conditions and performance is reached. The relative severity of the operating conditions is evaluated by comparing the fluid-pressure/rubbing-speed relation for specific types of seals and fluids with the current practices and recommended limits. This relation provides a guide to the feasibility of alternate configurations for the system design before the detail seal-assembly designs are established.

The types of turbopump seal systems currently used to contain and separate rocket engine propellants are illustrated and described in figures 1 through 10.

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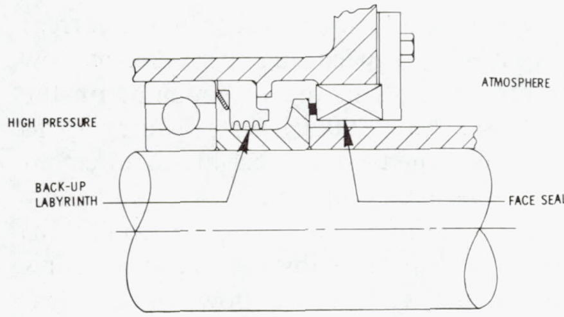
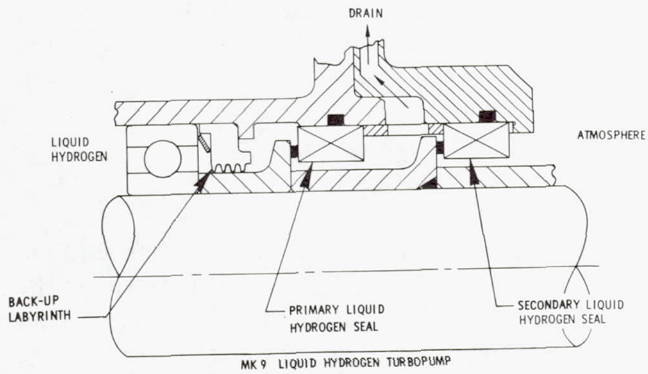
*Space Shuttle Main Engine.



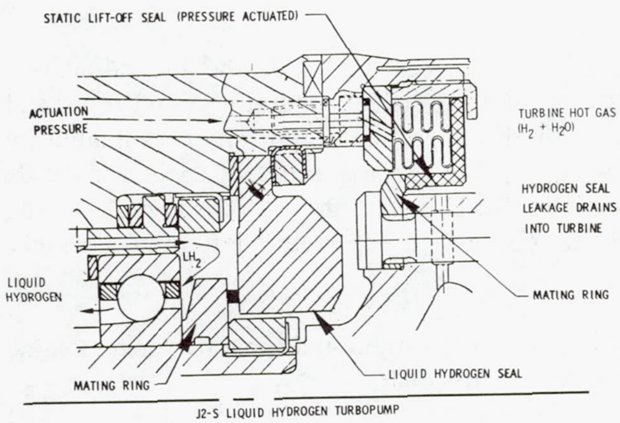
One-seal system; used when propellants can be safely drained to atmosphere or when external leakage is not harmful. Labyrinth seal provides backup in event of face seal failure.

Figure 1. — One-seal system with back-up labyrinth for containing liquid propellants.



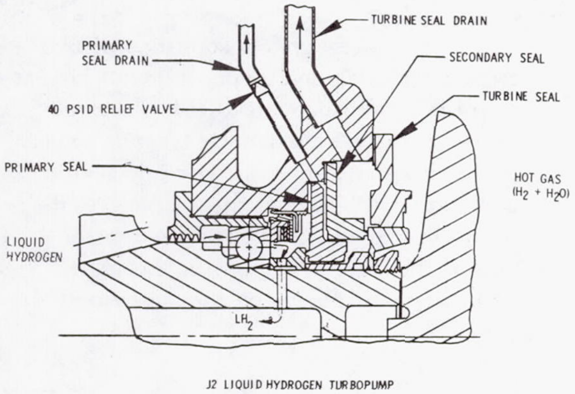
Two-seal system with a drain; used when propellants (e.g., liquid hydrogen) may be hazardous when mixed with atmospheric air or when external leakage would be harmful. Labyrinth seal provides backup in event of face-seal failure.

Figure 2. — Two-seal system with back-up labyrinth for containing liquid hydrogen.



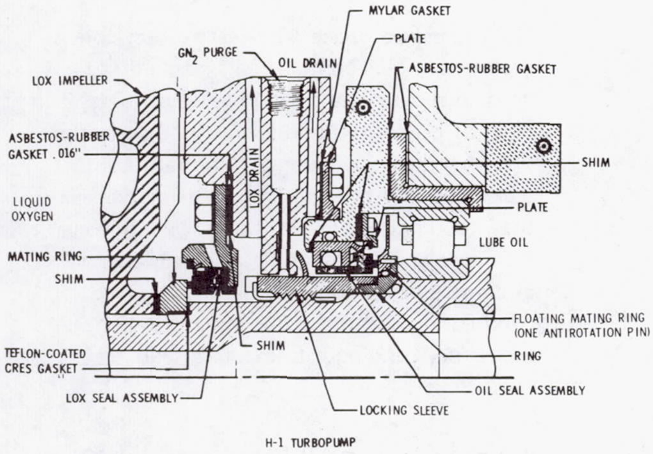
Two-seal system consisting of one face-contact seal and one static liftoff seal; used when propellants can be safely drained into the turbine hot-gas area and when minimum static leakage is desirable. The static liftoff seal provides static sealing during nonrotation periods and is pressure actuated to lift off the mating ring during rotation.

Figure 3. — Two-seal system for containing liquid hydrogen.



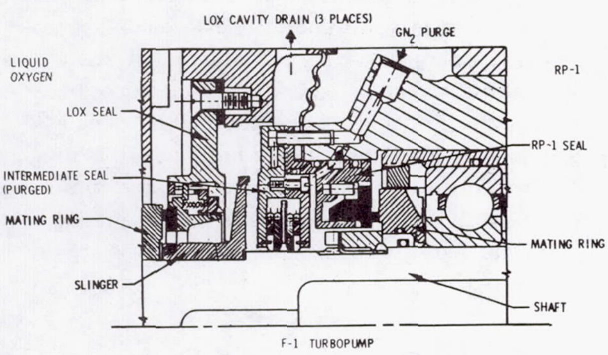
Compact three-seal system consisting of two face-contact seals and one circumferential seal with a drain; used when propellants and turbine hot gases can be safely mixed in a common drain cavity. The primary seal drain utilizes a relief valve to maintain a desired pressure level to reduce the pressure differential across the primary seal and ensure cooling of the secondary seal. The secondary seal provides additional sealing capacity in the event of a primary seal failure and prevents moisture from freezing on the primary seal.

Figure 4. — Three-seal system for containing liquid hydrogen and (H_2 + H_2O) hot gas.



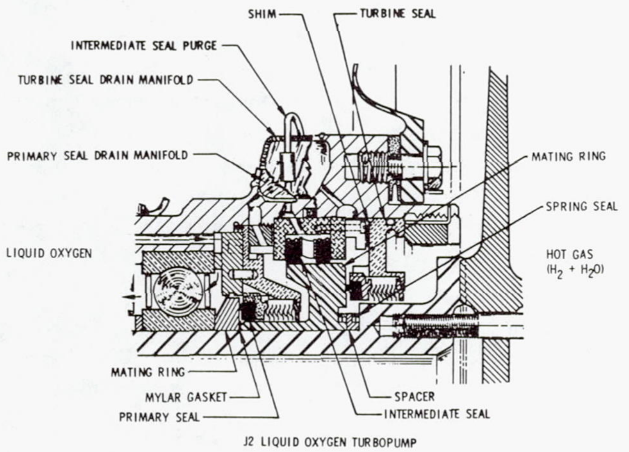
Three-seal system consisting of two face-contact seals and one purged labyrinth intermediate seal; used to separate two incompatible fluids such as liquid oxygen and lubricating oil. The leakage from the face-contact seals is drained out through separate drains. The labyrinth intermediate seal, which is gas purged, provides separation of the drain cavities during normal operation, but may not be effective in the event of a seal failure.

Figure 5. — Three-seal system for separating liquid oxygen and oil.



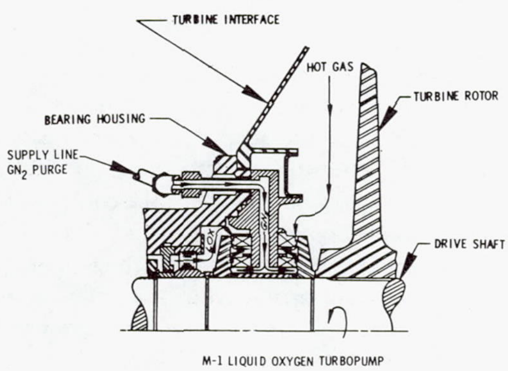
Four-seal system consisting of two face-contact seals and a purged, double circumferential intermediate seal; used to separate two incompatible fluids such as liquid oxygen and RP-1. The leakage from the face-contact seals is drained through separate drains. The double circumferential seal, which is inert-gas purged, provides effective separation of the drain cavities in the event of a seal failure. The slinger prevents direct impingement of high-velocity fluid and assists in routing the leakage through the drain.

Figure 6. — Four-seal system for separating liquid oxygen and RP-1.



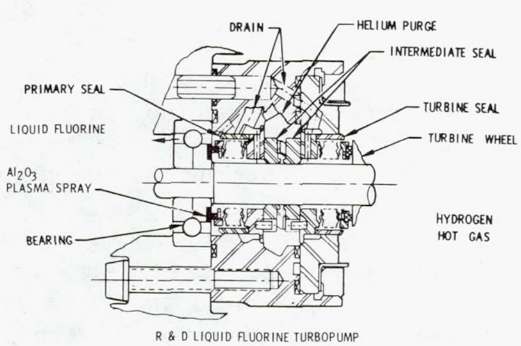
Compact four-seal system consisting of two face-contact seals and a purged double circumferential intermediate seal; used to separate two incompatible fluids such as liquid oxygen and hydrogen-rich hot gas. The principle of operation is the same as that in the system shown in figure 6. Overheating and distortion of the mating ring may be a problem with two seals rubbing on the same ring.

Figure 7. – Four-seal system for separating liquid oxygen and (H_2 + H_2O) hot gas.



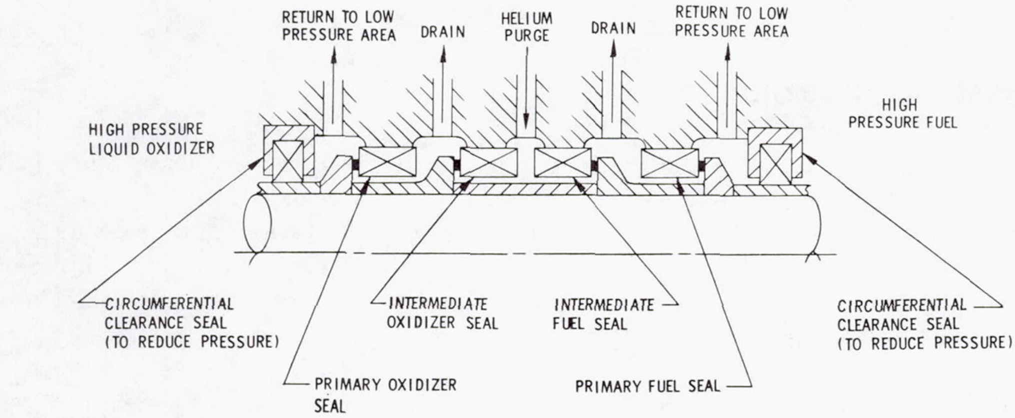
Minimum-axial-space four-seal system consisting of four face-contact seals stacked radially; used to separate two incompatible fluids. Principle of operation is the same as that in the system shown in figure 6. The face-contact seals generally provide a more effective seal than the circumferential shaft seals; however, the increased radial space required, additional complexity, and potential mating ring overheating and distortion may prevent their use as dry-running intermediate seals (ref. 2).

Figure 8. – Four-seal system for separating liquid oxygen and (H_2 + H_2O) hot gas (radial stack).



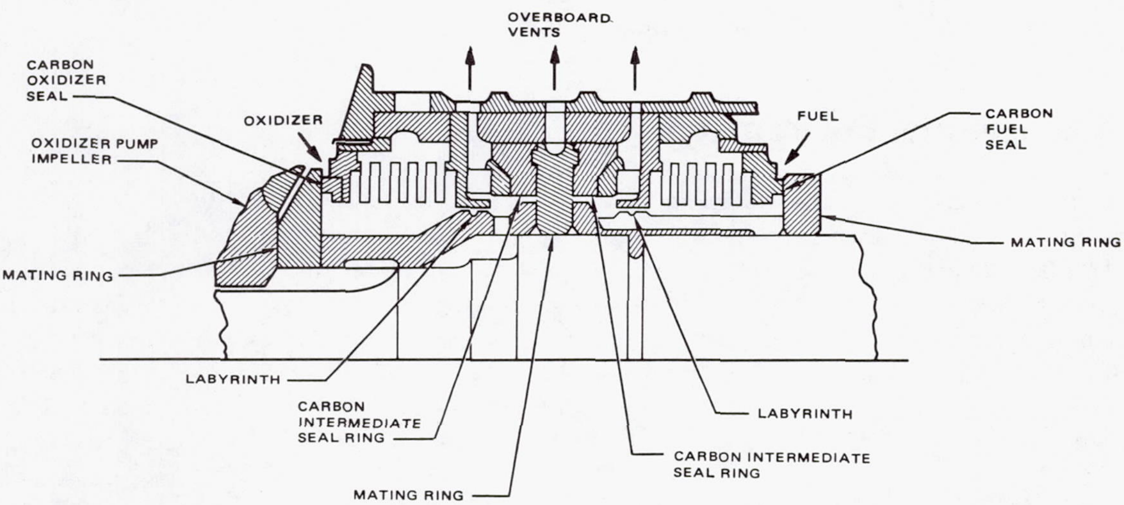
Compact minimum-radial- and axial-space four-seal system consisting of two face-contact seals and a purged, double circumferential intermediate seal; used to separate two incompatible fluids. Principle of operation is the same as that in the system shown in figure 6. Mating-ring distortion may be a problem due to use of structurally stressed parts for the mating ring. Separate, nonloaded mating rings are preferred.

Figure 9. – Four-seal system for separating liquid fluorine and hydrogen hot gas (minimum space).



Six-seal system consisting of four face-contact seals and two circumferential clearance seals; used to separate two high-pressure incompatible fluids when axial space is not critical. The circumferential clearance seals (e.g., floating ring and labyrinth) decrease the high pressure to a value acceptable for face-contact seals by allowing recirculation back to a low-pressure area such as the pump inlet. The leakage from the primary seals is drained through separate drains. The inert-gas purge between the two face-contact intermediate seals provides an effective separation of the drain cavities in the event of seal failure.

(a) Six-seal system using intermediate face seals



Six-seal system consisting of two face seals, two labyrinth seals, and two ring seals with both radial and circumferential contact faces; used to separate two high-pressure incompatible fluids when axial space is not critical. The face seals decrease the high pressures to a modest value such that the ring seal can ensure low overboard leakage. Should a face seal fail, a labyrinth seal limits the overboard leakage to an acceptable level. For ground test operations, where back pressure is high, the ring seals can be center pressurized with helium; this ensures safe separation of the two propellants by providing a helium dam between both vent cavities (refs. 3, 4, and 5).

(b) Six-seal system using intermediate ring seals

Figure 10. – Two six-seal systems for separating high-pressure liquid oxidizer and fuel.

2.1.1 Pressure Environment

High-pressure environments for seals generally result in lower turbopump reliability and larger propellant leakage. High pressures also increase the difficulty of separating incompatible fluids. The pressure level is reduced to be consistent with the current state of the art for specific types of seals and fluids by utilizing upstream labyrinth or circumferential clearance seals in conjunction with low-pressure return bleeds that allow recirculation to a low-pressure area (fig. 10).

High differential pressures are reduced by maintaining an established pressure level downstream of the primary seal. The primary seal may be drained directly into the turbine hot-gas area (fig. 3) if the fluids are compatible, or a relief valve may be utilized in the drain in conjunction with an additional seal in series with the primary seal (fig. 4) to reduce the differential pressure. It is also possible to maintain a pressure level downstream of the primary seal by pressurizing the drain cavity with an inert purge.

Seals located near pump impellers generally are subjected to excessive pressure oscillations arising from the high-frequency pulses of discharge. Fatigue failure of the seal components such as the secondary-element metal bellows or plastic lip may result if the seal is not protected. The amplitude of pressure oscillations at the seal is minimized by utilizing labyrinths upstream of the seal as damping devices (fig. 4).

2.1.2 Thermal Environment

The low temperature of the cryogenic propellants and the high temperature of the turbine hot gases result in severe temperature gradients in the seal system and across the seal assemblies. The thermal stresses created by the differential thermal expansion/contraction are minimized by designing the seal housings and mating rings to provide minimum restraint to thermal deflections and by using materials with similar thermal expansion/contraction rates. High temperatures at the seal location are minimized by designing the system to isolate the seal in a cooled area. Barriers or labyrinth devices are provided to prevent the direct flow of hot gases onto the seal assembly.

2.1.3 Vacuum Environment

Many rubbing materials tend to wear excessively or fail completely under vacuum conditions because the oxygen and water vapor that are required to form lubricating surface oxides are absent. It is general practice to design the turbopump seal system such that the rubbing seals are not exposed to a hard vacuum. The seal drains are routed to a safe disposal area internal to the turbopump or engine (e.g., the turbine exhaust or pump inlet), or the drains are eliminated and the seal leakage is allowed to vent directly into the turbine (fig. 3). Pressure-relief valves may be used in the seal drain system or a drain cavity purge may be used to ensure a positive pressure in the drain.

Atmospheric drains normally are long enough so that only partial diffusion of the drain-cavity atmosphere will occur. Also, the seal leakage normally is sufficient to maintain a positive pressure in the drain system; therefore, it may not be necessary to provide special allowances for vacuum operation.

2.1.4 Rubbing Speed

High seal rubbing speeds result in excessive heat generation and require additional cooling capacity to prevent thermal failures. Rubbing speeds are minimized by utilizing the smallest possible seal diameter. Compromises in the pump design to reduce the shaft rotational speed are considered when the rubbing speed is excessive. Larger-diameter impellers may be used or the pump operating head and flow relations may be adjusted. For reliable operation, the rubbing-speed/sealed-fluid relations for specific types of seals and fluids must be within the current state of the art.

2.1.5 Cooling and Lubrication

Rotating-shaft seals require adequate cooling to dissipate the heat generated at the seal face by rubbing friction and viscous shear of the sealed fluid. Also, seal materials generally require some form of lubrication for satisfactory performance. The sealed fluid usually provides adequate cooling and lubrication if provisions are made to allow the fluid to flow through the seal cavity. Bleed holes through the impeller from the seal cavity to the impeller inlet, or through the shaft back to the pump inlet, are used where possible. Pumping vanes on the impeller rear shroud also are utilized to ensure recirculation of the sealed fluid. Installation of the seal in a deadend cavity that may allow accumulation of a vapor pocket and result in poor cooling and lubrication is avoided.

Dry-running face-contact intermediate seals (fig. 10(a)) may overheat as a result of lack of cooling and lubrication. Circumferential shaft seals (fig. 6) generally are used for dry-running intermediate seals. Cooling and lubrication of dry-running seals are provided when required by purging with an inert fluid or by injecting a cooling fluid into the seal cavity or onto the seal mating ring. Separate cooling and lubrication systems, which are isolated from the sealed fluid, are used when the cooling fluid is not compatible with the sealed fluid.

Mating rings usually are located toward the sealed fluid with pressure on the outside (fig. 10(a)) in order to ensure flow of coolant across the back surface of the ring for maximum heat transfer to the sealed fluid. Mating rings also are drilled to allow coolant flow through the ring section. Pumping vanes or holes in the mating ring are used to ensure coolant flow around the seal face. However, modification of the mating-ring section by incorporating vanes or holes may result in distortions of the mating surface.

Overheating and distortion of the mating ring may be a problem with two seals rubbing on the same ring (figs. 4 and 8). If space is not limited, series seals on separate mating rings are preferred (fig. 10(a)).

2.1.6 Leakage Drains

Adequate drains are provided in the seal system to ensure safe disposal of the seal leakage. Seal drains are sized to accommodate the maximum anticipated leakage without building up significant back pressure. The drain effective flow area is calculated with conventional flow equations by estimating the maximum anticipated seal leakage and establishing the allowable drain cavity pressure. The drain pressure differential and leakage rate then establish the required drain size.

Seal leakage rate for a cryogenic fluid is calculated on the assumption of liquid conditons upstream of the seal. For calculation of the required drain size, the calculated liquid leakage is then converted to an equivalent gas volume at an assumed temperature. This method provides the most conservative design, since the cryogenic fluid may be partially liquid and partially vapor.

Seal drains for liquid hydrogen are routed to a safe disposal area because of the hazard of mixing hydrogen with atmospheric air. Leaking hydrogen is not allowed to accumulate inside of the vehicle structure. The general practice on liquid-hydrogen turbopumps is to eliminate external drains and allow the seal leakage to vent into the turbine area (fig. 3).

2.1.7 Fluid Separation

Keeping incompatible fluids separated generally is a critical requirement of oxidizer seal systems. The potential explosion hazard of mixing highly reactive oxidizers and fuels dictates the use of a positive system to maintain safe separation. Separation of incompatible fluids on the same shaft normally is maintained by utilizing (1) two face-contact seals to minimize the leakage, (2) separate drains for each propellant to vent the leakage to a safe disposal area, and (3) a purged double circumferential seal (fig. 6), two intermediate face-contact seals (fig. 10(a)), or two intermediate ring seals (fig. 10(b)) with a purge between them to separate the drain cavities. The purge pressure must be high enough to provide a pressure barrier between the drains that will prevent propellant mixing caused by leakage through the intermediate seal. Seal systems that require a purge for fail-safe operation must utilize a fail-proof purge system.

Separation is improved by utilizing a rotating slinger (fig. 6) or shoulders on the intermediate mating ring (fig. 7) to prevent direct impingement of high-velocity fluid and to assist in routing the leakage out through the drain. Slingers are more effective with viscous fluids. Fluids that are subject to capillary action and therefore tend to leak into the purge cavity are prevented from contacting the intermediate seal by use of an adequate reservoir with a gravity drain.

2.1.8 Fail-Safe Provisions

Fail-safe operation is provided for by designing the seal system to allow failure of a single seal without causing destructive failure of the turbopump or aborting a flight mission. The effect of a seal failure is minimized by utilizing labyrinth seals upstream of the primary seal to reduce the propellant leakage and by providing adequate drainage to safely dispose of the leakage. Systems for separating incompatible fluids must utilize a positive intermediate seal to maintain separation of the drain cavities (fig. 6). To ensure a pressure barrier between the drain cavities, the intermediate seal cavity normally is pressurized to a level higher than the maximum value anticipated in the seal drain. Redundant seals in series (fig. 4) are also utilized to prevent excessive leakage in the event of a seal failure.

A theoretical failure analysis is performed to estimate all of the operating parameters and the possible results for each different failure condition. All different modes of operation and variations of performance are considered.

2.1.9 Purge Requirements

To prevent condensation and freezing of moisture, seal cavities in cryogenic systems are purged with gaseous nitrogen or gaseous helium to remove trapped air prior to chilldown.

Hydrogen systems are purged with gaseous helium to remove air that could ignite the hydrogen. Gaseous nitrogen is not used in liquid-hydrogen systems because the system temperature is below the freezing point of nitrogen.

Liquid-fluorine systems are thoroughly purged and dried to remove all traces of moisture and thus preclude any combustible reaction of water and fluorine.

Seal cavities that are exposed to hot gas generated by the combustion of oxygen and hydrogen, which contains significant free moisture, are purged to remove moisture that may accumulate in the seal cavities and then freeze during subsequent chilldowns.

Purged cavities are provided with an inlet and an outlet port to allow the purge gas to flow through the cavity. The purge flowrate and length of time are established to ensure complete removal of air and moisture. After testing, the purge is kept on or air is prevented from re-entering the cavity until the hardware returns to ambient temperature.

2.2 SEAL ASSEMBLY

The seal assembly consists of a group of detail parts or a unitized assembly – including sealing surfaces, provisions for initial loading, and a secondary sealing mechanism – necessary for accomplishing a complete single sealing function. The design problems and performance considerations related to the selection of the seal assembly and its application to the turbopump are presented in the following discussion.

Face-contact seals are used to minimize leakage at pressures up to approximately 500 psi and speeds up to approximately 500 ft/sec; circumferential clearance seals are used for higher pressures and speeds. The low temperatures of the cryogenic propellants require seal types constructed of materials that maintain adequate ductility as the temperature decreases. High surface speeds or long-life applications require low face-contact loads or clearance-type seals. Circumferential clearance seals, particularly labyrinth seals, are used for maximum reliability when the increased leakage is acceptable. The hydrostatic and hydrodynamic seals are considered for effective sealing at high-speed, high-pressure, and long-life conditions.

A comparison of the advantages and disadvantages of current types of turbopump rotating-shaft seals is given in table II, which appears on pages 18 through 21; basic features of the seal types are shown in figures 11 through 25 that follow table II.

2.2.1 Pressure Capability

Pressure levels for primary shaft seals normally are controlled to values less than 500 psig to allow the use of minimum-leakage face-contact seals. Pressure levels greater than 500 psig are reduced by the seal system design methods described in section 2.1.1. Pressures up to approximately 1000 psig are feasible with turbopump-type face-contact seals when the rubbing speed is low, life requirement is short, size is small, and cooling and lubrication are adequate.

The fluid pressures in various seal applications in representative turbopumps are shown in figure 26.* The maximum pressure for current face-contact seals is 500 psig; in most applications, the pressure is approximately 200 psig.

The pressure limit for rubbing-contact-type seals normally is controlled by the capability for balancing the differential pressure forces on the seal face to maintain a reasonable contact load. The seal usually can be designed to withstand the structural loads caused by high pressure; however, it is not feasible to eliminate the increase in face-contact load as the fluid pressure increases. Therefore, the face-contact load increases in proportion to the fluid pressure increase, and the pressure limit is determined by the load-speed-life relationships for specific materials and fluids. Because the face load is proportional to the fluid pressure,

*Figure 26 appears on p. 31.

the relation of fluid pressure and rubbing speed for specific types of seals and fluids provides a guide for evaluating the relative severity of the application before the detail seal designs are established.

The effect of high pressure on face-contact bellows-type seals is more pronounced than the effect on elastomer or piston-ring types because of the change of bellows effective diameter with pressure. The bellows seal can be pressure balanced for a specific pressure range; however, if the pressure variation is great enough to cause the bellows effective diameter to change significantly, the seal pressure balance will be adversely affected. Higher pressure will increase the balance ratio and cause overloading of the seal face. Lower pressure will reduce the balance ratio and the face-contact load and may result in excessive leakage or face separation. Therefore, bellows seals generally are not used in applications with large pressure variations unless face-load control is not critical. Turbopump bellows seals have operated successfully with pressure levels of 500 psig and pressure variations of approximately 100 psi. Bellows-type seals that are designed for high pressure generally incorporate heavy plates with a narrow span to withstand the pressure-induced forces; therefore, the bellows spring rate is higher, and the axial travel may be limited unless additional space is available to allow an increase in the number of convolutions.

Circumferential shaft-riding segmented carbon seals (fig. 20) normally are not used as dry-gas seals in turbopumps above approximately 100 psig because of the difficulty of balancing the radial pressure load. Higher pressures (> 100 psig) may result in high contact loads and excessive wear or heat generation. The circumferential segmented seal can be used at higher pressures with liquids or lubricants because of the hydrodynamic lift effect from the fluid wedge that is developed at the contact surface (similar to the effect in a journal bearing); however, the leakage rate is significantly higher with liquids because the segments tend to wind up and lift away from the shaft surface. Recent experiments indicate that it may be feasible to utilize a hydrodynamic concept (Rayleigh step or spiral groove (fig. 22)) at the contact surface to control the fluid-film thickness.

Circumferential shaft-riding floating-ring controlled-gap seals (fig. 23) are capable of operating at very high pressures (> 1000 psig), because the contact-load increase with pressure is negligible. The seal supports the differential pressure radial load as compressive hoop stress in the floating ring; however, the axial contact load increases with pressure, and the radial force required to reposition the floating ring, if the shaft rotates eccentrically, increases slightly. The pressure normally is limited only by the capability of the floating ring to support the radial load structurally. Large pressure variations may cause excessive changes in clearance gap and can result in failure because of ring seizure on the shaft.

The hydrostatic and hydrodynamic seals (figs. 15 through 18) theoretically are capable of operating at pressures higher than 500 psig, since the differential pressure load is supported by a fluid film; however, these seals are very sensitive to distortions of the interface.

Table II. – Advantages and Disadvantages of Various Types of Rotating-Shaft Seals in Current Use

Seal type	Figure	Advantages	Disadvantages
Face-contact metal bellows	11	Low leakage across face; positive secondary seal; wide temperature range (-423° to 1000°F); materials compatible with toxic and reactive propellants; simplified construction; bellows provides spring load and secondary seal; little space required; materials not age-limited; high reliability.	High cost; welding is critical; stress levels and fatigue life difficult to predict; low fatigue life in extreme vibration and oscillating pressure environments; variation of effective pressure-balance diameter and resultant face load.
Face-contact plastic lip	12	Low-temperature (-320°F) capability; lip provides vibration damping; one-piece nose for minimum face distortion; constant pressure-balance diameter; resistant to extreme vibration environments.	Lip stress level and fatigue life difficult to predict; lip subject to damage and wear by contamination particles; reliability lower than bellows in most applications; lip drag at high pressure.
Face-contact piston ring	13	Wide temperature range (-423° to 1000°F); material compatible with toxic and reactive propellants; materials not age-limited; constant pressure-balance diameter for close face-load control; resistant to extreme oscillating pressure environments; high-pressure capability.	High leakage through secondary seal; wear of piston ring and mating surface can cause seal hangup, excessive drag, and increased leakage; piston ring subject to fretting damage.
Face-contact elastomer	14	Low leakage through secondary; low cost; resistant to extreme vibration and oscillating pressure environments; elastomer provides vibration damping; design highly developed and standardized.	Limited temperature range (-65° to 500°F); materials age-limited; requires lubrication for consistent performance; lubricant may deteriorate or be washed off; subject to hangup or inadequate response due to elastomer extrusion at high pressure; materials and lubricant not compatible with some propellants.

Self-energized hydrostatic face	15	Seal face supported by fluid film from sealed pressure; no rubbing contact during steady-state operation; no external pressurization; no dilution of propellant with purge gas; long wear life; high-speed and high-pressure capability; lift independent of speed.	High leakage; high cost; rubbing contact during transient operation; sensitive to face or mating-ring distortion; sensitive to mixed liquid and vapor fluids; spring load and pressure balance critical; marginal stability; lift dependent on pressure differential.
Externally pressurized hydrostatic face	16	Seal face supported by fluid film from externally pressurized source; no rubbing contact during transient or steady-state operation; long wear life; high-speed and high-pressure capability; low propellant leakage; lift independent of speed and sealed pressure.	Requires external inert pressurizing system or higher pressure propellant source; high cost; dilution of propellant with pressurizing fluid; spring load and pressure balance critical; marginal stability in cryogenic fluids; relatively large space required.
Hydrodynamic face	17	Seal face supported by fluid film developed from hydrodynamic lift forces due to rotation; no rubbing contact during steady-state operation; maximum lift force developed at minimum face clearance; long wear life; high-speed and high-pressure capability; lift independent of sealed pressure; no dilution of propellant with purge gas.	Rubbing contact during startup and shutdown or low-speed transients; face geometry easily damaged by rubbing contact; sensitive to face or mating-ring distortion; sensitive to mixed liquid and vapor fluids; marginal lift force with cryogenic propellants; sensitive to abrasive environments.
Hybrid face	18(a)	Hydrostatic action provides lift at low rotational speeds; hydrodynamic action provides additional lift and stability at high speed independent of sealed fluid pressure; no rubbing contact during transient or steady-state operation; long wear life; high-speed and high-pressure capability.	Sensitive to face or mating-ring distortion; sensitive to mixed liquid and vapor fluids; spring load and pressure balance critical; high static leakage; tolerances on face geometry critical; face geometry easily damaged by rubbing contact; relatively large space required.
Hybrid face	18(b)	Design allows for rubbing contact during transient operation; face contact load reduced at high speed by hydrodynamic lift force; lift independent of sealed pressure; longer wear life and higher speed and pressure capability than rubbing contact seal.	Rubbing contact may cause thermal distortions that destroy hydrodynamic lift; face geometry designed to allow for reasonable wear may not provide significant hydrodynamic lift; face material must be compatible with the face unit load and velocity rubbing limits; amount of hydrodynamic lift force is difficult to predict.

(continued)

Table II. – Advantages and Disadvantages of Various Types of Rotating-Shaft Seals in Current Use (concluded)

Seal type	Figure	Advantages	Disadvantages
Face contact/clearance	19	Seal face travel limited to provide minimum face clearance after initial wear-in at a fixed position; no additional face wear after initial wear-in; high-speed and high-pressure capability after wear-in; simplified construction; high reliability for steady-state repeatable operation.	Leakage dependent on repeatability of the mating ring position; rubbing surfaces may be damaged during wear-in; wear or overheating may occur if the mating ring position is not repeatable; excessive leakage or face overload may occur during startup and shutdown transients.
Circumferential shaft-riding segmented	20	Lower leakage than other types of circumferential shaft seals; segments adjust to diameter and radial location variations to maintain rubbing contact; unlimited axial travel; less sensitive to temperature differentials; wide temperature range (-423° to 1000°F); materials not age limited; little axial space; design highly developed.	Higher leakage than face-contact-type seals; leakage of liquids may be excessive because of hydrodynamic lift of the segments; limited to low pressure because of unbalanced pressure load; frictional torque and horsepower losses are high compared with balanced face seal; complex and critical construction; relatively high cost.
Circumferential shaft-riding arch-bound segmented	21	Lower leakage than floating-ring seal; segments are arch-bound at operating diameter to support pressure load in compressive hoop stress; low contact load for long life, high-speed and high-pressure capability; wide temperature range (-423° to 1000°F).	Higher leakage than conventional segmented seal; difficult to predict exact operating diameter; operating diameter must be repeatable; overheating or damage may occur during wear-in; machining tolerances more critical; complex and critical construction; relatively high cost.
Circumferential hydrodynamic segmented	22	Seal segments supported by fluid film developed from hydrodynamic lift forces due to rotation; higher speed and pressure capability and longer life than shaft-riding segmented seal; lower leakage than floating-ring seal; segments adjust to variations in diameter and radial location.	Higher leakage than shaft-riding segmented seal; rubbing contact during transient operation; conformation of segments to mating ring surface critical; tolerances on segment face critical; face geometry easily damaged by rubbing contact; high cost.

Circumferential floating-ring controlled-gap	23	Unlimited speed capability; high-pressure capability; negligible horsepower losses; negligible wear; unlimited axial travel; wide temperature range (-423° to 1000°F); low clearance gap; simple construction; less axial space; high reliability; effective for liquids, gases, and mixtures.	Higher leakage than segmented shaft seal; clearance gap sensitive to temperature differential between ring and shaft; clearance gap may be decreased by radial deflection of the ring caused by differential pressure load; subject to fretting damage.
Circumferential clearance labyrinth	24	Consistent predictable leakage; maximum reliability; unlimited speed, pressure, life, and axial-travel capability; minimum horsepower losses; maximum temperature range; no rubbing; materials not critical; simple construction; low cost; wide temperature range (-423° to 1800°F).	High leakage; clearance gap must be large enough to allow for mislocation tolerances and shaft radial movements.
Circumferential wear-in labyrinth	25	Lower leakage than clearance labyrinth; high reliability; unlimited speed, pressure, and life; materials allow wear-in for minimum clearance gap.	High leakage; torque may be high during wear-in; transient temperature differentials may cause high drag torque after wear-in; damage may occur during wear-in; wear-in materials expensive.

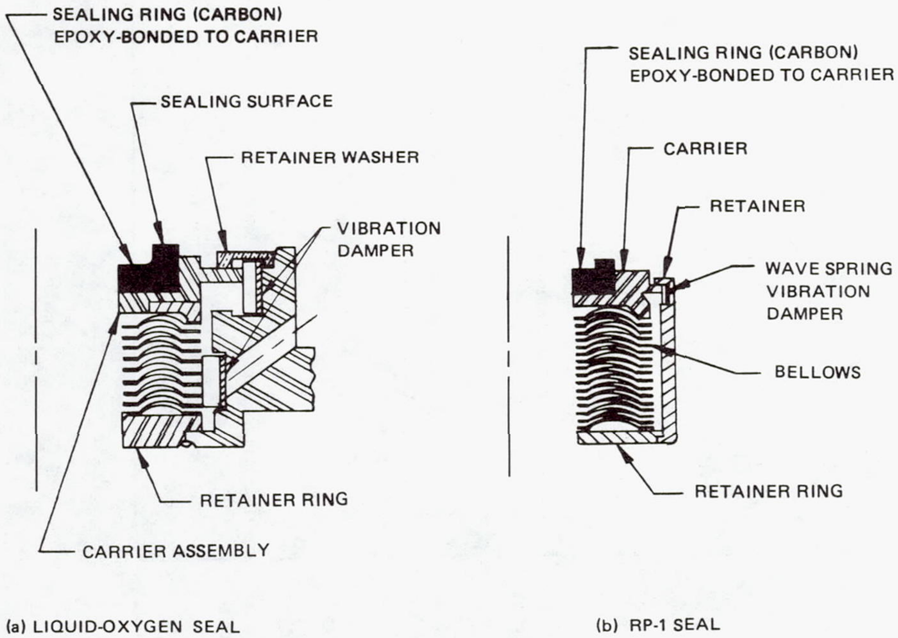


Figure 11. — Face-contact metal-bellows seal.

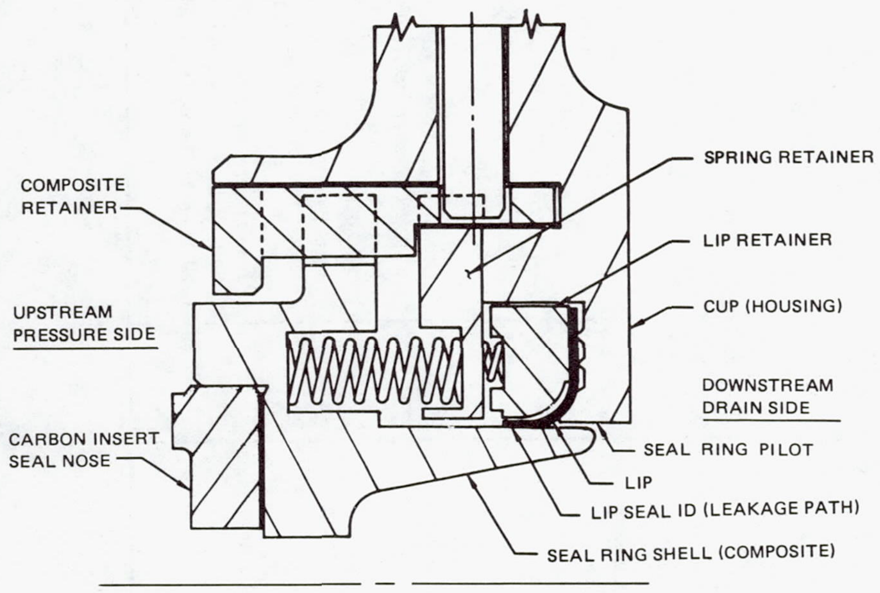


Figure 12. — Face-contact plastic-lip seal.

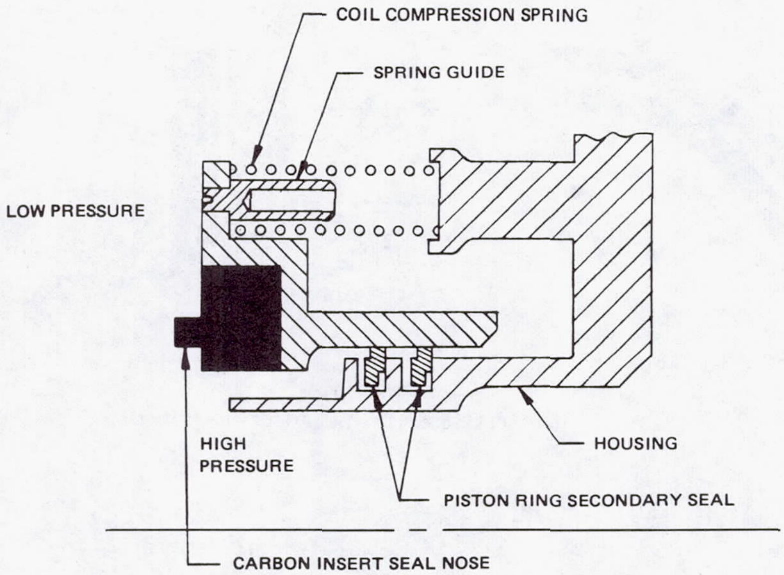


Figure 13. — Face-contact piston-ring seal.

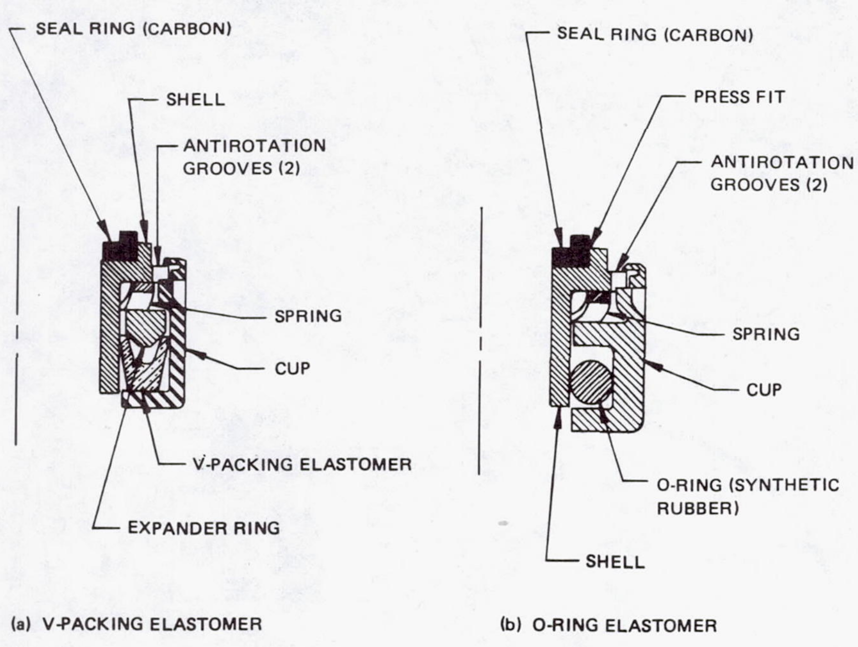


Figure 14. — Face-contact elastomer seal.

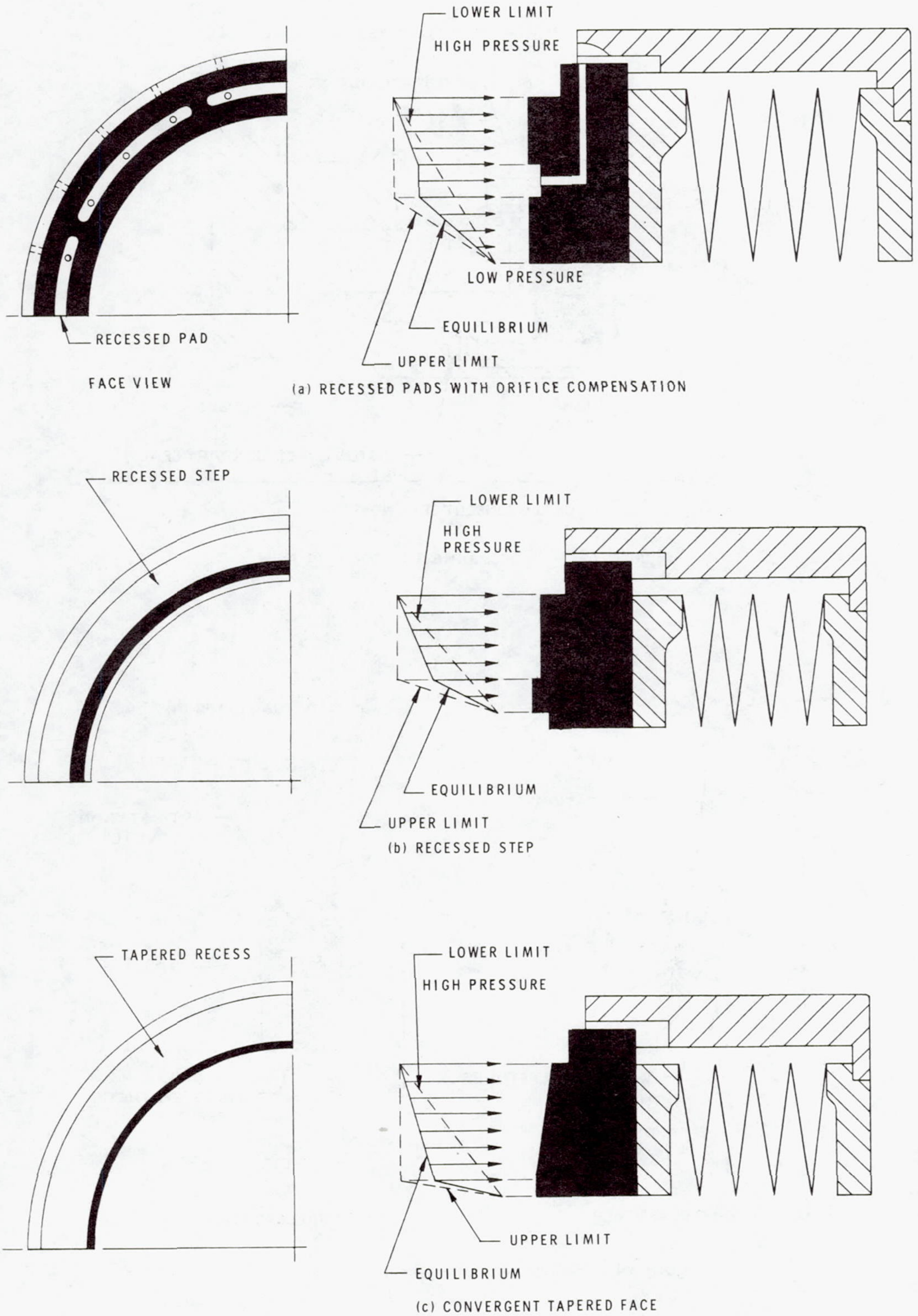
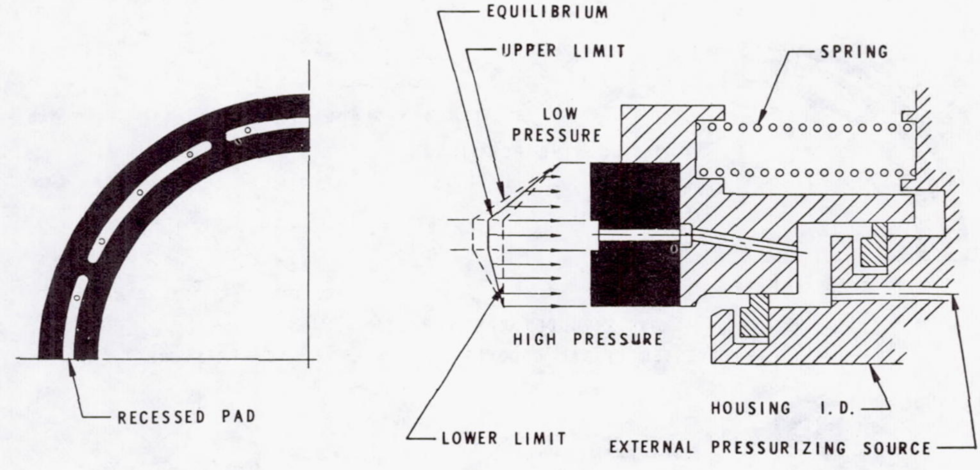
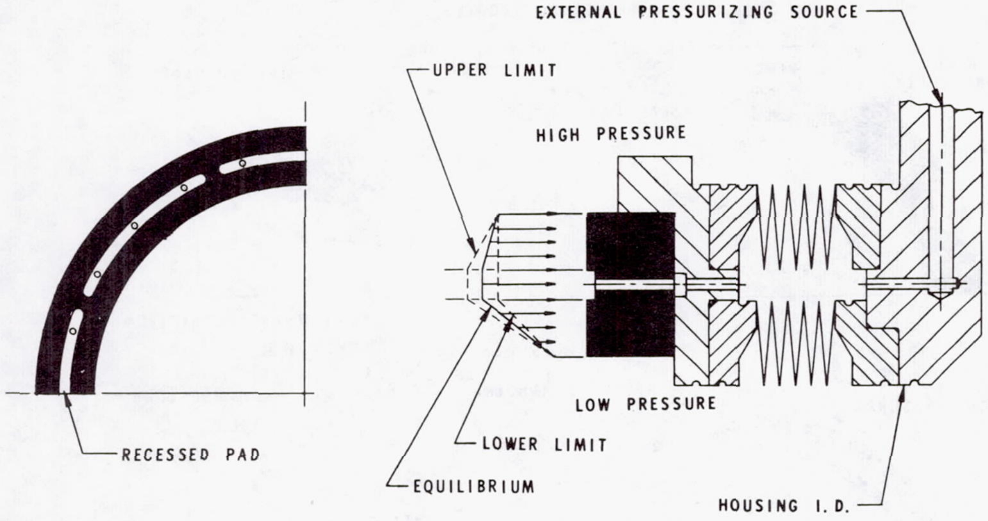


Figure 15. – Three configurations for a self-energized hydrostatic face seal.



(a) PISTON-RING SECONDARY ELEMENT



(b) BELLOWS SECONDARY ELEMENT

Figure 16. – Two configurations for an externally pressurized hydrostatic face seal.

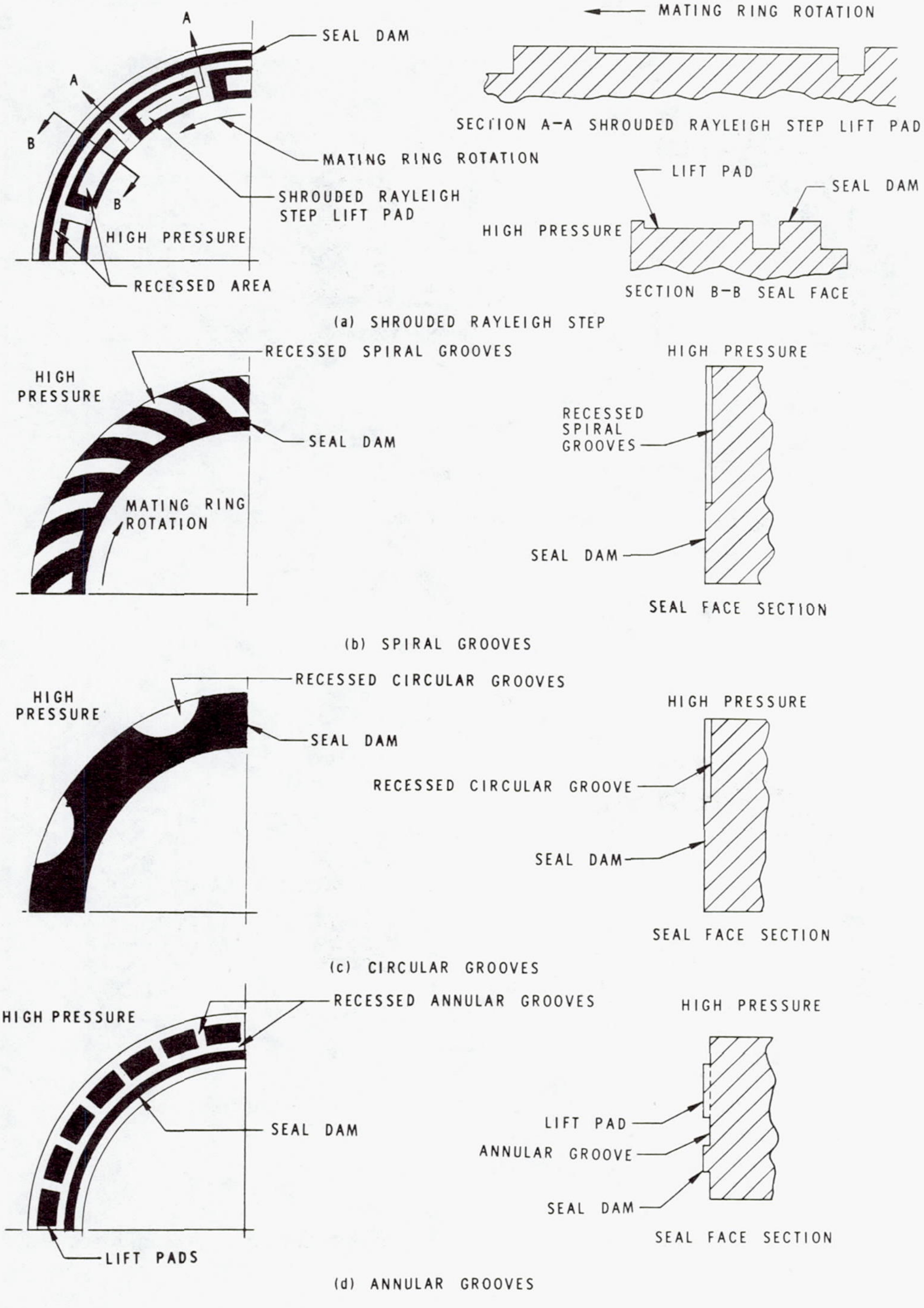


Figure 17. – Four configurations for a hydrodynamic face seal.

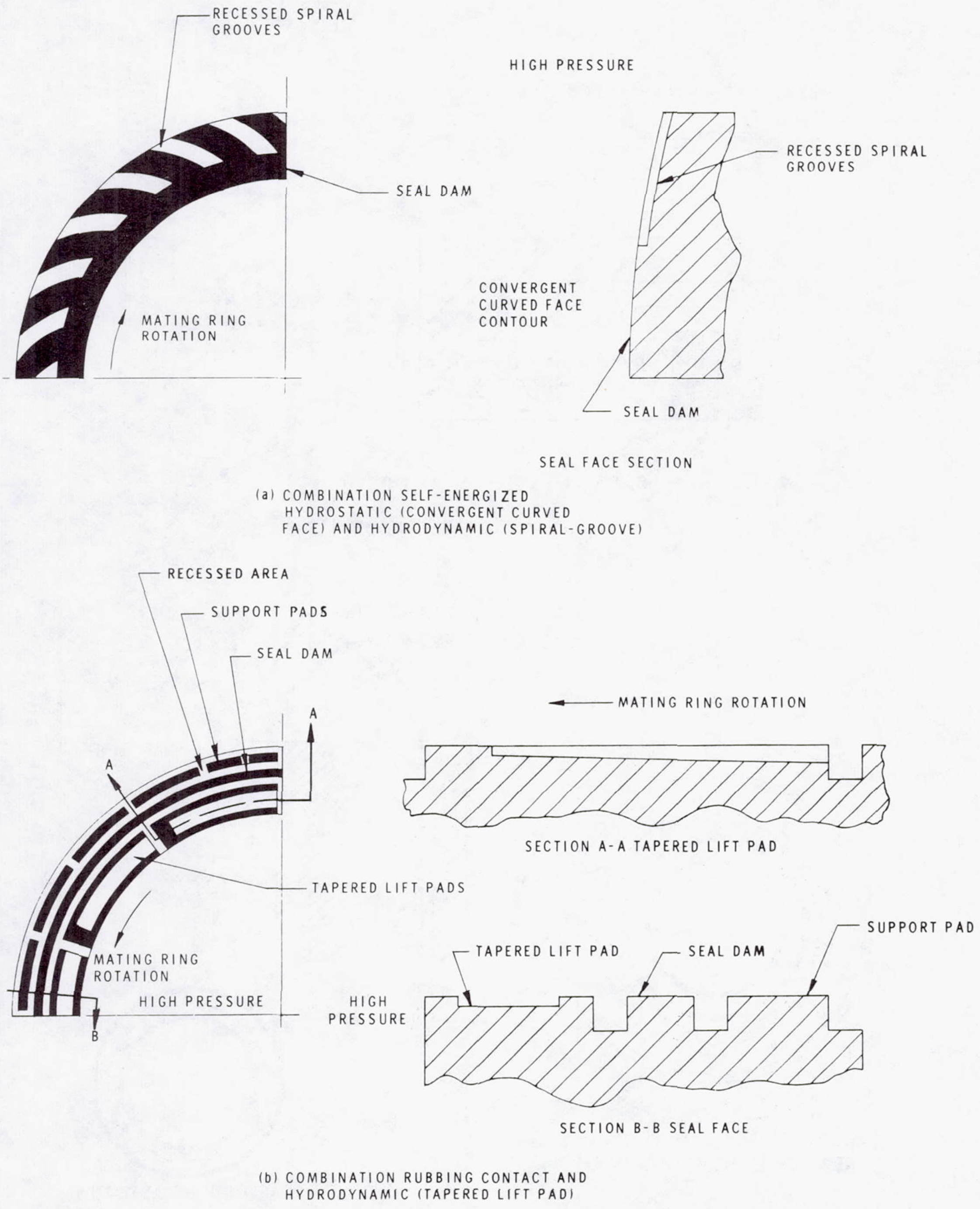


Figure 18. – Two configurations for a hybrid face seal.

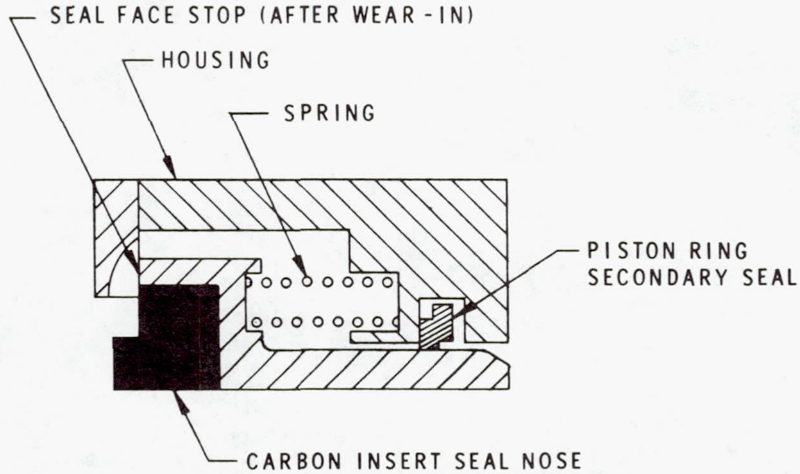


Figure 19. — Face contact/clearance seal.

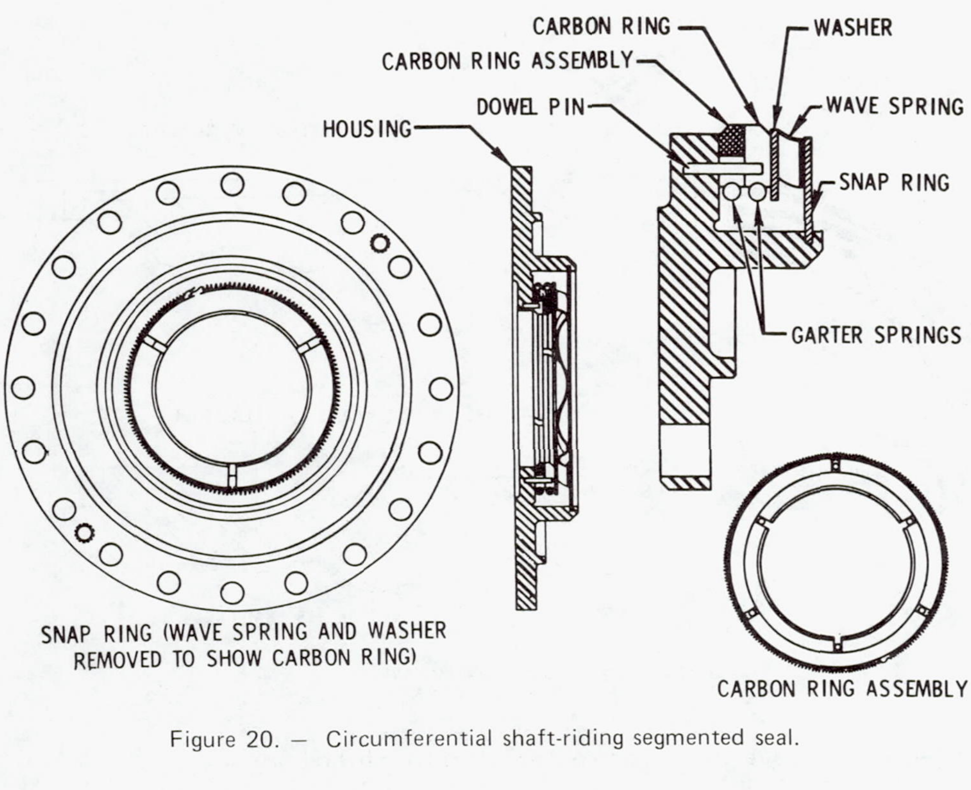


Figure 20. — Circumferential shaft-riding segmented seal.

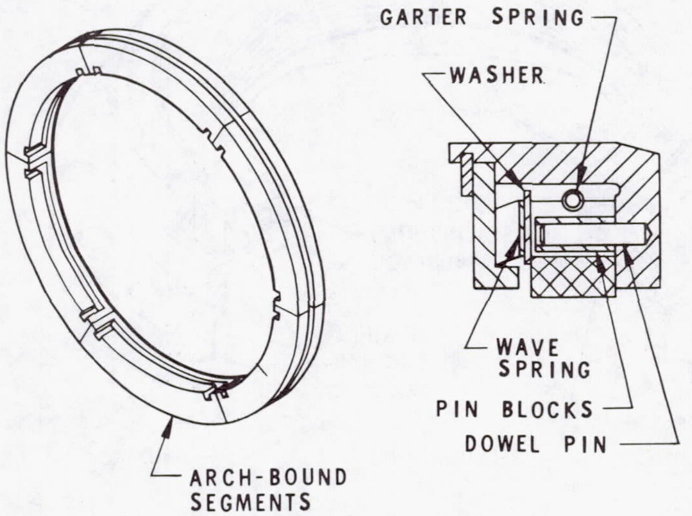


Figure 21. — Circumferential shaft-riding arch-bound segmented seal.

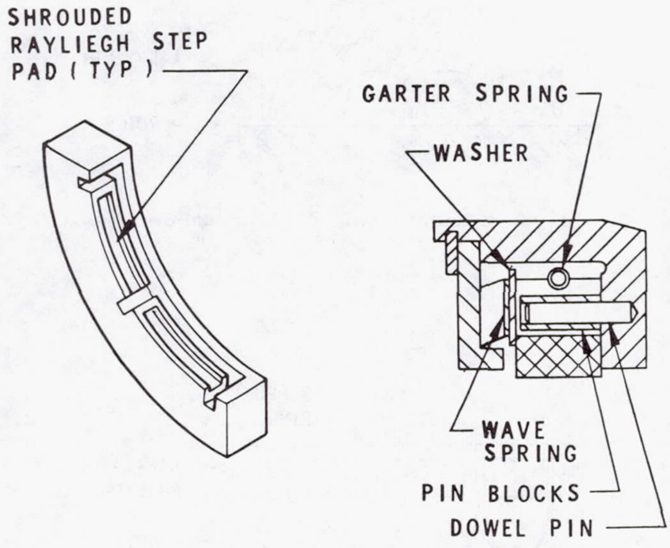


Figure 22. — Circumferential hydrodynamic segmented seal.

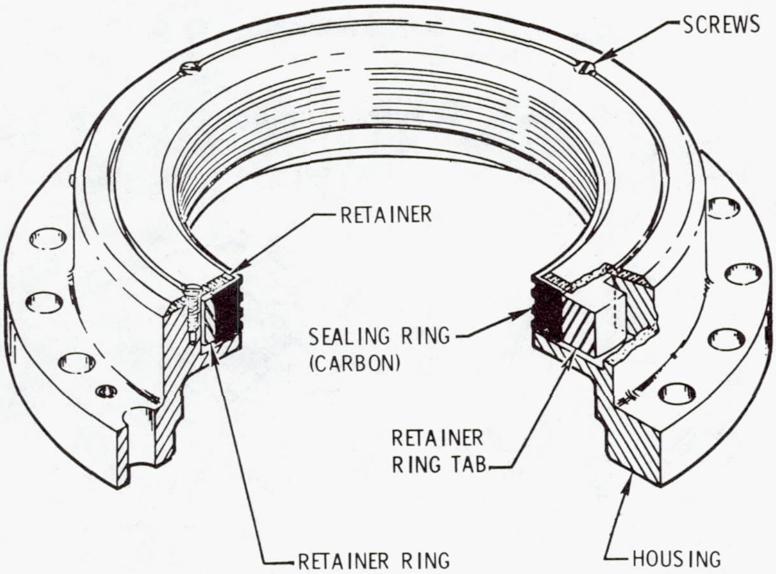


Figure 23. – Circumferential floating-ring controlled-gap seal.

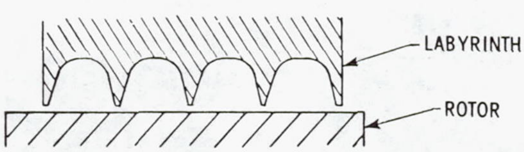


Figure 24. – Circumferential clearance labyrinth seal.

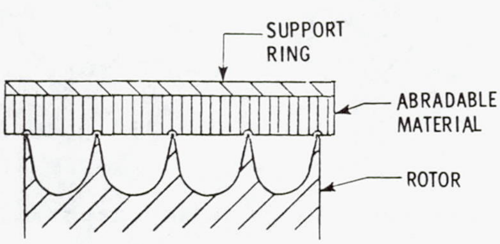


Figure 25. – Circumferential wear-in labyrinth seal.

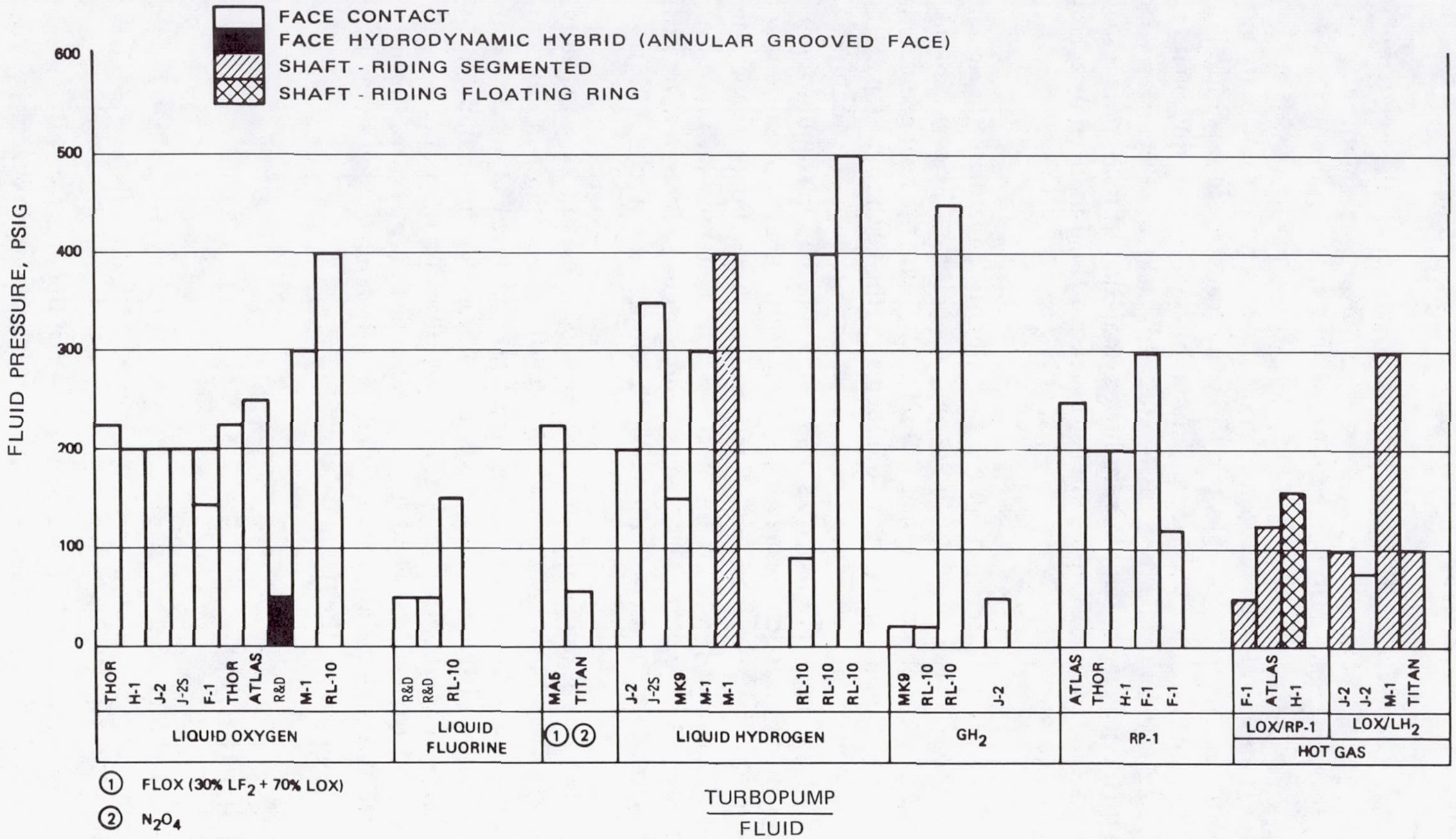


Figure 26. – Seal fluid pressures in representative turbopumps.

2.2.2 Temperature Capability

The use of cryogenic fluids limits the choice of seal materials because the low temperatures induce brittleness or loss of ductility; frequently temperature is the determining factor in the selection of the seal type. The most successful type of seal for cryogenics has been the face-contact welded bellows (fig. 11). Face-contact seals with a plastic (Kel-F or Mylar) lip secondary seal (fig. 12) also have been used successfully; however, their reliability generally is lower. Metal-piston-ring secondary seals (fig. 13) are capable of operation at low temperature; however, the greater leakage and drag of the piston ring may not be acceptable. Shaft-riding segmented carbon seals (fig. 20) also have been used to seal cryogenic liquids, but the leakage is significantly higher than that of face-contact seals, and the segmented seals are limited to lower pressure because of the load induced by unbalanced pressure.

High temperature of the sealed fluid normally is not a limiting factor in the selection of turbopump seals, except as it is related to the cooling capacity of the seal rubbing faces. Turbopump seals generally depend on the sealed fluid to dissipate the heat generated at the seal face; therefore, as the temperature of the fluid increases, the cooling capacity decreases. High temperatures also limit the choice of seal materials because of structural effects, thermal degradation of the material properties, and oxidation. The maximum temperature of the turbine hot gas in the seal area usually is less than 1000°F, a temperature that allows the use of conventional hot-gas seals (e.g., segmented-carbon, floating-gap, labyrinth, welded-bellows, and piston-ring types) if the heat generated at the seal interface can be dissipated satisfactorily. Metal-bellows face seals with carbon inserts require sufficient interference at the insert OD to ensure retention at high temperature (sec. 2.3.2.5).

Severe temperature gradients through the seal system and across the seal assembly occur when the cryogenic fluids and hot gases are sealed along the same shaft in adjacent areas. Additional temperature gradients are caused by the heat generated at the seal interface. Thermal stresses caused by differential expansion or contraction are minimized by using materials with similar expanion/contraction rates and by allowing the deflections to occur with a minimum amount of restraint. The thermal deflections are allowed for by making initial clearances sufficient to prevent excessive thermal interferences and by adjusting the seal dimensions to compensate for the thermal deflections.

2.2.3 Speed Limitations

The surface speed of a rubbing-contact type of seal normally is limited by the heat generation at the seal interface and the cooling capacity of the interface materials and surrounding environment. The primary source of heat generation is the friction torque at the seal interface caused by the contact load and friction of the interface rubbing materials. Additional heat is generated by viscous shear of the sealed fluid; however, this heat normally is not significant on turbopump rubbing-contact seals, which usually operate with boundary lubrication because of the low viscosity of cryogenic fluids. Rubbing-contact seals running with more viscous fluids (RP-1 fuel, lube oil) generally operate with hydrodynamic lubrication; the heat generation due to viscous shear therefore becomes significant and is considered.

The noncontact hydrodynamic and hydrostatic seals normally are not limited by surface speed, because the seal face and rotating mating surface are separated by a fluid film; however, the heat generated by viscous shear must be considered, since it may limit the speed because of the sensitivity of the seal interface to thermal distortion. The higher leakage rate of this type of seal normally provides adequate cooling capacity, particularly with cryogenic fluids.

The rate of heat dissipation from the seal interface determines the resultant interface temperature and is equal in importance to the heat generation rate in establishing the speed limit. The analytical methods used to calculate the temperature profile and thermal distortions in the seal face and mating ring are given in references 6 and 7.

The maximum allowable surface speed normally is established by the temperature limit at the seal interface required to prevent thermal failure of the interface materials or structural failure of the seal components and mating ring. In some cases, the limit may be established by the temperature required to prevent vaporization or thermal decomposition of the sealed fluid.

The structural loads on high-speed seals caused by centrifugal force also are considered, because dynamic deflections of the mating-ring surface can adversely affect the interface pressure profile. The centrifugal stresses may be significant on high-speed mating rings as a result of the decrease in yield strength of some materials at high temperature. The thermal loads due to the rubbing-surface temperature gradient may also cause distortions of the seal interface. Failure will occur if the temperature gradient causes interface distortions and results in higher contact loads that compound the heat generation problem.

The values for rubbing speeds in various seal applications in representative turbopumps are shown in figure 27. The highest speeds (up to 450 ft/sec) are required on liquid and gaseous hydrogen and hot-gas applications. With liquid oxygen and RP-1, rubbing speeds usually are lower than 200 ft/sec.

The heat-generation rate on rubbing-contact seals is a direct function of the resultant contact load (spring load plus pressure closing force minus pressure opening force), coefficient of friction of the rubbing materials, and the velocity of the rubbing surfaces, as shown in the following equation:

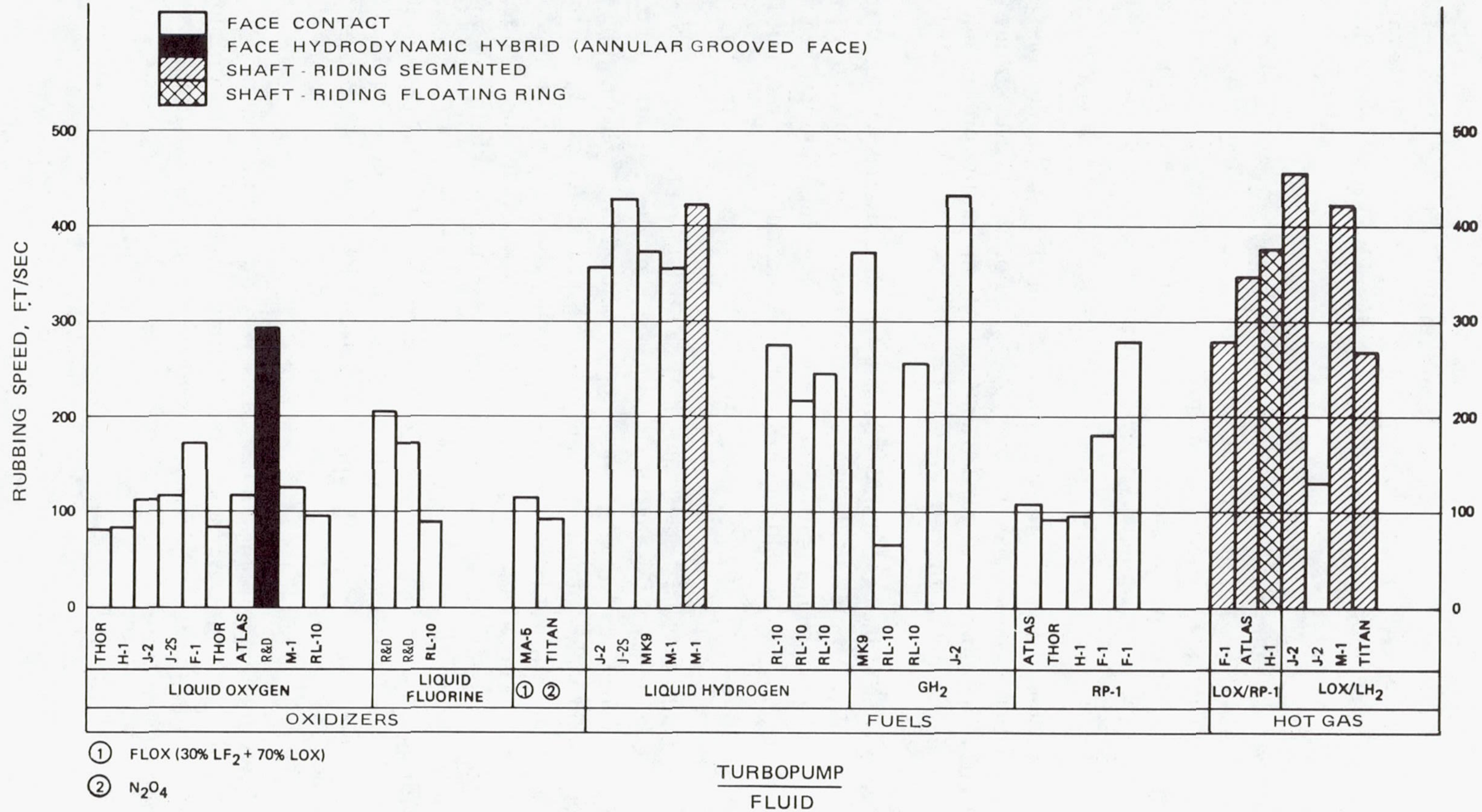


Figure 27. — Values for seal rubbing speeds in representative turbopumps.

$$\dot{q} = \frac{(FV)\,f}{J} \tag{1}$$

where

$\dot{q}$ = heat generation rate, Btu/sec

F = resultant total contact load, lbf

V = surface speed, ft/sec

f = coefficient of friction

J = 777.6 ft-lbf/Btu

The equation can also be expressed in terms of heat-generation rate per unit area and contact unit load:

$$\dot{q}' = \frac{(PV)\,f}{J} \tag{2}$$

where

$\dot{q}'$ = heat generation rate per unit contact area, Btu/sec-in.^{2}

P = contact unit load, psi

The contact load is a function of the seal spring load, pressure closing force, interface pressure profile, and hydrodynamic lift. The spring load and pressure closing force are known values. The effective interface pressure profile can vary from approximately 0.2 to 0.8 times the pressure differential. The hydrodynamic lift can vary from zero to a value sufficient to maintain fluid-film face separation. The coefficient of friction is related to the lubrication and hydrodynamic lift; it can vary from approximately 0.05 to 0.4 for the commonly used seal materials.

Because the heat generation is proportional to the product of face load and velocity, and because the speed limit normally is established by the interface temperature limit, it is convenient to establish a face load-velocity relation to provide a design guide for establishing the face load and surface speed limits. This simple relation does not account for all of the variables; therefore, a separate relation must be established to allow a basis of comparison for each seal type, face material combination, and sealed fluid. If the coefficient of friction and the heat-transfer capacity of the seal materials and surrounding environment are similar, the load-velocity relation can be utilized with reasonable accuracy.

Load-velocity relations have been established for turbopump seals on the basis of unit face load (PV factor) and face load per unit circumferential length (FV factor). The FV factor is proportional to the total rate of heat generation and is independent of face area. The PV factor is proportional to heat generation per unit area and is a better indication of heat-transfer capacity. The FV relation is useful for maintaining practical limits and also is convenient for relating the face load requirements to other design parameters (spring load, secondary seal friction, inertia forces, pressure forces).

The values for FV (lbf/in. $\times$ ft/sec) and PV (lbf/in.^{2} $\times$ ft/sec) factors for face-contact seals in various applications in representative turbopumps are shown in figures 28 and 29.

2.2.4 Wear Life

The turbopump-seal life requirements in operational engines are low ($\approx$ 2 hr) in comparison with life requirements in the SSME (10 hr) and in most other applications (500 to 10 000 hr). Therefore, relatively high seal wear rates can be accepted and may be the only feasible compromise for some of the more severe applications. The average wear rate of the carbon seal face on the liquid-oxygen and liquid-hydrogen seals in the J-2 turbopumps ranges from 0.005 to 0.010 in./hr. The variation in measured wear rates for the same type of seals may be as much as $\pm$ 100 percent.

Allowance is made for the high wear rate by designing the seal contact face with sufficient height to wear away while continuing to maintain an effective seal for the required life. The carbon nose height on most turbopump seals is approximately 0.050 in., which would provide a minimum wear life of 2.5 hours at a maximum wear rate of 0.020 in./hr. Usually, it is not practical to obtain longer wear life by using greater nose heights, because of the structural weakness and distortion of the relatively thin cylindrical sections at high pressures. Also, it may not be feasible to design the seal with sufficient axial travel to allow for the additional movement required to compensate for greater seal face wear.

Seal wear life of more than 10 hours in liquid oxygen has been demonstrated on a combination hydrodynamic and rubbing-contact face seal (table I). The seal face was grooved to provide additional hydrodynamic lift at high speed to reduce the face contact load. The hydrodynamic/hydrostatic noncontact seals potentially are capable of much greater wear life; however, the reliability of hydrodynamic/hydrostatic seals has not been demonstrated with rocket engine propellants. Noncontact types of seals are considered for turbopump applications that require a wear life greater than approximately 4 hours, except when the PV factor is low and good lubrication is available.

The wear life of rotating seal faces is highly dependent on the lubrication available at the seal interface. Fluids such as RP-1 and lube oil, which are capable of developing hydrodynamic lubrication at the seal interface, make possible relatively long wear life.

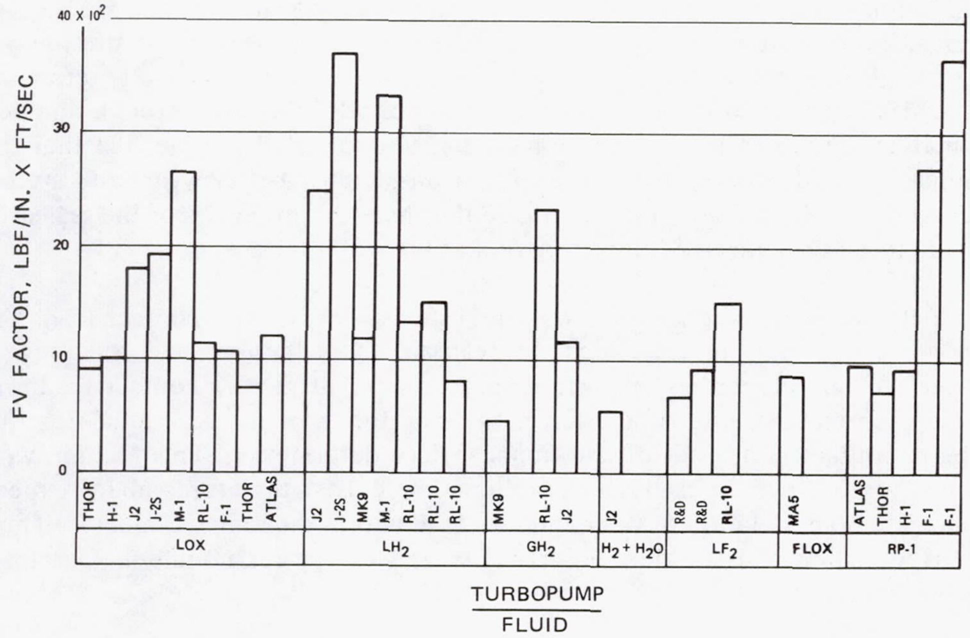


Figure 28. — Values for FV factor for face-contact seals in representative turbopumps.

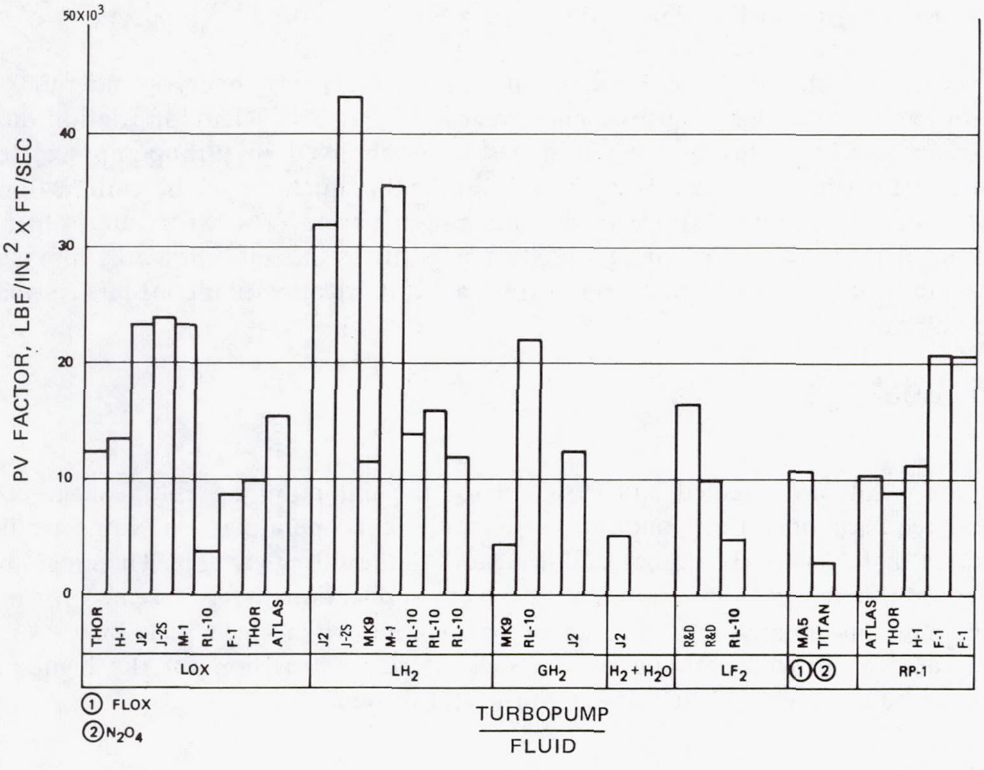


Figure 29. — Values for PV factor for face-contact seals in representative turbopumps.

Cryogenic fluids such as liquid oxygen and liquid hydrogen normally do not develop hydrodynamic lubrication or fluid-film interface separation, because of the low viscosity and change of fluid state (liquid to vapor) across the seal interface. Therefore, rubbing contact generally exists, with boundary lubrication or self-lubrication from a film deposited on the mating surface. The wear rate may be a function of the oxide film that forms on most metals exposed to oxygen and moisture. Considerably better wear rates are obtained by selecting material combinations that utilize the chemical properties of the sealed fluids to assist in the formation of a self-lubricating film at the seal interface.

It is generally accepted that seal face wear rate increases with face contact load. The wear phenomenon generally is explained as caused by adhesion and shearing of the contact-surface microasperities; therefore, higher wear at greater contact load would be theoretically correct because of the larger shearing forces of the additional asperities that would be in contact as a result of additional surface deformation. This relation would not necessarily apply when hydrodynamic lubrication exists; therefore, many experimental results indicate no correlation of wear and face load within the ranges evaluated. The general practice is to compromise other requirements and design for minimum face load when longer life is required.

The effect of rubbing speed on wear rate, when converted to wear per distance traveled, does not appear to be significant, except for the effects of the resultant surface-temperature increase. The primary effect of higher speed is to increase the heat-generation rate. The design objective for long-life seals is to minimize the PV factor.

Most wear and friction measurements on carbon materials in cryogenic fluids or dry conditions indicate a significantly higher wear rate and coefficient of friction during the initial run-in period. This phenomenon has been observed in turbopump seal tests and generally is explained as caused by the self-lubricating qualities of the carbon film that is deposited on the mating surface during the run-in period. The wear rate is high during formation of the carbon film and gradually tapers off as the self-lubricating film is formed. Materials in rubbing contact that do not form a self-lubricating film continue to wear at the high initial rate.

2.2.5 Leakage

Turbopump seals are selected and designed for the minimum possible leakage consistent with the required operating conditions. Generally, the operating conditions are the fixed parameters that control the design, and leakage is the resulting variable. The most important consideration is reliable operation, and in severe applications, seals designed for minimum leakage may fail prematurely because of overheating. High-speed seals may require high leakage rates to provide cooling of the seal interface. Provision for the higher leakages usually can be made with proper design of the seal system.

Rubbing-contact seals with low-speed, low-pressure, and short-life requirements may utilize high spring loads and large pressure-balance ratios to ensure a high face-contact load for minimum leakage; however, as the speed and life requirements increase, the face-contact load must decrease until the point is reached where face separation occurs or until a controlled clearance is maintained at the seal interface. The seal leakage increases as the face-contact load decreases or as the face separation increases, as a result of the larger effective leakage path. The effects of thermal, pressure, and centrifugal distortions of the seal interface may have more influence on leakage than the contact load. In some cases, lower leakage may result from a lower face-contact load, because less thermal distortion is caused by heat generation at the seal interface.

Seals designed for minimum cryogenic fluid leakage normally are face-contact welded-bellows types with the maximum face-contact load allowed by the load-velocity relations for specific materials and fluids. For less severe temperature applications (–65° to 500°F), elastomeric secondary seals may be selected instead of the welded-bellows component. A thorough heat-transfer and stress analysis of the seal face and mating ring usually is performed to minimize seal interface distortions that can cause excessive leakage.

Effective sealing at load-velocity-life relations greater than those allowed by the current state of the art for rubbing-contact seals may require hydrostatic or hydrodynamic seals. The other noncontact seal types (e.g., floating ring and labyrinth) normally will not provide adequate control of leakage for a primary liquid application but may be satisfactory in hot-gas applications or in situations where a backup seal to control leakage can be provided.

Various theoretical methods to predict seal leakage have been developed, ideal conditions being assumed. Most leakage theory is based on steady-state laminar flow through very small uniform flow channels with full fluid/film interface separation. Experimental studies have indicated reasonable correlation with the theory when the variables are known and adequately controlled. However, the variables are much more difficult to control on rocket engine turbopump seals because of the extreme thermal gradients and two-phase fluid conditions; therefore, the theoretical relations must be supplemented with empirical data for practical solution.

Theoretical prediction of seal leakage requires knowledge of the leakage-path geometry and fluid condition. The geometry of rubbing-contact seals usually is altered by the wear process; therefore, any predictions based on initial surface measurements are not valid after wear-in. Also, the effect of thermal distortion on cryogenic seals generally is much greater than the effect caused by the normal variation of surface finishes. Experimental measurements have indicated that the effects of thermal distortion due to chilldown to –320°F can increase the static leakage rate by as much as 500 percent. Additional interface thermal distortion caused by heat generation would be expected during high-speed rotation. As noted, rubbing-surface temperatures in excess of 1000°F have been measured on liquid-oxygen seals where the fluid environment was at –297°F.

The condition of cryogenic fluids at the seal interface rarely is known with any accuracy, and because of heat transfer from the rubbing surface, the fluid usually changes from partial liquid to vapor as it flows across the interface. Because the leakage is proportional to the fluid viscosity and density, the difference between liquid and vapor is significant. The viscosity variation with pressure and temperature is given for oxygen in figure 30 and for para-hydrogen in figure 31. The vaporization process may create higher pressure regions across the seal interface and thereby affect the pressure profile. The flow process can change from laminar incompressible-liquid flow to compressible-gas choked or turbulent flow.

The theoretical methods for predicting seal leakage are discussed in reference 8. Flow regimes ranging from molecular flow to laminar flow are discussed for liquids and gases. An empirical relation between initial surface geometry measurements and static-leakage-path effective gap is established.

A more sophisticated theoretical approach for predicting seal leakage, which considers the effects of misaligned seal faces, interface waviness, fluid inertial forces, and interface fluid film cavitation, is presented in reference 9. The analysis assumes a full fluid film within the interface clearance space and a known leakage-path effective gap. Theoretical methods are presented for predicting leakage in the turbulent-flow regime and in the extended regions of the laminar-flow regime. The effects of rotationally induced turbulence are discussed. This theory applies to most high-pressure seals for cryogenic propellants because of the high leakage rates, high speeds, and low fluid viscosity.

The flow regimes for leakage are defined by either the Reynolds number or the molecular mean free path, as shown in figure 32. The flow regime must be established to determine the applicable theoretical leakage relationship. Because the leakage flow is required for the Reynolds number calculation, it is necessary to assume a flow regime and iterate the leakage and Reynolds number calculations.

The molecular regime exists when the molecular mean free path is equal to or greater than the effective leakage gap. As the leakage gap is increased relative to the mean free path, the flow enters the transition regime. This regime is the combination of molecular and laminar (ref. 10). The laminar regime is entered when the leakage gap is increased to the point where molecular collisions with the wall are no longer significant. The transition from laminar to turbulent flow may be caused by either increased leakage or higher rotational speeds. Rotationally induced turbulence may cause the otherwise laminar leakage to become turbulent (ref. 9).

The flow regime criteria are summarized below (refs. 8 through 11):

$$\text{Molecular regime:} \quad \frac{\lambda}{h} \geqslant 1.0$$

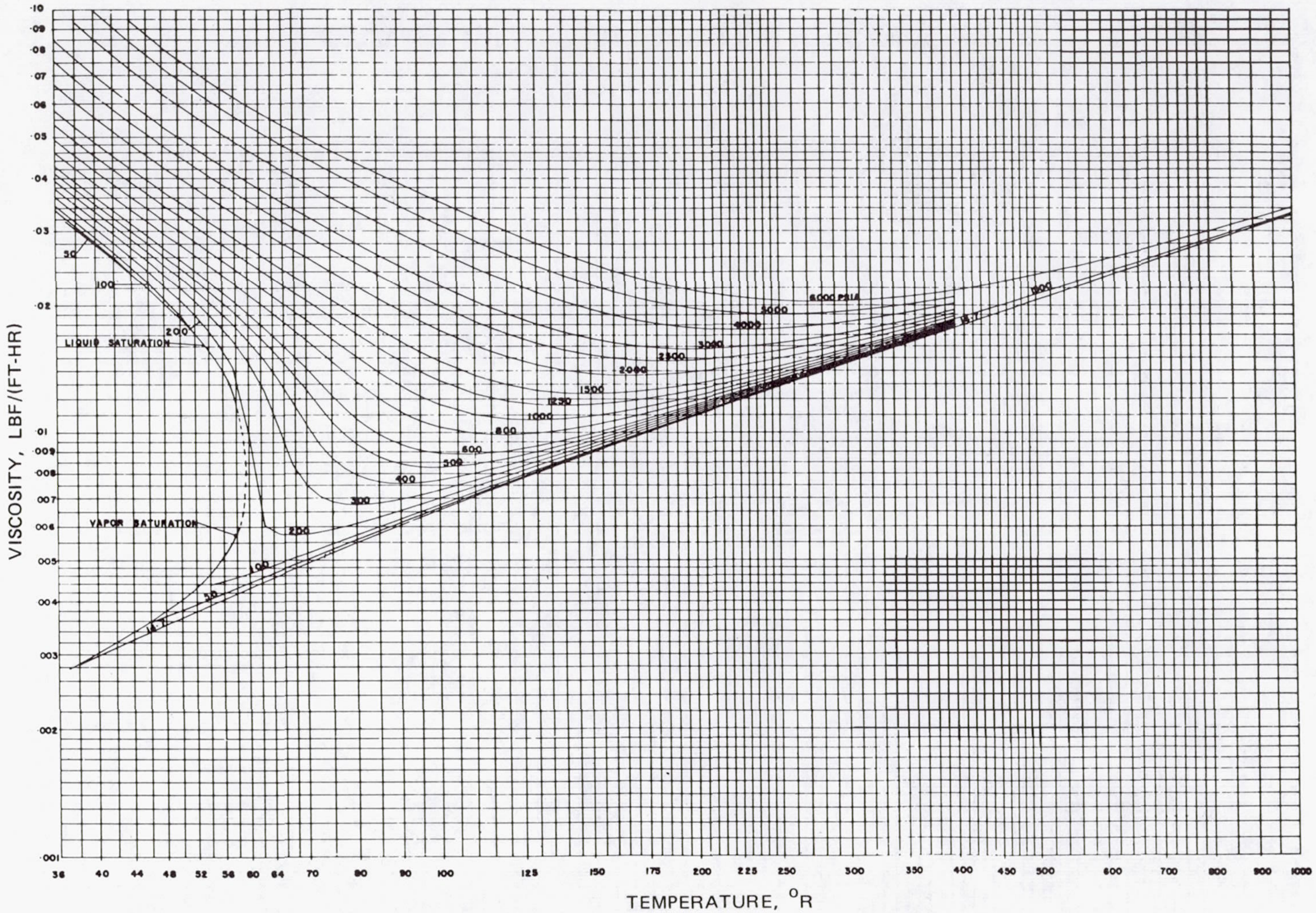


Figure 30. — Viscosity of oxygen as a function of temperature and pressure.

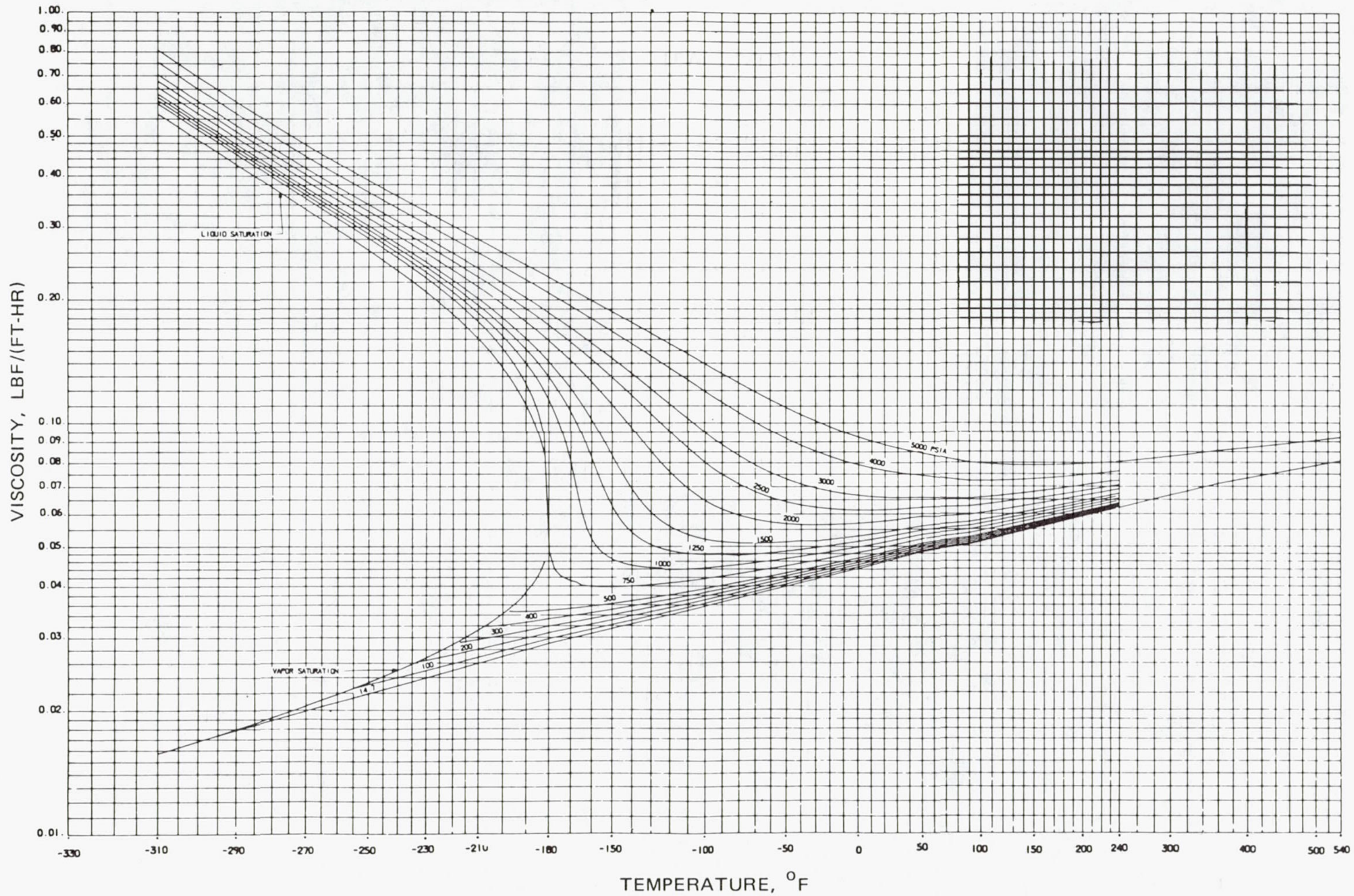


Figure 31. – Viscosity of para-hydrogen as a function of temperature and pressure.

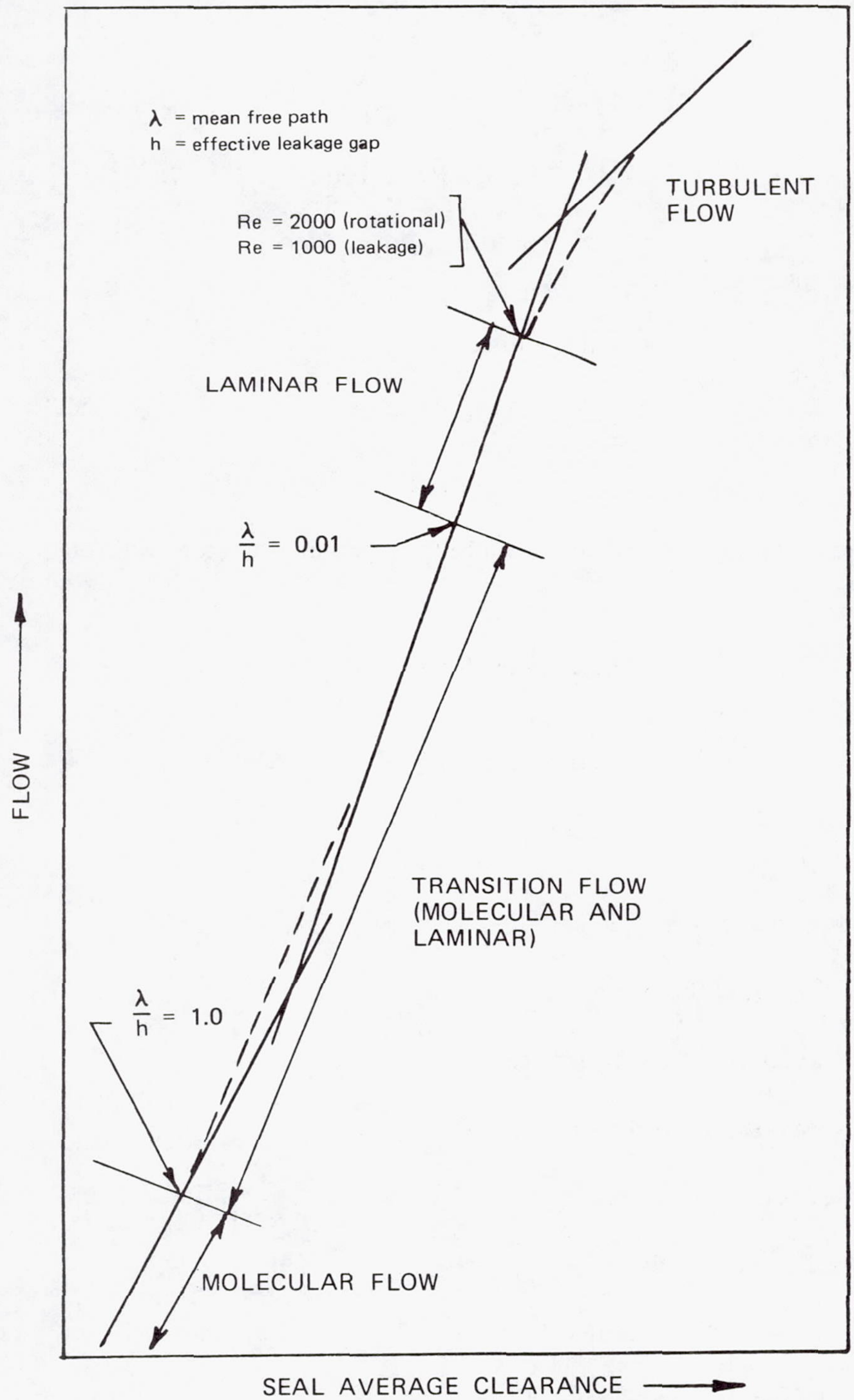


Figure 32. — Flow regimes in seal leakage.

Transition regime: $\frac{\lambda}{h}$ = 1.0 to 0.01

Laminar regime: $\frac{\lambda}{h} \leqslant 0.01$

Leakage Re $\leqslant$ 1000

Rotational Re $\leqslant$ 2000

Turbulent regime:

Leakage Re $\geqslant$ 1000

Rotational Re $\geqslant$ 2000

The Reynolds number is defined by the following equations (refs. 8 through 11):

$$\text{Leakage Re} = \frac{Uh}{\nu} = \frac{Q}{\pi \bar{d} \nu} = \frac{\dot{w}}{\pi \bar{d} \mu} \tag{3}$$

$$\text{Rotational Re} = \frac{\bar{r} \omega h}{\nu} = \frac{2\pi \, N \bar{r} \, h}{\nu} \tag{4}$$

where

λ = molecular mean free path at average pressure, in.

h = effective leakage gap, in.

U = leakage flow velocity, in./sec

ν = kinematic viscosity, in.^{2}/sec

μ = viscosity, lbm/in.-sec

Q = volume flowrate, in.^{3}/sec

$\dot{w}$ = weight flowrate, lbm/sec

$\bar{d}$ = seal face average diameter, in.

$\bar{r}$ = seal face average radius, in.

ω = angular velocity, rad/sec

N = rotational speed, rev/sec

Molecular flow. – The molecular regime in general does not apply to turbopump seals because the effective leakage gap is relatively large. The molecular mean free path of liquids generally is sufficiently short to ensure flow beyond the molecular regime for most practical seals; therefore, the theory presented is for compressible-fluid flow.

The leakage flow for compressible gas is given by the following equation (ref. 8):

$$\dot{w} = \frac{0.532\,\lambda\,\bar{P}\,(P_2 - P_1)\,\bar{b}h^2}{RT\,\mu_a\,L} \tag{5}$$

where

$\bar{P}$ = average pressure, $(P_1 + P_2)/2$, psia

P_2 = upstream pressure, psia

P_1 = downstream pressure, psia

$\bar{b}$ = seal face average circumferential length, in.

R = specific gas constant, (in.-lbf)/(lbm-°R)

T = absolute temperature, °R

μ_a = absolute viscosity, lbf-sec/in.^{2}

L = seal face radial length, in.

Transition flow. – The transition regime generally applies to turbopump seals only for leakage of low-pressure compressible gas under static conditions. The leakage in this regime is equal to the sum of the laminar flow and the molecular flow. The molecular correction usually is negligible and may be neglected in the upper region of the transition regime. The leakage flow for compressible gas is given by the following equation (ref. 8):

$$\dot{w} = \dot{w}_{laminar} + \dot{w}_{molecular}$$

$$= \frac{\bar{b}h^3 (P_2^2 - P_1^2)}{24RT \mu_a L} + \frac{0.532 \lambda \bar{P} (P_2 - P_1) \bar{b}h^2}{RT \mu_a L} \qquad (6a)$$

With simplified arrangement of terms:

$$\dot{w} = \frac{\bar{b}h^3 (P_2^2 - P_1^2)}{24RT \mu_a L} \left(1 + 6.38 \frac{\lambda}{h}\right) \qquad (6b)$$

Laminar compressible flow. – The laminar compressible-flow relation applies to low-pressure gas and cryogenic-fluid leakage. Low-pressure and low-flowrate cryogenic fluids generally will vaporize before entering the seal interface; therefore, the flow process is best described by compressible theory. The leakage flow can become choked at the exit for pressure ratios greater than about 4:1 ($\Delta r/h > 100$). An approximate quasi-one-dimensional analytical model that considers the inertia terms has been constructed for analysis of compressible choked flow across a sealing dam (ref. 12). The leakage for subsonic flow is given by the following equation (ref. 8):

$$\dot{w} = \frac{\bar{b}h^3 (P_2^2 - P_1^2)}{24RT \mu_a L} \qquad (7)$$

Laminar liquid flow without inertia. – The theory of laminar liquid flow without inertia generally is satisfactory for predicting most low-speed or static liquid leakage. Cryogenic fluids at medium pressure ($\approx$ 200 psi) generally fall into this regime. The radial leakage flow is given by the following equation (ref. 8):

$$\dot{w} = \frac{\rho \pi h^3 (P_2 - P_1)}{6 \mu_a \ln(r_2/r_1)} \qquad (8)$$

where

ρ = density, lbm/in.^{3}

r_2 = seal face outer radius, in.

r_1 = seal face inner radius, in.

Laminar liquid flow with inertia. – The centrifugal effects caused by rotation are negligible with low-density cryogenic fluids (e.g., liquid hydrogen), but may be significant with higher density cryogenics (e.g., liquid oxygen) at low pressure and high speed. The inertia effects with more viscous fluids may be significant at low speed if the pressure is also low. The laminar radial leakage flow with consideration of the inertia effects is given by the following equation (adptd. from ref. 9):

$$\dot{w} = \frac{\rho \pi h^3}{6\,\mu_a \ln(r_2/r_1)} \left[P_{r1} - P_{r2} + \frac{3}{20}\,\rho_m\,\omega^2 (r_2^2 - r_1^2) \right] \tag{9}$$

where

P_{r1} = pressure at seal face inner radius, psia

P_{r2} = pressure at seal face outer radius, psia

ρ_m = mass density = ρ/g, lbm-sec^2/in.^{4}

g = acceleration due to gravity, in./sec^2

Turbulent liquid flow without inertia. – Most seals for high-pressure cryogenic pumps operate in the turbulent regime because of the high leakage flow, high rotational speeds, and low fluid viscosity. The turbulence may be induced by either leakage or rotation. The turbulent radial leakage flow without inertia is given by the following equation (adptd. from ref. 9):

$$\dot{w} = 26.8\,g \left(\frac{h^{12}}{\mu_a} \right)^{1/7} \left[\frac{\rho_m (P_{r2} - P_{r1})}{r_1^{-3/4} - r_2^{-3/4}} \right]^{4/7} \tag{10}$$

Turbulent liquid flow with inertia. – The rotational Reynolds number ($\bar{r}\omega h/\nu$) indicates that rotationally induced turbulence will exist on most cryogenic turbopump seals because of the low fluid viscosity. The radial leakage flow with rotational induced turbulence is given by the following equation (adptd. from ref. 9):

$$\dot{w} = 161\rho\left(\frac{h^9}{\rho_m^3 \ \mu_a \ \omega^3}\right)^{1/4} \left[\frac{P_{r1} - P_{r2} + 0.512\,\rho_m\left[\left(\frac{r_2\,\omega}{2}\right)^2 - \left(\frac{r_1\,\omega}{2}\right)^2\right]}{1.333\left(r_1^{3/4} - r_2^{3/4}\right)}\right] \qquad (11)$$

Empirical data. – Reasonable estimates of dynamic leakage of face-contact seals for cryogenics may be obtained by utilizing either the laminar-flow-with-inertia or the turbulent-flow-without-inertia theory (depending on the Reynolds number), with the assumptions of liquid conditions and a leakage path effective gap of approximately 200 microinches ($h \approx 200\ \mu$in.) This assumption allows for the normal thermal distortions that generally occur on conventional turbopump cryogenic seals; it results in calculated leakage rates that are in the approximate range of most measured values. A smaller effective gap ($h \approx 100\ \mu$in.) is possible on designs that compensate for the major thermal effects. A larger effective gap (up to $h \approx 400\ \mu$in.) may result if the thermal gradients are extreme or if the effects of thermal distortion are not adequately considered in the design. When two-phase fluid conditions exist, a leakage range can be estimated by calculating the leakage based on both liquid and gas conditions.

An empirical relation for estimating static leakage of liquid hydrogen at low pressure (60 psig) is presented in reference 13. A leakage parameter for carbon face-contact seals based on face geometry was obtained from a correlation of transition flow theory and experimental test results. The results indicate that a leakage path effective gap of approximately 25 to 50 μin. may be assumed for predicting static cryogenic leakage of solid carbon ring seals. An effective gap of 50 to 100 μin. should be assumed for seal-ring-insert designs, because of the additional thermal distortions.

Additional emprical relations for correlating gaseous helium static leakage on new and used carbon face seals with static leakage of liquid hydrogen and liquid oxygen are presented in references 14 and 15. The relations were established to allow prediction of static leakage of propellant on the J-2 engine.

2.2.6 Misalignment Tolerances

The alignment of the rotating mating surface and the stationary seal nosepiece is more critical on high-speed seals because of the larger forces required to compensate for the inertia effects and ensure adequate dynamic response or tracking of the nosepiece.

Turbopump hardware generally is machined with precision tolerances to maintain initial installation alignment.

The dynamic loading on the rotating shaft and the pump casings usually is very high in comparison with that in most other applications. Therefore, the initial alignments usually are altered by the dynamic deflections. The effect of the deflections can be minimized by locating the seal as closely as possible to the shaft bearing support.

The seal design must compensate for the misalignments if the performance requirements are to be satisfied. The effects of tolerance variations, dynamic deflections, thermal expansions and contractions, shaft transient movements, and seal face wear on axial operating length, radial location, mating surface normality, parallelism, eccentricity, wobble, and runout must be considered.

2.2.6.1 AXIAL OPERATING LENGTH

Shaft-riding seals (e.g., segmented carbon and floating ring) are used when the variation in axial operating length is greater than the travel capability of face-contact seals. The travel limit of a face seal is dependent on the axial space available and generally is established by the bellows or spring load variation or stress level. Allowable travel can be increased by using longer, lower-rate springs. Turbopump face seals usually are limited by available space to an axial travel of approximately ±0.015 in. in small sizes (1-in. diameter), ±0.050 in. in medium sizes (3- to 6-in. diameter), and ±0.100 in. in large sizes (10-in. diameter).

2.2.6.2 RADIAL LOCATION

The radial location of the seal face relative to the center of rotation is held within approximately 0.002 in./in. of diameter on high-speed face seals in order to prevent excessive wiping action on the seal face. The effect of the wiping action is controversial. It may cause leakage or additional face wear by shifting the wear track. It may improve the heat dissipation capacity by exposing a portion of the rubbing surface to the sealed fluid at each revolution. The radial location of shaft-riding seals does not affect performance, but is generally held within approximately 0.005 in./in. of diameter because of internal clearances.

2.2.6.3 MATING SURFACE NORMALITY

In high-speed ($>$ 10 000 rpm) seals, the normality of the seal mating surface to the axis of rotation generally is maintained within 0.0001 in./in. of diameter (total indicator reading – T.I.R.) to prevent excessive mating-ring wobble. Wobble of the mating ring at high speed results in excessive face loading because of the inertia of the stationary nosepiece as it attempts to track the mating surface. The capability of the seal face to track a wobbling mating ring is limited by the inertia of the nosepiece and the resultant closing force (resultant pressure force plus spring force minus friction drag due to secondary seal or vibration damper). Excessive leakage, edge chipping of the seal nose, or excessive wear may occur if the limit is exceeded. Bellows-type seals normally have better dynamic response than other secondary seals because friction drag is lower.

2.2.6.4 ROTATIONAL ECCENTRICITY

The mating surface diameter for high-speed (>10 000 rpm) shaft-riding seals usually is held concentric to the center of rotation within 0.0005 in./in. of diameter (T.I.R.) to prevent excessive rotational eccentricity. The effects of inertia forces and dynamic response required to track the eccentric motion must be considered. Segmented carbon seals tend to leak excessively when the runout exceeds the stated limit, because the tracking capability of the segments is limited. Floating-ring shaft seals are less sensitive to eccentricity, but tend to wear at larger values because of the higher inertial forces.

2.2.7 Vibration Control

Turbopump vibration. – Rocket engine turbopump seals are exposed to extreme vibration environments that can cause premature failure unless proper consideration is given to vibration in seal design. The seals on the J-2 engine are exposed to vibration in each of the two major axes at levels up to 15-g peak at a frequency range from 30 to 500 Hz and up to 30-g peak at a frequency range from 500 to 2000 Hz throughout the operational life of the seal. Several critical frequency ranges, which correspond to engine or turbopump resonant points, exist.

Seal vibration. – Face-contact seals running dry or in poor lubricants such as cryogenic fluids may excite a self-generated natural-frequency vibration as a result of the stick-slip conditions at the seal rubbing face. The difference between the static and dynamic coefficient of friction can initiate a circumferential vibration mode at the natural frequency of the seal face, which excites the axial vibration modes. There are two basic modes of axial vibration. One is vibration of the seal face composite as a free body supported by the bellows or springs with the motion restricted by the mating ring surface; this mode causes face separation, leakage, and fatigue failure. The other mode is vibration or surging of the bellows or springs between the seal face and the housing; this mode causes fatigue failure.

Mating-ring runout and shaft axial motion have been assumed to be the primary exciting force for initiating seal resonant-frequency vibration; however, experimental test results (ref. 16) and many other turbopump tests indicate that the stick-slip condition is the major cause of undamped vibration of face-contact seals. Lubrication at the seal rubbing surface eliminates the stick-slip condition. Therefore, the rubbing-material combination and type of sealed or coolant fluid have a great effect on self-excited seal vibration.

Seal vibration can be controlled by designing the seal to avoid the major resonant frequencies of the turbopump and by providing seal vibration dampers. The general practice is to design the seal for a natural frequency that is higher than any expected on the turbopump; this practice prevents resonant seal vibration excited by the turbopump or engine operation. Mechanical-friction vibration dampers are provided at the seal face outside diameter and at the bellows component outside diameter (fig. 11) to limit the amplitude of the self-excited vibration modes. Some experiments have indicated that effective damping at either location will prevent both vibration modes. The friction drag force required for effective vibration damping of turbopump-type welded-bellows seals is approximately 5 to 10 percent of the seal spring load.

The secondary-seal friction drag on conventional elastomer, piston-ring, and lip seals generally will provide effective vibration damping. Surging of the seal loading springs has not been a significant problem; therefore, spring dampers usually are not provided.

2.2.8 Contamination Allowances

Rocket engine fluid systems usually are cleaned and filtered, filters in the 10 to 100μ range being used; also, the turbopump shaft seals are protected from atmospheric dirt. Therefore, foreign-particle contamination is not a serious problem, and special protection systems (e.g., neutral fluid injection, buffer zones, centrifugal separators, slingers) ordinarily are not utilized. However, many turbopump shaft seals are located in sump areas that tend to collect foreign-particle settlings (metal filings, machine chips, weld slag, braze flux, wear debris from other components). The sensitivity of the seal to foreign-particle damage must be considered in the effort to achieve high reliability. A seal that is not capable of operating with a small amount of foreign-particle contamination would not be practical for most turbopump applications.

Rubbing-contact carbon face seals have demonstrated satisfactory reliability when exposed to the foreign particles normally found in turbopump systems. The particles generally are too large to enter between the rubbing surfaces, and little damage from scoring or gouging of the seal face occurs. However, the particles tend to congregate in the secondary seal area and cause malfunctions as a result of hangup and leakage. Plastic-lip secondary seals tend to wear and become gouged, the result being excessive leakage. Also, foreign particles tend to become lodged in the plastic, and the abrasive action scores the mating surface. The same problem exists to a lesser extent on elastomer secondary seals. Piston-ring secondary seals also are subject to abrasive damage from foreign particles. The most reliable secondary element for abrasive service is the bellows seal.

Hydrodynamic and self-energized hydrostatic seals are very sensitive to foreign-particle contamination. The particles tend to enter the face clearance gap and gouge or wear the seal face. The face damage can result in loss of the lift potential and subsequent failure. The externally pressurized hydrostatic seal (fig. 16) is resistant to damage by sealed fluid contamination, because the face is separated by the purge fluid. The cleanliness of the purge fluid is critical.

The possibility that the sealed fluid will form abrasive crystals as a result of thermal decomposition caused by heat transfer at the seal rubbing face or chemical reaction must be considered. The load-times-speed limit may be established by the critical temperature. A neutral fluid buffer may be required to prevent the sealed fluid from contacting the seal face.

The harder seal face materials (tungsten carbide, titanium carbide, aluminum oxide) are most resistant to abrasive damage.

2.2.9 Seal Mounting Requirements

2.2.9.1 SEAL PILOT

Mounting pilots for turbopump seals must compensate for extreme thermal gradients and differential thermal contraction or expansion of dissimilar materials. Excessive distortions and high stresses between the turbopump housing and the seal housing pilot may result from increased interference if provisions for the thermal differential are not allowed. The general practice is to use materials with similar thermal contraction rates for the seal housing. When dissimilar materials are used, the seal pilot fit at ambient temperature must be adjusted to allow for the thermal changes.

Applications that utilize similar materials may require special mounting methods if the temperature gradient between the turbopump housing and the seal housing is excessive. Diametral pilots may not maintain adequate seal radial alignment because of the large pilot clearance required to allow for the thermal changes. A mounting device consisting of three equally spaced radial pins that engaged radial slots in the seal housing was used on the seal on the J-2 liquid-hydrogen turbine to elimimate carbon breakage caused by excessive interference at the seal pilot. The seal was mounted on the pump housing, which was chilled down to approximately $-400°F$, and the flange was exposed to the turbine hot gas at approximately 1000°F. Therefore, the seal flange temperature increased more than that of the pump housing pilot, and thus excessive interference developed. The three-pin mount allowed thermal growth while maintaining radial alignment. Flexible pilots that allow radial deflection without stressing the seal housing may also be utilized.

2.2.9.2 FLANGE SEALS

Many flange gasket materials tend to shrink away from the mating surfaces at low temperature. The reduced seating stress may result in excessive leakage. The gasket seating stress can be maintained by spring loading the gasket or the flange to compensate for the dimensional change. The conventional elastomer O-rings (Teflon, Buna, Viton) are not satisfactory below about $-100°F$ because of excessive thermal-contraction differential and insufficient resiliency to compensate for the dimensional change.

Generally it is more satisfactory to locate the static seal on an axial surface rather than in the radial direction because of the greatly increased difficulty of sealing at low temperature. Axial seals can be clamped across a thin section to minimize the dimensional change caused by thermal contraction. Low-temperature gaskets are made as thin as possible. The most effective static seals for extreme temperatures are the metallic spring-loaded pressure-actuated types. Elastic deflection of the seal provides the necessary resiliency to compensate for dimensional variations at the sealing joint. The seal seating stress is increased at higher pressures for more effective sealing by the inherent pressure loading feature. The all-metal construction allows the seal to be used with reactive and corrosive propellants at temperatures from $-423°F$ to $1500°F$. The seals usually are plated with a softer material (copper, silver, gold, Kel-F, Teflon) to improve the compliance to irregular surfaces and reduce the required seating stress.

2.2.9.3 FLANGE LOADING

The required flange load is determined by the seating stress for gaskets and the load to deflect the elastic members for spring-loaded seals. The effect of thermal changes and pressure loading on the flange is taken into account to ensure adequate load control at the extremes of operation. The thermal changes and pressure-separating force on the flange must not reduce the flange preload below the minimum seating stress required for effective sealing. When possible, the flange is designed to cause the pressure force to increase the flange seating stress. The effect of thermal contraction can be used to increase the gasket seating stress by utilizing material combinations that result in more shrinkage in the loading members than in the seal element. Bolts that shrink more than the flange generally will compensate for the shrinkage of thin (0.005 to 0.015 in.) gaskets.

The seal flange can be effectively spring loaded by the use of several small-diameter long-length bolts for clamping. The elastic elongation caused by bolt preload generally is sufficient to compensate for gasket shrinkage. Low-temperature gaskets are loaded by elastic members to prevent loss of preload due to thermal contraction. The use of large ring nuts to clamp gasket flanges is avoided because of the low elastic elongation and the nonuniform loading caused by the nut surface not being exactly parallel to the seal surface. Several small bolts around the flange will provide more uniform loading without precision tolerances.

2.2.9.4 SHIMS

In most turbopumps, the seal axial operating length must be adjusted by using shims to position the seal or mating ring. The tolerance stackup between the stationary mounting surface and the rotating mating surface generally is too large to allow installation of the seal without shims. Most turbopump face seals are installed to a tolerance of ±0.005 to 0.010 in.

Usually the shim is combined with the flange gasket to provide the static seal; therefore, the shim must satisfy the gasket requirements previously discussed. Gasket shims often are ground metal washers of various thicknesses with narrow contact lands around the inner and outer edges to increase the seating stress. The shims generally are coated with a softer material (copper, silver, gold, Kel-F, Teflon) to provide more effective static sealing. Shims also are constructed with recessed grooves to allow the use of spring-loaded, pressure-actuated static seals (fig. 7). Bonded laminated shims usually are not satisfactory as gaskets because of the irregular seating surface and leakage between the laminations. Laminated shims have been used between asbestos rubber gaskets in liquid oxygen on the H-1 turbopumps (fig. 5); however, leakage occurred, and the later designs were changed to either ground metal washers or spring-loaded seals.

2.2.9.5 PROVISION FOR SEAL REMOVAL

Seals frequently become stuck or wedged in the pump housings as a result of distortion or yielding of the pilots. Considerable difficulty is avoided by use of special pulling tools to assist in seal removal.

2.2.9.6 LOCKING DEVICES

The bolts or nuts used to mount the seal are securely locked to prevent loosening and backing out. A loose bolt or nut in a liquid oxygen system can result in a pump explosion caused by ignition of the rubbing metal. When possible, bolts and nuts are trapped in position by the adjacent hardware.

2.2.9.7 MATING-RING MOUNTING

Excessive seal leakage due to mating-ring distortion can result from improper mating-ring mounting. Free-floating nonloaded mating rings are preferred for minimizing distortions from the clamping forces; however, this method generally is not feasible at cryogenic temperatures because of the difficulty of sealing around a free-floating ring. Therefore, most cryogenic mating rings are clamped axially along the turbopump shaft. The distortions due to the clamping loads are minimized by ensuring that the mating surfaces are flat and normal to the direction of applied load. When possible, the surfaces are lapped flat within 3 helium light bands.

The mating rings must be designed to minimize the bending moments from the clamping loads, thermal loads, and pressure forces. One method is to separate the clamping stresses from the seal mating surface by using a thin web to support the mating ring. The other approach is to make the mating ring rigid enough to withstand the clamping forces without being distorted.

The joints between the mating ring and shaft or shaft spacers, which are exposed to pressure, are sealed to prevent leakage.

The mating rings are prevented from rotating relative to the shaft and thus are prevented from damaging the static seal and pilot. The axial clamping force generally is adequate to prevent rotation, except when the differential thermal contraction or the Poisson effect of the centrifugal deflections causes the axial stackup to loosen. This problem was solved on the H-1 turbopump by utilizing radial splines between the mating ring and shaft.

2.3 SEAL COMPONENTS

The design considerations related to the detail requirements and components of the seal assembly are presented in the following discussion. The detail component design analysis is an essential part of the seal design and must be integrated with the design processes for the seal system and seal assembly.

2.3.1 Seal Materials

The selection of materials for turbopump seals is based primarily on compatibility with the fluid medium, with consideration of temperature limitations, corrosion-resistance, thermal contraction and expansion, heat conductivity, thermal stability, strength, ductility, hardness, modulus of elasticity, resiliency, fatigue, creep, wear and friction, self-lubricity, fabricability, availability, and cost.

A summary of materials currently used for rotating-shaft seals on rocket engine turbopumps is presented in table III. Additional material compatibility considerations are discussed in references 17 through 20.

2.3.1.1 COMPATIBILITY

The selection of materials for turbopump seals is limited by the requirement that the material must be compatible with the rocket engine propellants. Strong oxidizers such as

Table III. – Summary of Materials Currently Used for Turbopump Rotating-Shaft Seals

Fluid	Face combinations (nose - mate) 1	Mating-ring base	Secondary seal	Bellows	Housing and structural elements	Insert adhesive	Metal plating
Liquid Oxygen	Carbon P692-Chrome Carbon P5N-Chrome Carbon P5N-LW5 Carbon 5AG-Chrome	4130 Inconel 750 Inconel 718 440C	Kel-F Mylar	347 Inconel 750 Inconel 718	302 303 304 321 Inconel 750 Inconel 718 Inconel 600 Invar 36 Carpenter 42	Epon 901/B3 Teflon Fusion	Chromium Cadmium Silver Copper Nickel Gold
Liquid Fluorine	K162B-K162B Al_2O_3-K162B Al_2O_3-Al_2O_3	K162B Inconel 750 Inconel 718	2	Inconel 750 Inconel 718	Inconel 750 Inconel 718	2	Chromium Copper Nickel Gold
FLOX : 30% Fluorine 70% Oxygen	K162B-K162B	K162B	Kel-F	Inconel 750 Inconel 718	3	Teflon Fusion	3
Nitrogen Tetroxide	Carbon EY105-LW5	321	2	AM 350	321 347	2	
Liquid Hydrogen	Carbon P5N-Chrome Carbon 5AG-Chrome Carbon P03N-LW5	Inconel 750 Inconel 718 310	2	Inconel 750 Inconel 718	3	Epon 422 HT 424	3
50% Hydrazine 50% UDMH	Carbon EY105-LW5	321	2	AM 350	347	2	
RP-1	Carbon G39-Chrome Carbon CCA72-Chrome	4130 4340 440C	Viton A Buna N	AM 350	300 Aluminum	Epon 901/B3 Epon 422	3
Hot Gas LOX + LH_2 1000°F Max.	Carbon P5N-Chrome Carbon G84-Chrome	Inconel 750 Inconel 718	2	Inconel 750 Inconel 718 AM 350	Inconel 750 Inconel 718 Inconel 600	Epon 422 HT 424	3
Hot Gas LOX + RP-1 1000°F Max.	Carbon P2003-Chrome Carbon G84-Chrome Carbon G84-LW1	4130 4340 440C	2	Inconel 750 Inconel 718 AM 350	Hastelloy Inconel 750 Inconel 718	Epon 422 HT 424	3

1 Listed in order of preference

2 Data not available

3 Same as for liquid oxygen

liquid oxygen and liquid fluorine are capable of combustion with all available seal materials except fully oxidized oxides, fluorides, and carbides. Hypergolic reaction (i.e., ignition on contact without an external energy source) can occur with some propellant and material combinations. Combustion generally will continue until either the oxidizer or the combustible material is consumed; therefore, the result of combustion may be catastrophic failure of the engine and potentially the total vehicle.

Metals. – Many metals (Monel, nickel-base alloys, and stainless steel) are sufficiently resistant to oxidation leading to combustion that they may be used as structural components of liquid-oxidizer seals; however, rubbing contact of metal against metal in strong oxidizers is avoided because of the potential ignition hazard. The protective oxide coating may be destroyed by the rubbing contact, and the frictional heat generated at the surface asperities may be sufficiently high to initiate combustion. Instances of metal-to-metal rubbing in liquid oxidizers without ignition have occurred; however, this result is unpredictable, and the hazard is generally too great to risk. Liquid-oxidizer pumps have exploded as a result of rubbing contact of metal surfaces.

Metal-to-metal contact at seal nosepiece pilots, antirotation tangs, and friction-type vibration dampers in liquid oxidizers usually is acceptable, provided that care is taken to minimize the contact energy and oxidation-resistant materials are utilized. Most metals are resistant to detonation from impact loads in liquid oxidizers. No instances of metal reaction in liquid oxygen have been reported during the standard 70-ft-lbf impact tests, and no metal-oxygen reaction at seal antirotation tangs have been noted.

The martensitic steel alloys (e.g., 17-7 PH, AM-350, 4130, 4340) are not compatible with cryogenic propellants because of the loss of ductility at low temperature. Some of these materials (e.g., 4130 and 4340) have been used at cryogenic temperature for structural components when ductility or elongation is not critical; however, these materials are not used as flexing elements.

Metal platings (silver, gold, cadmium, chromium, nickel, copper) generally are very resistant to ignition in liquid oxidizers. Some exceptions are silver and cadmium in liquid or gaseous fluorine. Static seals and fasteners used in fluorine usually are plated with copper or gold. The metal platings used for most propellants are copper or silver for static seals, cadmium or silver for fasterners, and chromium or cadmium for corrosion protection. Gold is the most resistant to oxidation and used for critical applications. Silver is resistant to oxidation in liquid oxygen and therefore may be used in transient or intermittent rubbing contact.

Carbon. – Carbonaceous materials have demonstrated adequate resistance to combustion and ignition in rubbing contact against hard-chrome-plated steel and tungsten/chromium carbide (LW5)-coated steel mating rings in liquid oxygen. Carbonaceous materials with reactive impregnants are potentially capable of ignition in liquid oxygen; however, no combustible reactions are known to have occurred with the materials listed in table III. The failure mode for carbons exposed to high rubbing-surface temperatures (≈ 500 to 1000°F) in liquid oxygen is generally cracking and breakage caused by thermal stresses and chemical erosion. Impact tests of carbon (P5N, P692, P03N) in liquid oxygen at energy levels of 70 ft-lbf resulted in the carbon being pulverized without any reactions. Carbonaceous materials are not used in liquid fluorine because of lack of compatibility and instances of explosive reactions. Hygroscopic carbons (i.e., those that tend to absorb water) are not used in cryogenic service because of possible freezing.

Plastics. – Plastic materials (Kel-F, Teflon, Mylar) have demonstrated adequate resistance to oxidation and ignition both during rubbing contact at fluid pressures up to 1000 psi and impact (70 ft-lbf) tests in liquid oxygen. The compatibility at very high fluid pressures (5000 psi) may be marginal. The plastic materials are not compatible with stronger oxidizers such as liquid fluorine. The compatibility of Mylar is marginal in liquid oxygen at impact energy levels in excess of 70 ft-lbf. Detonations have occurred during impact tests at energy levels of 80 ft-lbf; however, no known reactions have occurred during liquid oxygen testing on lip seals or gaskets. Plastic materials generally are not used below –320° or above 600°F.

Elastomers. – Elastomeric materials (e.g., Viton A and Buna-N) are not useful with cryogenic propellants because they lack elongation and resiliency at low temperature (< 65°F). These materials are not satisfactory above approximately 500°F because of thermal degradation. Chemical compatibility is a problem with some propellants. The effects of shrinkage and swelling of elastomeric materials in some propellants must be considered. Some elastomeric materials are subject to surface deterioration caused by high ozone content in atmospheric air; Viton A is resistant to deterioration by ozone. Most elastomeric materials, except Viton A, are age-limited and require replacement after specific time periods from the cure date. Viton A generally is used for turbopump RP-1 and lubricating oil seals.

Adhesives. – The compatibility of epoxy adhesives with liquid oxygen is marginal. Impact tests have indicated that reactions will occur at impact energy levels in excess of 2 kg-m (14.46 lbf-ft). Epon 901/B3 is the most nearly compatible epoxy adhesive available and has been used successfully to bond carbon inserts to metal carriers on the J-2 liquid oxygen seals. The only other bonding method satisfactory in liquid oxygen is Teflon fusion bonding, which is compatible with liquid oxygen but is not as strong or as consistent as epoxy adhesives. Epoxy adhesives are not used in liquid oxygen unless the bonded surfaces are protected from direct contact with the circulating fluid and there is no possibility of high-energy impact loads or rubbing contact. No adhesive bonding methods compatible with liquid fluorine are known.

Ceramics and cermets. – The ceramic and cermet materials (aluminum oxide, titanium carbide, tungsten carbide, tungsten/chromium carbide) are among the most inert and oxidation-resistant materials available; therefore, they are generally used when chemical compatibility, reactive oxidizers, or high-temperature oxidizing environments are a problem.

The ceramics are used at temperatures up to approximately 2000°F in oxidizing environments.

2.3.1.2 CORROSION

Corrosion protection usually is provided by utilizing corrosion-resistant materials (stainless steels and nickel-base alloys) for seal construction. Chromium and cadmium plating are used to protect ferrous metals (e.g., 4130 and 4340) in mildly corrosive environments such as liquid oxygen, RP-1, lube oil, and (LOX + RP-1) hot gas; however, plating will not provide adequate protection when moisture is present in hydrogen or fluorine environments, particularly at high temperatures. The base metal must be corrosion resistant to prevent corrosion underneath the plating due to plating porosity. The hot gas generated by the combustion of liquid oxygen and liquid hydrogen (hydrogen-rich steam: $H_2 + H_2 0$) will corrode stainless steel through hard-chrome plate.

The electrolytic action caused by the combination of free hydrogen, moisture, and heat results in excessive corrosive pitting of the 300- and 400-series stainless steels if foreign-particle contaminants are present on the metal surface. This problem was solved on the seals on the J-2 engine by utilizing Inconel alloys (600, X-750, and 718).

The combustion of hydrogen and fluorine or the presence of moisture in fluorine will produce hydrofluoric acid (HF), which is very corrosive to aluminum alloys, ferrous metals, and 400-series stainless steels. The 300-series stainless steels are satisfactory for dry HF but not for wet HF. The nickel-base alloys, copper, chromium, and carbon P5N appear to be satisfactory for both wet and dry HF.

2.3.1.3 HYDROGEN EMBRITTLEMENT

Structural elements in hydrogen environments may experience brittle-fracture failures under load as a result of hydrogen embrittlement of the metal. The embrittlement may be caused by reaction of hydrogen with the metal constituents or by absorption of hydrogen by the metal. The nickel-base and Monel alloys appear to be affected by hydrogen embrittlement in hydrogen environments at temperatures above approximately –200°F. The nickel-base alloys (Inconel X-750, Inconel 718) are satisfactory for use in hydrogen environments at temperatures below approximately –200°F, provided the proper heat treatment is used. Hydrogen embrittlement has not been a problem with the Inconel X-750 welded-metal bellows seals used on the J-2 liquid hydrogen turbopumps. Aluminum and copper alloys are not affected. The stable stainless steels (those that remain austenitic and do not transform to martensite during deformation: 310, 316, 347, A286) are resistant to hydrogen embrittlement and generally are used in hydrogen environments above –200°F. Additional discussion of hydrogen embrittlement may be found in references 21 and 22.

2.3.1.4 MATERIAL PROPERTIES AND TEMPERATURE EFFECTS

The basic material properties of strength, ductility, elongation, modulus of elasticity, resiliency, creep, hardness, thermal contraction and expansion, heat conductivity, and fatigue life are considered for the materials used in turbopump seals. The effects of operating temperatures (-423° to 1000°F) on these properties must be taken into account. In most cases, the variation of material properties with temperature is not a linear relationship, and extrapolation of data measured through narrow temperature ranges can result in gross errors. For some typical seal materials, the variation with temperature is given for tensile yield strength in figure 33, for elongation in figure 34, and for thermal expansion in figure 35. Additional material properties at cryogenic temperatures are given in reference 23. The properties at elevated temperatures are given in reference 24.

2.3.1.5 WEAR AND FRICTION

Most current turbopump seals are rubbing-contact types that operate in the boundary lubrication regime or with self-lubrication from a film deposited on the mating surface. The wear and friction properties of the seal-face and mating-surface material combinations in the sealed fluid are highly dependent on the lubrication available at the seal interface. Most friction data have been obtained with button-type specimens; however, this test procedure does not account for the hydrodynamic lift and heat buildup that normally occur at seal interfaces. The hydrodynamic life tends to reduce the resultant face-contact load and therefore the coefficient of friction. The effect of higher rubbing speed and contact load is increased interface temperature, which generally increases the coefficient of friction. The chemical reaction between the seal interface materials and the sealed fluid has a significant effect on the formation of a self-lubricating film at the seal rubbing face.

The ceramic and fused-cermet materials depend on the formation of an oxide or fluoride film for lubrication in liquid oxygen or liquid fluorine. Titanium carbide (Kentanium K162B) and aluminum oxide have demonstrated satisfactory wear and friction in liquid fluorine because a self-lubricating fluoride film forms when these materials are exposed to fluorine. The application of fused fluoride coatings to the mating surface of ceramic and cermet materials is beneficial in reducing wear and friction in liquid oxygen, liquid fluorine, liquid sodium, hydrogen, vacuum, and air (refs. 25 and 26).

Composite materials (AmCerMet 701-65) consisting of porous sintered nickel-chrome alloy matrix infiltrated with inorganic fluorides (62% BaF_2, 38% CaF_2) have demonstrated satisfactory wear and friction when tested in dry gaseous helium and exposed to fluorine. Carbonaceous materials used for turbopump seals (table III) are impregnated with inorganic resins, metal fluorides, silver, and various other additives to improve the wear and friction characteristics and assist in the formation of a self-lubricating film on the mating surface. The mechanism of transfer-film formation with carbonaceous materials and the effect on

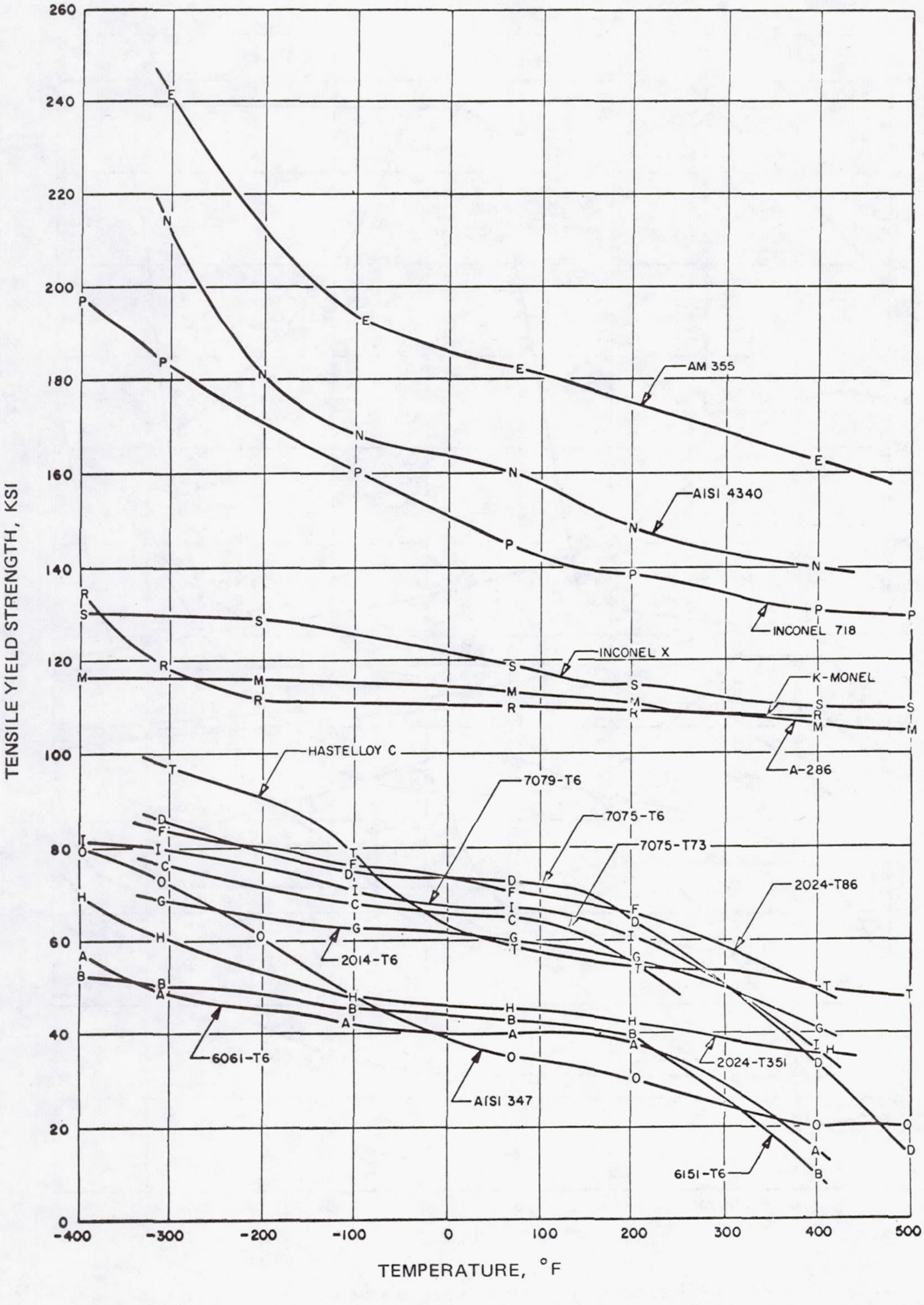


Figure 33. — Tensile yield strength of typical alloys as a function of temperature.

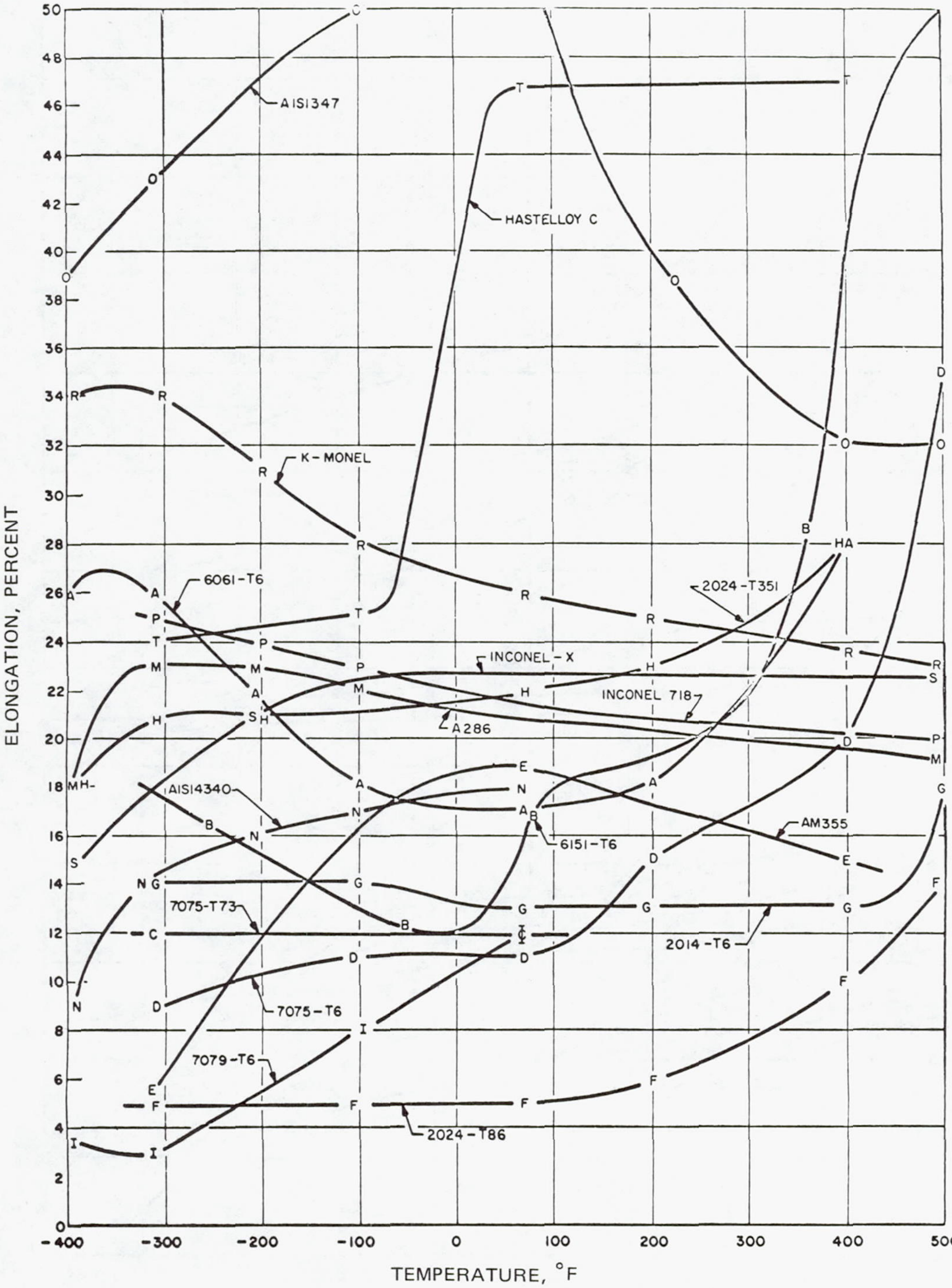


Figure 34. — Elongation of typical alloys as a function of temperature.

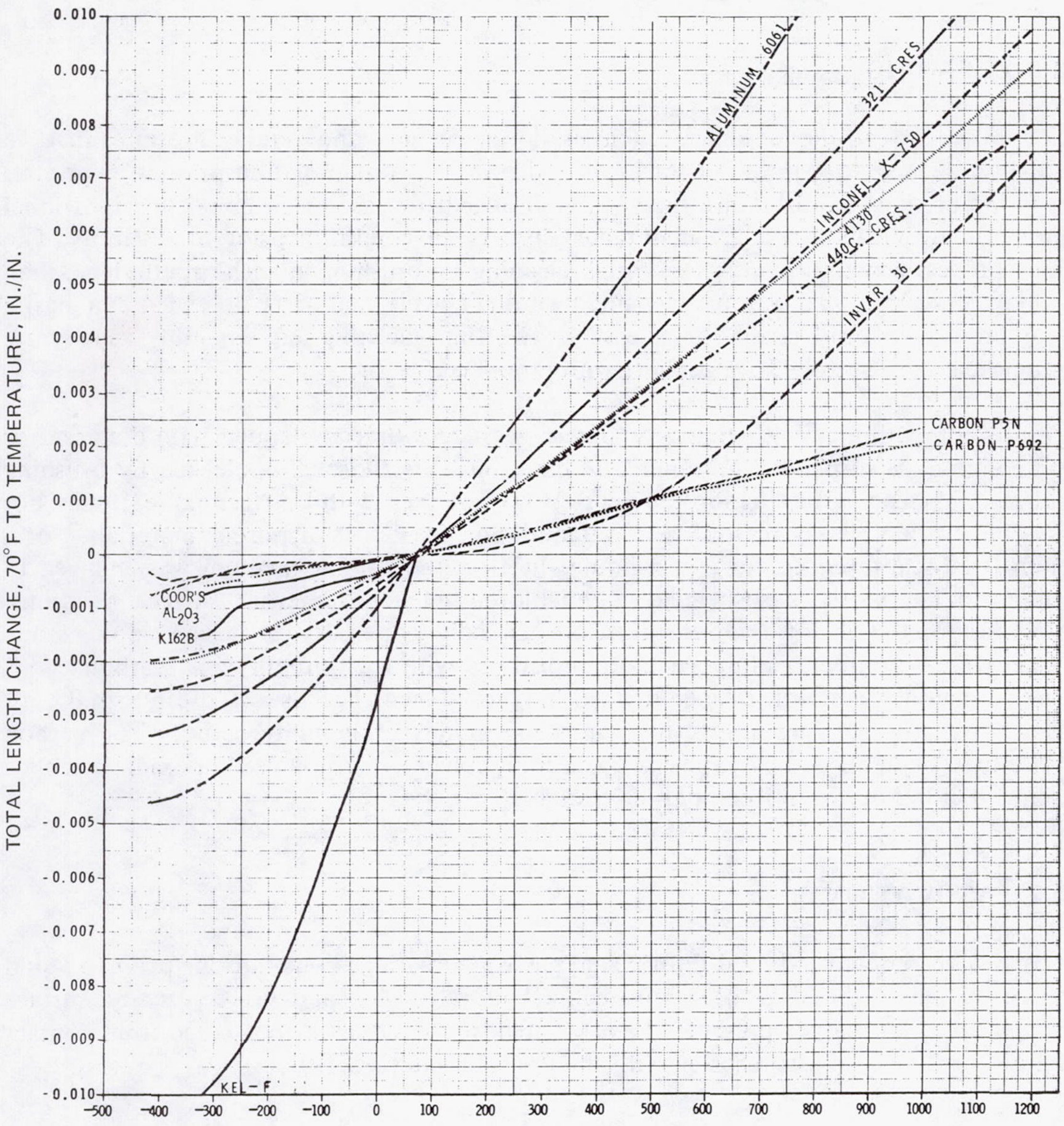


Figure 35. — Thermal expansion of typical seal materials as a function of temperature.

wear and friction are discussed in reference 27. The factors that affect wear and friction of carbon materials are discussed in reference 28. The wear and friction of carbon materials in liquid nitrogen and hydrogen are discussed in reference 29.

2.3.1.6 CLEANING

Parts exposed to strong oxidizers such as liquid oxygen and liquid fluorine must be thoroughly cleaned to remove all traces of hydrocarbon contamination prior to exposure. Seals to be used in oxidizers are specifically designed for ease of cleaning. The individual components are cleaned and are maintained in a clean condition prior to assembly. The cleaning process generally consists of rough cleaning by brushing in trichloroethylene, then hot vapor degreasing or ultrasonic cleaning or both. The cleaned parts are sealed in a plastic bag for protection until ready for assembly. The final cleaning and assembly usually are accomplished in a controlled-atmosphere white room (ref. 19).

To ensure compatibility with liquid oxidizers, seal faces that are lapped with diamond or silicon compounds suspended in mineral-oil bases must be thoroughly cleaned by brushing and ultrasonic cleaning in trichloroethylene. However, even with this technique, it may not be possible to completely remove the contamination from soft or porous materials. For a Bearium B-10 seal intended for use in liquid fluorine, the lapping medium was changed to aluminum oxide particles suspended in trichloroethylene to prevent surface reactions. Carbonaceous materials for oxidizer service are generally lapped dry on a clean lapping stone and then wiped clean with a lint-free cloth moistened with trichloroethylene. Carbonaceous materials generally are not cleaned by flushing in solvent because of the possibility of reaction with the bonding agents used in some grades of graphite carbon. The carbon wear-life may be decreased by exposure to cleaning solvents. Carbon used in oxidizer service is machined either dry or in a compatible coolant.

2.3.1.7 PASSIVATION

Materials to be exposed to liquid fluorine are passivated with gaseous fluorine prior to liquid exposure to allow buildup of a protective fluoride film and to react any surface contamination that may be present. The passivated surfaces must be protected from contact with moisture to prevent its reaction with the fluoride film (ref. 19).

2.3.2 Face-Contact-Seal Rubbing Elements

Four configurations for the rubbing element of a face-contact seal are shown in figure 36: a solid seal ring (fig. 36(a)), a seal ring insert (fig. 36(b)), a lapped-joint seal ring (fig. 36(c)), or spray-coated seal ring (fig. 36(d)). The seal-ring-insert configuration generally is used for

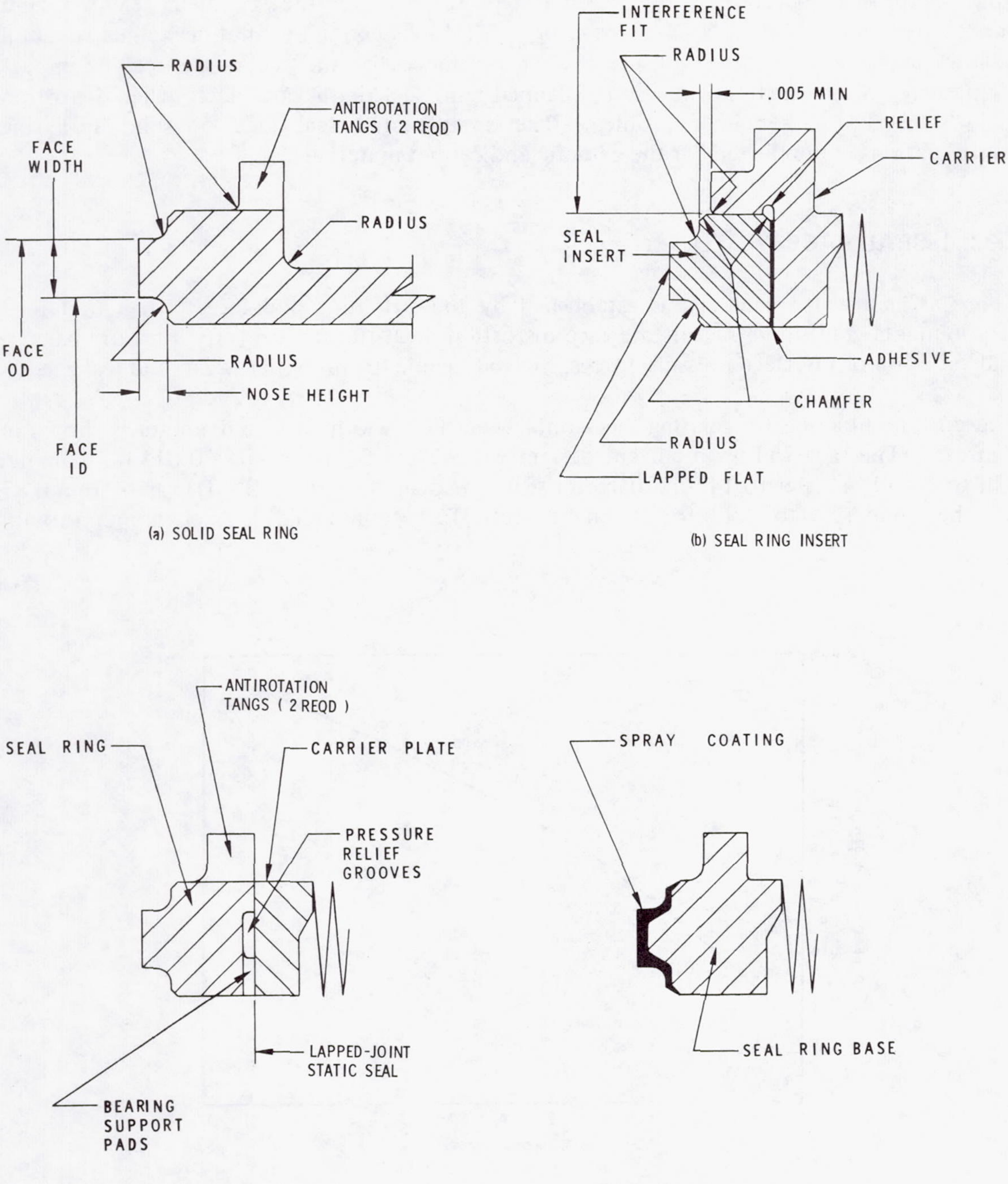


Figure 36. – Four types of rubbing elements for face-contact seals.

turbopump seals because of the material requirements for the secondary element and antirotation tangs. Seal materials satisfactory for rubbing contact cannot be welded to metal bellows and may not have sufficient wear resistance for the secondary seal or impact strength for the antirotation tangs. The lapped-joint seal ring is considered when distortion of seal-insert retainer is a problem. The spray-coated seal face or solid seal ring configurations are preferred for the ceramic and cermet materials.

2.3.2.1 SEAL FACE WIDTH

The seal face width generally is established by the structural and face-contact unit load requirements; additional factors are face distortion, heat-transfer capacity, pressure balance and variation of interface pressure forces, hydrodynamic lift potential, wear, and leakage.

The current practice for relating face-contact-seal face width to face diameter is shown in figure 37. The face width on current designs varies from 0.040 in. on a 0.615-in. diameter seal to 0.160 in. on a 10.142-in.-diameter seal. The diameter ratio (ID/OD) varies from 0.87 for the small seal to 0.97 for the large seal. Most commercial seals that operate with

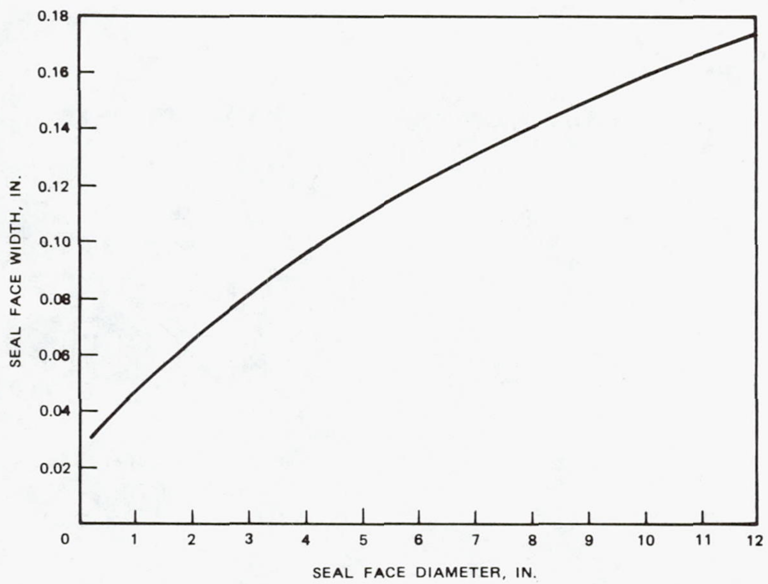


Figure 37. — Relation of face-contact-seal face width and diameter.

hydrodynamic lubrication and negligible face distortion have wider seal faces (ID/OD $\approx$ 0.75 to 0.85) to reduce the contact unit load and to increase the hydrodynamic lift so that face wear is reduced.

The seal face width on high-speed cryogenic seals usually is a compromise among minimum contact unit load, heat-transfer capacity, pressure-force balance, and the effects of face distortion. Wider seal faces decrease the unit load but result in face distortion and greater variation in interface pressure load. The variation of the average interface pressure profile due to face distortions and fluid vaporization causes a larger force variation because of the larger face area. Therefore, the closing force must be increased to prevent face separation at the maximum interface pressure; this increase may result in face overload at minimum interface pressures. The effects of face distortion are amplified on wider faces by the larger contact surface and greater temperature gradient across the face. The optimum face width for pressure-force balance and minimum distortion effects would be a knife edge.

Increasing the face width decreases the heat generation per unit contact area and allows more heat dissipation by conduction into the seal nose ring and rotating mating ring; however, the heat dissipated by convection to the circulating sealed fluid decreases as the face width is increased (refs. 6 and 7). Therefore, the optimum face width for heat transfer is dependent on the relative cooling capacity of the heat flow paths. Since seals on cryogenic turbopumps depend on the cooling capacity of the circulating fluids, the face widths generally are narrower than those in most other applications.

The face-contact unit load can be decreased without increasing the seal face width by adding pressure-vented support pads (fig. 18(b)) around the seal face to assist in load support. The support pads reduce the seal nose structural requirements and allow a narrow (0.030 in.) seal face to be utilized for more precise balancing of the pressure forces. This concept is not widely used for insert-type cryogenic seals because of the possibility of face separation and excessive leakage caused by thermal distortions. The distortions may cause the seal face to twist and be supported entirely by the pads. The concept has been used successfully in gas-turbine engines and has a potential capability for turbine seals, provided that compensation is made for the thermal distortions (e.g., centroid alignment, lapped joint, or spray-coated or solid seal rings).

2.3.2.2 SEAL NOSE HEIGHT

The seal nose height (fig. 36(a)) is established by the structural requirements and the allowance necessary for face wear. The stress level at the junction of the seal nose and the base of the seal ring can become excessive as nose height is increased. Thermal stresses develop as a result of the temperature gradient from the seal face to the base of the seal ring. Bending and shear stresses result from the pressure deflection of the seal nose cylindrical section and the deflections caused by interference between the seal insert and carrier.

Seal-nose cross sections with a height-to-width ratio of 0.3 to 0.8 usually are employed on turbopump seals to minimize face distortions and nose stresses.

It is essential that the junctions of the nose OD and ID to the seal ring base (fig. 36(a)) be provided with minimum fillet radius of 0.020 to 0.030 in. Without the fillet, nose breakage may occur because of the stress-concentration factor at the sharp corner. It was necessary to add the nose fillet on the H-1 and J-2 liquid-oxygen seals to eliminate failures. The edges of the seal face are radiused 0.005 to 0.010 in. to eliminate edge chipping caused by corner loading. All sharp corners on the seal ring are either chamfered or radiused.

The seal nose height used on current turbopump seals to provide for face wear ranges from 0.030 to 0.060 in. Turbopump seals are considered to be worn out when the nose is worn within 0.010 in. of the seal ring base.

The seal ring base on seal inserts must be a minimum of 0.005 in. above the insert carrier (fig. 36(b)) to prevent the carrier from rubbing the mating ring in the event of nose wearout or breakage. This design feature is particularly important on liquid-oxidizer seals because of the combustion hazard created by rubbing metals.

2.3.2.3 SEAL RING ANTIROTATION DEVICE

Because of balance requirements and centrifugal forces, high-speed ($>$10 000 rpm) seals generally incorporate stationary seal rings and rotating mating rings. The stationary seal ring is prevented from rotating by antirotation tangs or drive lugs between the seal ring and housing. Metal-bellows-type seals do not require antirotation devices, because of the torsional rigidity of the bellows element. Antirotation tangs have been used as vibration dampers on some bellows seals; however, the tang-to-slot clearances are very critical for the tangs to be effective as a vibration damper without hanging up. Radial splines have also been used for a combination antirotation device, seal pilot, and vibration damper.

Various arrangements of tangs, blocks, and pins that engage slots have been used on turbopump seals. Usually two tangs or blocks are located 180° apart with a sufficiently large contact surface area to prevent indentations in the slots from the impact loads. The tang-to-slot clearance is minimized to reduce impact loads. Pins usually will wear an indentation into the slots and thereby may restrict the axial movement. Also, high-impact loads tend to loosen press-fit pins and may cause failure if the pins are not restrained. When possible, the tang is made of material softer than the slot material to minimize axial hangup from slot indentations. To prevent tang breakage on brittle materials such as carbon or aluminum oxide, the antirotation slots are located in the seal ring. Radiused fillets are provided at all corners. Materials resistant to fretting and galling are utilized for the antirotation device.

2.3.2.4 SEAL RING PILOT

Except for metal-bellows seals, which are located by the bellows element, seal rings are piloted by the stationary housing to maintain radial alignment. The pilots usually are located at the secondary seal to minimize the relative radial motion required for the seal. The pilot length-to-diameter ratio must be small ($\approx$ 0.05 to 0.1) to allow angular misalignment of the seal face without interference at the pilot. Pilot diametral clearances of approximately 0.003 in./in. of diameter at operating conditions generally are adequate for radial alignment.

The pilot clearance at ambient conditions is adjusted to compensate for the dimensional changes caused by thermal contraction and pressure deflection. Close-fitting radial splines that allow differential thermal contraction while maintaining radial alignment also have been used to compensate for the dimensional changes at the seal ring pilot. The diametral clearance of a 6-in. carbon seal ring in a stainless-steel housing decreases by 0.015 in. at $-320°F$ as a result of differential thermal contraction. The pilot clearance may be decreased further if the seal housing is installed in an aluminum pump casing that is strong enough to deflect the seal housing at the higher thermal contraction rate of aluminum. The same considerations apply to other radial clearances.

Turbopump seal pilots are subjected to high impact and vibration loads that can cause fretting damage. Hard-chrome plating has been effectively utilized to eliminate this kind of damage.

2.3.2.5 SEAL INSERT RETENTION

Seal ring inserts generally are retained in the carrier with an interference fit and adhesive bond. The amount of interference depends on the required operating temperature, the relative coefficients of contraction and expansion, modulus of elasticity, and allowable stress level. Seal insert materials usually are strong in compression and weak in tension; therefore, the inserts are maintained in compressive hoop stress with the interference at the OD of the insert. Carbonaceous materials generally have a low modulus of elasticity (1.5 to 3.0×10^6 psi) and therefore may be installed with relatively large interference fits (0.003 to 0.006 in./in. of seal diameter). The ceramic and cermet materials are limited to lower interferences because of the high modulus of elasticity (50 to 60×10^6 psi).

The seal insert materials contract and expand less than most steel alloys; therefore, the effect of the maximum operating temperature range (including the temperature increase generated by friction) on the interference fit and stress level is considered. Special steels with low contraction and expansion rates (Invar 36, Carpenter 42, molybdenum steel) may be utilized to minimize the change of insert interference with temperature. Heat-transfer calculations or estimates of operating temperatures are made to ensure that adequate interference for insert retention remains at operating conditions. A minimum interference of approximately 0.001 in./in. of diameter usually is adequate for carbon inserts.

Seal inserts are bonded to the insert carrier to obtain additional retention and to provide a positive seal between the insert and carrier. Leakage through the insert interference fit has been a significant problem on cryogenic seals that are not bonded or have defective bonds. A liquid-nitrogen leak test to verify the bond sealing quality was incorporated into acceptance specifications for the liquid-oxygen seals on the H-1 and F-1 engines.

The adhesive bond is applied to the bottom of the insert and the carrier counterbore (fig. 36(b)) in accordance with the manufacturer's process specification. The insert is installed by pressing it into the carrier at room temperature. Heating the carrier or chilling the insert to decrease the interference for ease of installation is not satisfactory because of the adverse effects on the bond quality. The edge of the carrier counterbore is radiused and the corner of the insert is chamfered to allow installation without damage.

The bond strength of epoxy adhesives drops sharply at high temperature (e.g., Epon 422 decreases from 2800 psi at 70°F to 500 psi at 800°F). Adhesive bonding is not used as the only retention method on seals where the insert temperature may exceed approximately 300°F. Many adhesives are brittle at low temperature and therefore are not satisfactory in cryogenic fluids. The adhesives generally used are listed in table III.

2.3.2.6 INSERT/CARRIER SEPARATION

Insert-type seals that are exposed to pressure on the inside diameter may fail as a result of the pressure separating force between the insert and carrier. Either insert breakage caused by complete separation or seal face distortion caused by partial separation may occur. The pressure separating force is minimized by making the insert OD the same as the seal nose OD (fig. 38). The interference fit must be sufficient to ensure that the retention force is greater than the separating force at operating conditions.

Epoxy adhesives are used to minimize the possibility of insert/carrier separation by excluding the high-pressure fluid from the joint; however, the porosity of the adhesive may allow seepage of the high-pressure fluid into the joint after prolonged exposure periods.

Mechanical locks are used for insert retention when it is not feasible to provide sufficient retention force with an interference fit. The mechanical locks are spring loaded against the insert with a load greater than the separating force in order to prevent slight movements that can cause distortion of the seal face.

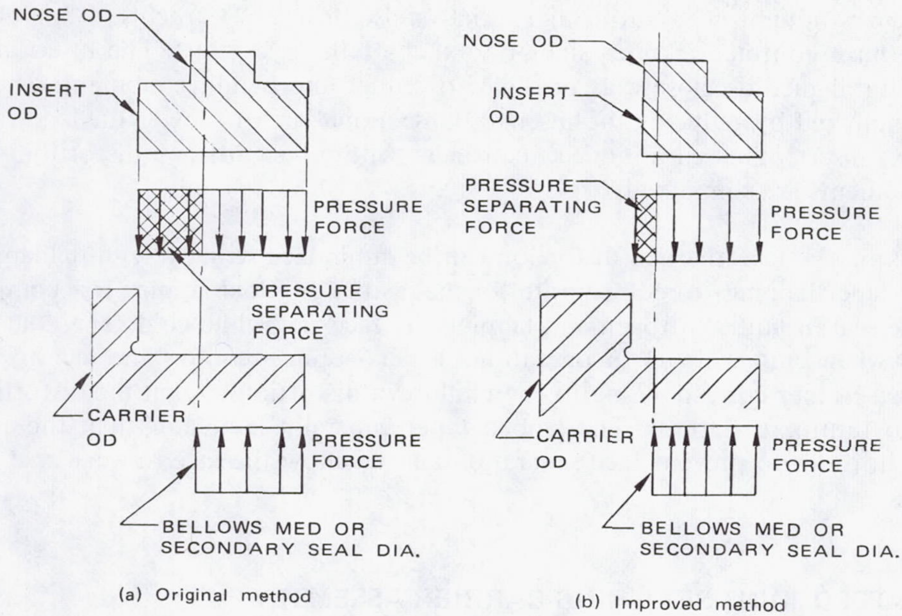


Figure 38. – Method for reducing pressure separating force between seal insert and carrier for seals with higher pressure at the inside diameter.

2.3.2.7 INSERT DISTORTION

Seal inserts with a thermal-contraction rate lower than that of the carrier material generally are distorted when chilled to cryogenic temperature because of the bending moment created by the increased interference and misalignment of the insert and carrier centroids (fig. 39).

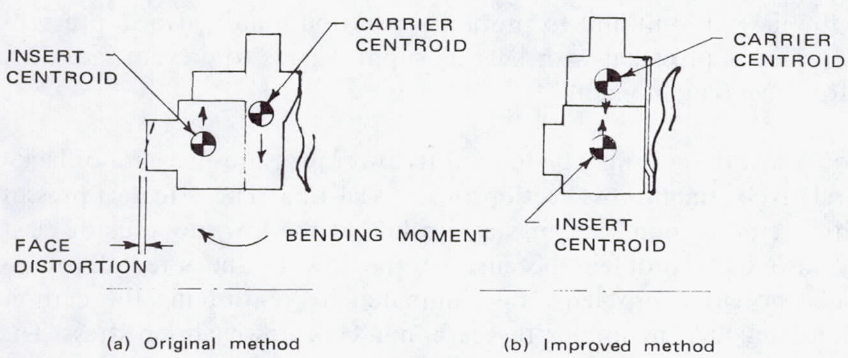


Figure 39. – Method for reducing seal insert distortion due to thermal contraction and centroid misalignment.

The problem is minimized by utilizing special steels (Invar 36, Carpenter 42, molybdenum steel) that have contraction rates similar to that of the seal insert. The insert and carrier centroids are aligned as closely as possible to reduce the bending moment. The bending force is minimized by reducing the insert cross-sectional area to lower the load required to deflect the insert. The carrier piece contracts without significant distortion when the bending moments are eliminated.

The effects of seal-face thermal distortion can be minimized with preferential lapping (cold lapping or taper lapping) to compensate for the distortion. Cold lapping is accomplished by chilling the seal in liquid nitrogen and lapping the face flat while cold. The cold flatness is repeatable when chilled down in operation. Taper lapping requires the face to be lapped with a reverse taper corresponding to the chilldown distortion so that the distortion causes the face to return to flatness. The lapped taper generally is greater than the anticipated distortion in order to prevent face separation and to allow the face to wear in at the actual conditions.

2.3.2.8 LAPPED-JOINT SEAL-RING CARRIER ASSEMBLY

Seal-ring distortion can be reduced significantly by utilizing a lapped-joint seal ring (fig. 36(c)) that eliminates the thermally induced loads caused by carrier contraction. The lapped joint allows relative movement between the seal ring and carrier plate to compensate for the differential thermal contraction. The joint is loaded axially by the sealed pressure to effect a static seal. The seal ring is restrained from rotation by a device that locks it to the housing or the carrier plate. On bellows seals, antirotation tangs to the housing are preferred, because they provide vibration damping.

The pressure closing force to load the carrier plate against the seal ring is provided by relieving either the seal ring or the carrier plate lapped surface to allow the pressure drop to occur on a smaller effective area than the secondary seal or bellows effective area (fig. 40). The relief diameter is established to provide sufficient closing force to effect a satisfactory static seal at the lapped joint and to prevent separation under adverse pressure conditions. The relieved surface is provided with bearing support pads to prevent face distortions from the pressure-force bending moments.

The lapped-joint seal designs may be limited to lower pressures because of lack of structural support and relatively small cross section of the seal ring. The effect of pressure deflection may be significant on carbon seal rings as a result of the low modulus of elasticity. Inside pressure may also be a problem because of the low tensile strength of most seal ring materials. These potential problems are eliminated by reinforcing the carbon ring with a metal retention band that maintains the carbon in compressive hoop stress. The metal band will cause thermal distortion of the carbon unless the bending moments are eliminated by centroid alignment.

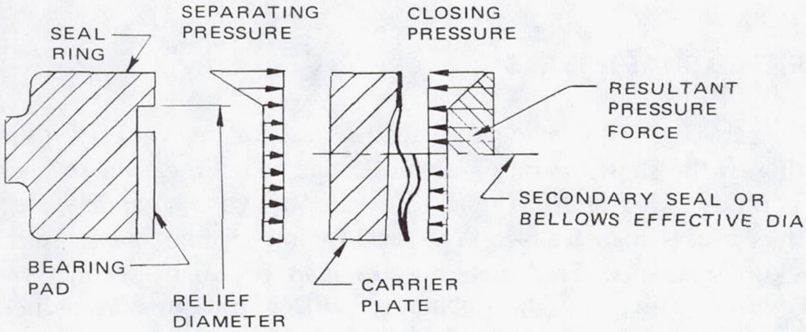


Figure 40. – Pressure forces on a lapped-joint seal ring.

2.3.2.9 SPRAY-COATED SEAL RINGS

Spray-coated seal rings (fig. 36(d)), instead of the insert designs, generally are used for the ceramic materials because of the retention, distortion, and leakage problems with ceramic inserts. The ceramic materials cannot be attached directly to a metal-bellows secondary element, and satisfactory secondary-seal materials that allow the use of a solid seal ring may not be available. Seal-face thermal distortion generally is not a problem with the spray-coated seal rings because of the low resistance to deflection offered by the relatively thin cross section of material. The thermal distortion is minimized by applying the coating to both sides of the seal ring to equalize the thermal loads.

The ceramic coating is applied by either the flame-plating process or the plasma-flame-spray process. The plasma-spray process usually is employed for aluminum-oxide coating because it has better resistance to thermal shock. Since the ceramic materials have contraction rates lower than those of most metals, thermal stresses are developed at the coating bond. The stresses may be sufficiently high to cause bond failure or breakage of the coating. The thermal stress is minimized by using a thin coating (0.010 to 0.020 in.) that reduces the forces and the temperature gradient. Aluminum oxide generally is sprayed on a transition layer of Nichrome to reduce the thermal stresses and improve the bond strength. Pure Nichrome is sprayed on the base metal, followed by a mixture of 50-percent Nichrome and 50-percent aluminum oxide, and finally pure aluminum oxide. More gradual gradations may be required for severe applications.

The seal ring base must provide a foundation to support the spray coating and allow sufficient surface for bonding. A raised tapered structure similar to that shown in figure 36(d) is preferred. All sharp corners and edges must be radiused or chamfered.

2.3.3 Circumferential-Seal Rubbing Elements

2.3.3.1 SEGMENTED SHAFT SEALS

Segmented shaft seals (fig. 20) provide an effective seal for low-viscosity fluids (gas, LOX, LH_2) by adjusting to the shaft operating diameter and radial location to maintain rubbing contact. Viscous fluids (RP-1 and oil) tend to develop a hydrodynamic wedge that lifts the segments and thus causes high leakage. The sealing-ring segment gaps are sealed off with overlapping backup segments. The segments are loaded radially against the shaft by a circumferential garter spring and the unbalanced differential-pressure-induced force. The segments also are loaded axially against the stationary housing by a wave spring and the unbalanced differential-pressure-induced force.

The forces induced by differential pressure may be partially balanced by relieving the contact surfaces (fig. 41). The relieved surfaces are provided with bearing pads for load support. It is not practical to balance the pressure forces completely because of the seal construction; therefore, dry-running segmented seals are limited to lower pressures ($\approx$ 100 psid) than are balanced face-contact seals.

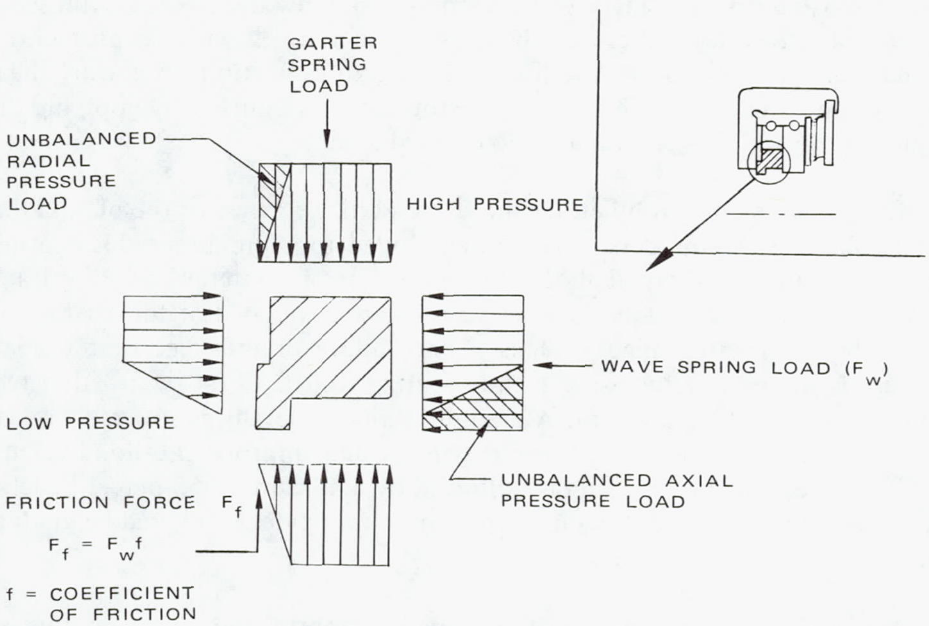


Figure 41. — Pressure forces on a segmented shaft-seal ring (balanced design).

The garter-spring load is made higher than the friction force caused by wave spring in order to relocate the segments and maintain shaft contact at low pressures. At pressures exceeding 50 psid, the unbalanced differential-pressure load is sufficient to maintain the segment loading. The unbalanced differential-pressure load is approximately 0.2 to 0.3 lbf/in. of circumference for each 10-psid pressure increment (assuming a seal nose width of 0.05 in. and linear pressure profile). The resultant shaft contact load must be consistent with the load-speed-life relations for specific materials and fluids.

The segments are prevented from rotating relative to the housing by antirotation pins located at the segment gaps. Severe conditions may require the use of pin blocks to distribute the antirotation load over a larger contact area to prevent breakage of the segments.

The design of segmented seals for extreme temperatures (cryogenic fluids or hot gas) considers the effect of the differential thermal contraction or expansion of the segment material on the operating diameter and internal clearances. The design diameter of the segments is made equal to the shaft diameter at operating conditions in order to minimize the required wear-in and prevent breakage.

Additional discussion of the theory and practice of circumferential segmented shaft seals is given in reference 30.

2.3.4 Hydrostatic/Hydrodynamic Face Seal Elements

The hydrostatic and hydrodynamic mechanisms for controlled fluid-film support of the seal face potentially are capable of extending the pressure-speed-life limitations of rubbing-contact seals and therefore are considered for high-pressure, high-speed, long-life applications. The feasibility of the hydrostatic/hydrodynamic concepts has been demonstrated in several programs on development of jet engine compressor and mainshaft seals (refs. 31 through 36) and in other applications (refs. 37, 38, and 39); however, successful operation in a rocket engine has not been demonstrated.

Because the hydrostatic/hydrodynamic seals are sensitive to the adverse conditions that exist in turbopumps, anticipated development problems are related to rubbing material compatibility in liquid oxidizers, wear of critical face geometry by transient rubbing contact, variations of the seal interface pressure profile caused by face distortion and vaporization of the cryogenic fluid, dynamic instability, low hydrodynamic-lift potential, and high leakage rates. The disastrous failure mode resulting from face-geometry damage or loss of lift potential must be considered for reliability evaluations and fail-safe requirements.

The theoretical analysis of the hydrostatic/hydrodynamic principle is based on fundamental lubrication and fluid-flow theory that has been modified to satisfy the conditions of a face seal. The theory assumes steady-state ideal fluid conditions, which seldom exist on actual turbopump seals. It is expected that empirical relations based on experimental test results and fundamental theory, similar to those developed for hydrostatic and hydrodynamic bearings, will be required for successful turbopump seal design. These relations are not currently available for rocket engine propellants. Theoretical methods and design considerations currently available for hydrostatic/hydrodynamic seal elements are given in references 31 through 53.

2.3.4.1 SELF-ENERGIZED HYDROSTATIC SEALS

The self-energized hydrostatic seals (fig. 15) utilize the sealed pressure differential to maintain controlled face separation. The lift force induced by the hydrostatic pressure depends on a minimum pressure differential that may not exist until after rotation starts; therefore, transient rubbing contact usually occurs. The seal face materials must be selected for rubbing compatibility in liquid oxidizers and sufficient wear resistance to prevent damage to the critical face geometry.

The self-compensating hydrostatic-pressure-induced lift force requires leakage flow across the seal interface to create a pressure profile proportional to the clearance gap. The hydrostatic face seals generally operate with a larger effective interface clearance than a rubbing-contact seal; therefore, the leakage rate is significantly higher.

The recessed pads with orifice compensation (fig. 15(a)) develop self-compensating pressure-induced lift forces that are dependent on the relative flow between the outer seal face and the pad orifices. The pressure-induced lift force increases for small face clearances and decreases for larger clearances so that force balance is maintained at the design clearance. A minimum of three separate pads around the seal face is utilized to provide face alignment stability. The volume of the recessed pads is minimized to prevent dynamic instabilities caused by slow response to pressure changes (low fluid-film stiffness). The dynamic stability is improved by eliminating the recessed pads; however, the pressure-induced lift forces become more difficult to predict as a result of indefinite pressure boundaries and flow effects. The orifice size must be large enough to prevent clogging by contaminants in the sealed fluid.

The recessed-step (fig. 15(b)) and convergent-tapered-face (fig. 15(c)) designs develop maximum pressure-induced lift force as the face clearance decreases, because the pressure drop across the seal face is higher than the drop across the recessed step or convergent surface. At larger face clearances, the effect of the recessed step or convergent surface becomes negligible, and the pressure drop occurs across the entire surface; this condition reduces the pressure-induced lift force and maintains force balance at the design clearance.

The depth of the recessed step or taper must be very small ($\approx$ 0.0001 to 0.0005 in.), the dimension depending on the fluid-film design thickness required to obtain adequate fluid-film stiffness for dynamic stability. The fluid-film stiffness depends primarily on seal face area, pressure differential, and fluid-film thickness. There is an optimum recess depth for a given design fluid-film thickness and pressure differential.

2.3.4.2 EXTERNALLY PRESSURIZED HYDROSTATIC SEALS

The externally pressurized hydrostatic seals (fig. 16) maintain controlled face separation by fluid-film support from an external pressure source. The pressure-induced lift force increases for small face clearances and decreases for larger clearances to maintain force balance at the design clearance. The external pressure source allows the seal face to be lifted prior to start of rotation to eliminate transient rubbing contact. Since the pressure-induced lift force is independent of the sealed pressure, sealed-fluid viscosity, and rotational speed, this concept may be utilized when the other types of hydrostatic/hydrodynamic seals are not feasible. However, the additional complexity of the pressurizing system, availability of the pressurizing fluid, and dilution of the propellant with the pressurizing fluid may pose problems. Dynamic instability of the seal ring also has been a significant problem. The stability is improved by increasing the fluid-film stiffness by the methods described for the self-energized hydrostatic seal.

2.3.4.3 HYDRODYNAMIC SEALS

The hydrodynamic seal (fig. 17) maintains controlled face separation by means of lift forces induced by the hydrodynamic pressure developed at the seal interface by the rotational speed. The hydrodynamic lift is independent of the sealed pressure and is proportional to the rotational speed. Therefore, the hydrodynamic seal may be utilized at pressures below the minimum required for hydrostatic action; however, a minimum speed is required to develop sufficient pressure-induced lift forces for face separation, and rubbing contact generally occurs during the start and stop transients. The potential problems of rubbing-material compatibility in liquid oxidizers and wear of the critical face geometry exist. The available hydrodynamic lift is marginal with cryogenic fluids because of the low viscosity and vaporization of the fluid. Two-phase (liquid and vapor) fluids may disrupt the force balance by their effects on the interface pressure profile. The hydrodynamic lift force and fluid-film stiffness are maximum at minimum face clearances, and the rate of increase is high; therefore, hydrodynamic seals generally are more stable than the hydrostatic type at low clearances and low pressures.

The shrouded-Rayleigh-step concept (fig. 17(a)) is pressure balanced like a conventional rubbing-contact face seal; therefore, the failure mode caused by loss of lift potential is not as disastrous as that of the other concepts. Test programs (ref. 34) have indicated that this concept is capable of developing hydrodynamic lift with low-viscosity fluids (1000°F gas) if compensation for seal face distortions can be provided. Because seals for cryogenic turbopumps also operate with thermal differentials and low-viscosity fluids, it is reasonable to assume that the Rayleigh-step concept has potential capability for turbopump seals (ref. 39). Other hydrodynamic mechanisms (surface waves, surface microasperities, nonsymmetric rotation) have been investigated; however, the current state of the art is not sufficiently developed for practical application on seals for rotating shafts in rocket engine turbopumps.

2.3.4.4 HYBRID SEALS

The hybrid-seal concepts (fig. 18) utilize the combined hydrostatic and hydrodynamic or combined rubbing-contact and hydrostatic/hydrodynamic principles. The hydrostatic action provides the lift force at low rotational speeds, and the hydrodynamic action provides additional lift force and dynamic stability at high speed independent of the sealed pressure. Use of the concept shown in figure 18(a) has resulted in improved operation with mixtures of oil and gas at low pressures (ref. 45). The concept has not been evaluated with cryogenic propellants.

The combined rubbing-contact and hydrodynamic concepts (fig. 18(b)) offer the advantage of a seal designed for rubbing contact to allow for transient operation and for reduced face-contact load at higher rotational speeds as a result of the additional hydrodynamic lift force. The reduced face load allows higher speeds and longer wear life than the conventional rubbing-contact seals. The potential problems with this concept are thermal distortions of the seal face by frictional heat and wear of the critical face geometry. The depth of the tapered lift pads must be approximately 0.001 in. or less with low-viscosity fluids to develop significant pressure-induced lift forces; therefore, the allowance for face wear is very low. Other methods (e.g., use of face grooves) provide more wear allowance but less lift force induced by hydrodynamic pressure.

2.3.5 Circumferential-Seal Clearance Elements

Clearance-type seals are used when the pressure-speed-life limits of rubbing-contact seals are exceeded and the increased leakage is acceptable. Circumferential-seal clearance elements consist of labyrinth devices, floating rings, and arch-bound* segmented rings. Labyrinth seals have the best reliability, but the leakage is approximately ten times greater than that of the arch-bound seals and approximately five times greater than that of the floating-ring seals. Floating-ring seals generally provide the best compromise between sealing effectiveness and reliability for high-pressure, high-speed, long-life applications. The arch-bound segmented-ring seal provides the most effective sealing of clearance-type seals; however, the potential reliability is lower because of the more complex design and required wear-in.

*Ends of the segments bottom out against each other, so that there are no gaps in the ring (fig. 21).

2.3.5.1 LABYRINTH SEALS

Labyrinth seal elements (fig. 42) are clearance devices that restrict fluid leakage by dissipating the kinetic energy of fluid flow through a series of flow constrictions and cavities that accelerate and decelerate the fluid or change the direction of flow abruptly to create the maximum flow friction and turbulence. The ideal labyrinth transforms all of the kinetic energy at each throttling into internal energy in each cavity. Practical labyrinths, however, generally transfer significant kinetic energy from one throttling to the next. Therefore, the ideal thermodynamic and fluid-mechanics relations are modified with empirical factors for practical solution.

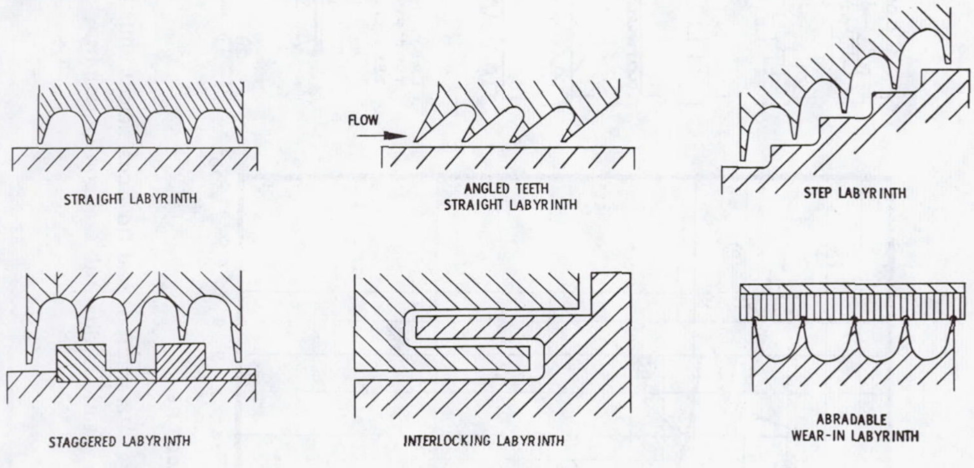


Figure 42. — Six configurations for a labyrinth seal.

Empirical relations for compressible fluids are available for many different types of labyrinths; however, data on incompressible fluids are more limited, and usually it is necessary to either conduct experimental tests to establish the flow factor or estimate a factor from extrapolated data.

The theoretical and empirical methods used to optimize the labyrinth geometry and predict leakage are given in references 54 through 65. Empirical flow coefficients for incompressible fluid based on experimental tests in water are given in figure 43 for some typical turbopump labyrinth seals.

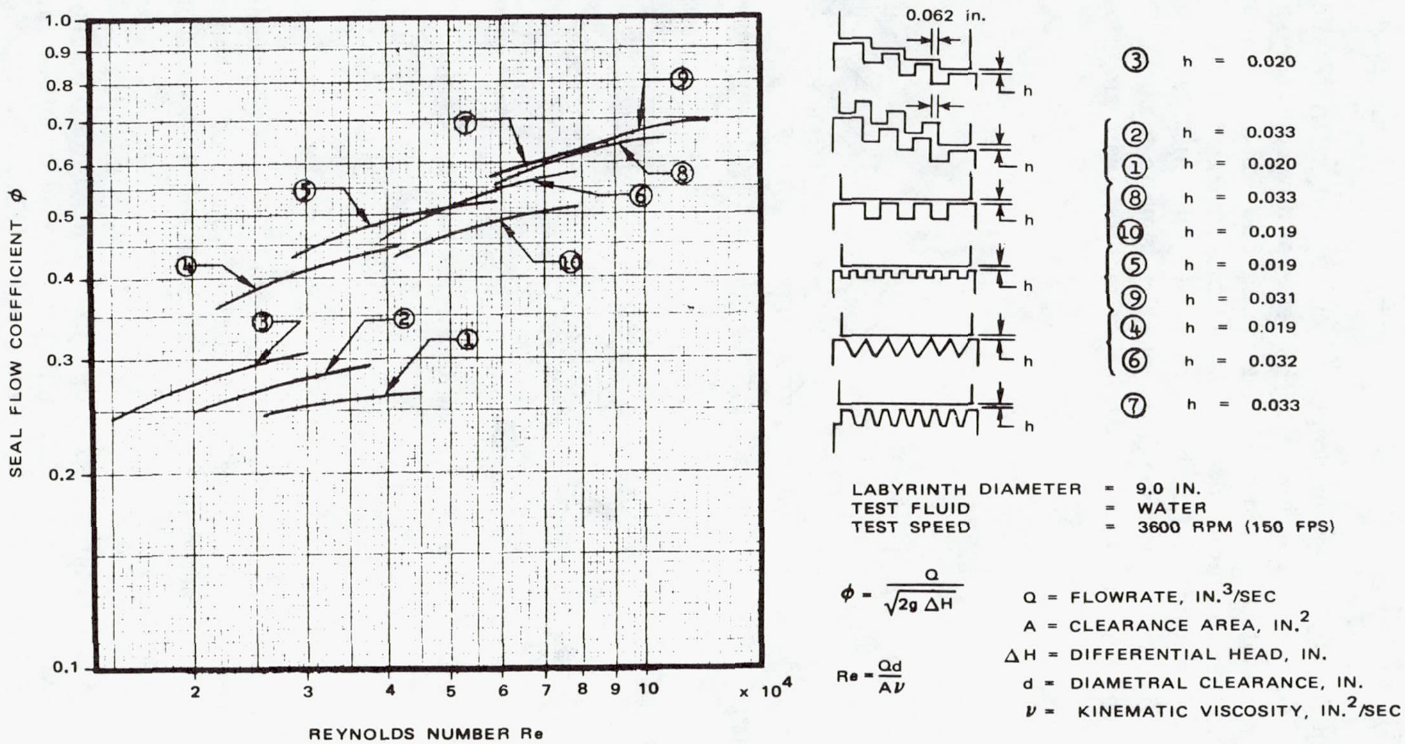


Figure 43. — Variation of seal flow coefficient with Reynolds number, various labyrinth seal configurations.

2.3.5.1.1 Labyrinth Geometry

Optimization of the labyrinth geometry has a significant effect on sealing effectiveness. Leakage flow through a step or staggered labyrinth is approximately 50 percent of the leakage of a straight labyrinth for similar conditions (fig. 43). Step labyrinths require more radial space, are more difficult to manufacture, and may produce an undesirable thrust load because of the unbalanced pressure force. Staggered labyrinths are more complex to machine and require special assembly techniques.

The relative sharpness of the tooth (ratio of tooth tip thickness to clearance) can vary the leakage as much as 20 percent (ref. 60). The teeth tips usually are made sharp (0.005 to 0.015 in. rad.) to minimize the flow coefficient and rubbing area for improved wear-in capability. Test data indicate that a tooth angle of attack of 40° provides the optimum resistance to flow (ref. 62).

Leakage is reduced significantly by increasing the number of teeth or throttlings; however, the effect diminishes with large numbers of teeth. The variation of the leakage function ψ with throttling is shown in figure 44.

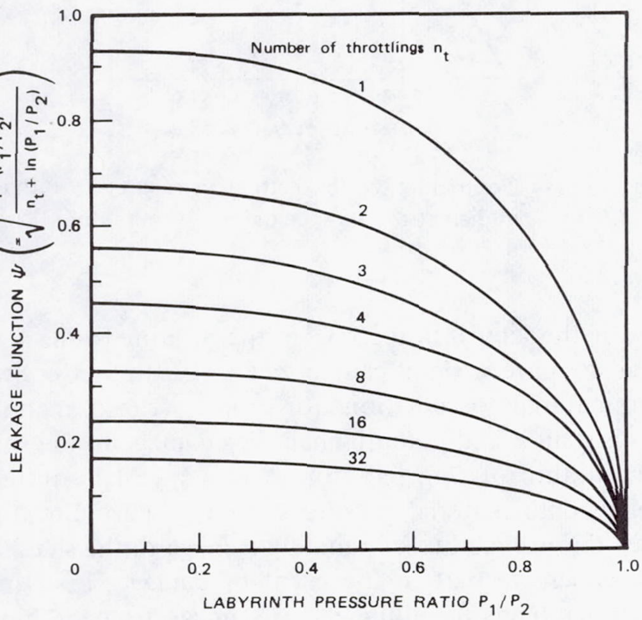


Figure 44. — Leakage function for labyrinth seals as a function of labyrinth pressure ratio (adptd. from ref. 56).

Tooth pitch affects the interstage cavity size and is optimized relative to the number of teeth for a given seal length. The effect of tooth pitch is more significant on straight labyrinths than on step labyrinths. The optimum pitch for straight labyrinths is displayed on figure 45 as a function of diametral clearance. The pitch on step labyrinths is minimized to obtain the maximum number of constrictions. The optimum cavity depth is approximately equal to the tooth pitch (ref. 61).

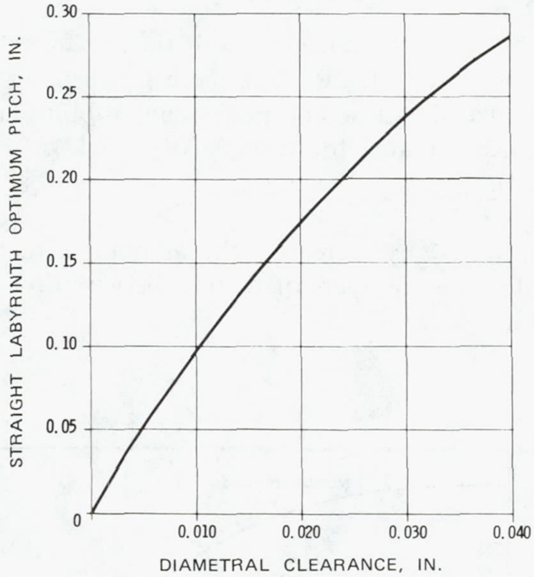


Figure 45. – Optimum pitch for teeth of a straight labyrinth as a function of diametral clearance (adptd. from ref. 56).

Changing the location of the labyrinth teeth from the rotating to the stationary part or vice versa does not appear to have a significant effect on sealing effectiveness. The primary considerations are the material combinations for proper wear-in, manufacturing difficulty, erosion or cavitation resistance, and potential handling damage to the sharp teeth. The teeth generally are located on the rotating part for wear-in-type labyrinths that utilize a soft abradable or thin honeycomb material for the stationary part. Locating the teeth on the stationary part reduces the probability of handling damage to the sharp tips and allows easy replacement of a less expensive part in the event of damage. Locating the teeth on the outside stationary member tends to minimize the damage from rubbing contact, provided that the rotating surface is wear resistant and the supporting structure for the stationary member is flexible. Dimensional stability and fatigue may be problems with flexible seal members.

The thermal expansion caused by the heat generated through rubbing contact is taken into account in order to prevent damage caused by the compounding tendency of decreased clearance and additional rubbing. The temperature of the teeth generally increases rapidly because of the small section in rubbing contact, except when the sealed fluid has sufficient cooling capacity to make use of the large exposed surface area and transfer the heat from the tooth area quickly. A heat-transfer analysis is required to evaluate the thermal effects caused by rubbing contact.

The operating clearance and seal diameter are minimized for maximum sealing effectiveness. The leakage is directly proportional to the operating clearance (assuming a constant flow coefficient) and approximately proportional to the square of the diameter. Small seal diameters reduce the flow area and generally allow closer clearances as a result of better dimensional control. The clearance can be made to increase or decrease at operating temperature by selecting materials with different thermal expansion rates. In some cases it may be advantageous to provide a large clearance for assembly and utilize the thermal-expansion differential to reduce the clearance at operating conditions. The effects of pressure deflection and centrifugal growth of the rotor are also considered.

2.3.5.1.2 Wear-in Labyrinths

Wear-in labyrinths are used to minimize the operating clearance. Conventional labyrinths require additional clearance to allow for accumulation of dimensional tolerances, dynamic deflections, and thermal differentials. The wear-in labyrinths can be installed with practically zero clearance or even a slight interference. Closer operating clearances are obtained if the radial location of the seal housing is adjustable at installation to compensate for concentricity tolerances. The materials are selected to deform easily or wear away during initial contact, so that minimum operating clearance and negligible damage to the rubbing parts are ensured. The grooves formed by the wear-in on a straight labyrinth reduce the flow coefficient to a value between that of a straight labyrinth and that of a staggered labyrinth. The flow coefficient will increase if the sharp edges of the labyrinth teeth are rounded off during wear-in.

Turbopump wear-in labyrinths for non-oxidizing fluids generally have a strip of metal honeycomb material (ref. 66) for the stationary part. The honeycomb (fig. 46) usually consists of Inconel 600, Hastelloy C, or stainless-steel foil 0.002 to 0.005 in. thick with a 1/16-in. cell width; cell depth is one to two times the width. The cells are oriented normal to the direction of rotation. Foil thickness and cell size are a tradeoff between ease of deformation and erosion resistance. The honeycomb strip usually is brazed to the support ring. The advantage of the honeycomb material is that the edges of the thin foil cells are easily deformed or bent by the initial contact of the sharp labyrinth teeth or shaft so that only the required operating clearance results. The fact that the foil is deformed rather than worn away prevents contamination of the fluid system with wear debris.

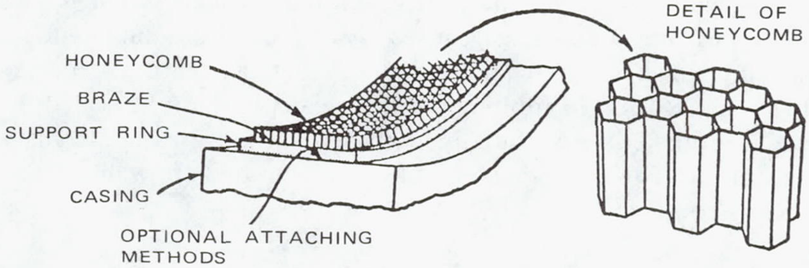


Figure 46. – Honeycomb construction for stationary part of a wear-in labyrinth seal (ref. 66).

Abradable materials constructed from sintered metal fibers have been used successfully for wear-in labyrinths; however, the wear debris may be a problem in some applications. Carbonaceous materials also have been used to provide wear capability; however, carbon is more wear resistant than the honeycomb or abradable materials, and damage to the seal may occur during wear-in if the interference is too large. Carbon has a lower expansion rate than steel, is weak in tension, and is relatively brittle. Therefore, the carbon usually is pressed into a steel ring that provides strength and thermal-expansion control.

2.3.5.1.3 Plastic Labyrinths

Labyrinths for liquid oxidizers require either (1) very large clearances to eliminate the possibility of rubbing contact or (2) compatible materials that can rub without the hazard of combustion. In cryogenics, the low temperature (–297°F) also limits the number of usable materials because of thermal-contraction differentials and loss of elongation. Kel-F materials are compatible with liquid oxygen; however, their thermal contraction rates are approximately four times greater than that of steel, and the materials are not structurally stable. Plastic rings cannot be retained with an interference fit, because of cold flow at ambient or elevated temperatures. The interference fit is decreased after temperature cycling, and the greater contraction rate causes the plastic to loosen at low temperature.

The Kel-F structural problem was solved on the J-2 and F-1 programs by locking relatively thin sections of Kel-F into a metal housing with the rubbing portion exposed (fig. 47). The thin plastic section is restrained from its natural thermal contraction rate with retention locks on the metal housing. Dimensional stability and thermal contraction rate thus are determined by the stronger housing. The housing is designed to provide a tight fit to the plastic. A static seal flange and pressure vents are provided on high-pressure seals to minimize the stresses on the Kel-F caused by the differential-pressure-induced load. The span length between the retention locks is made short ($\approx$ 0.5 to 1.0 in.) to minimize the

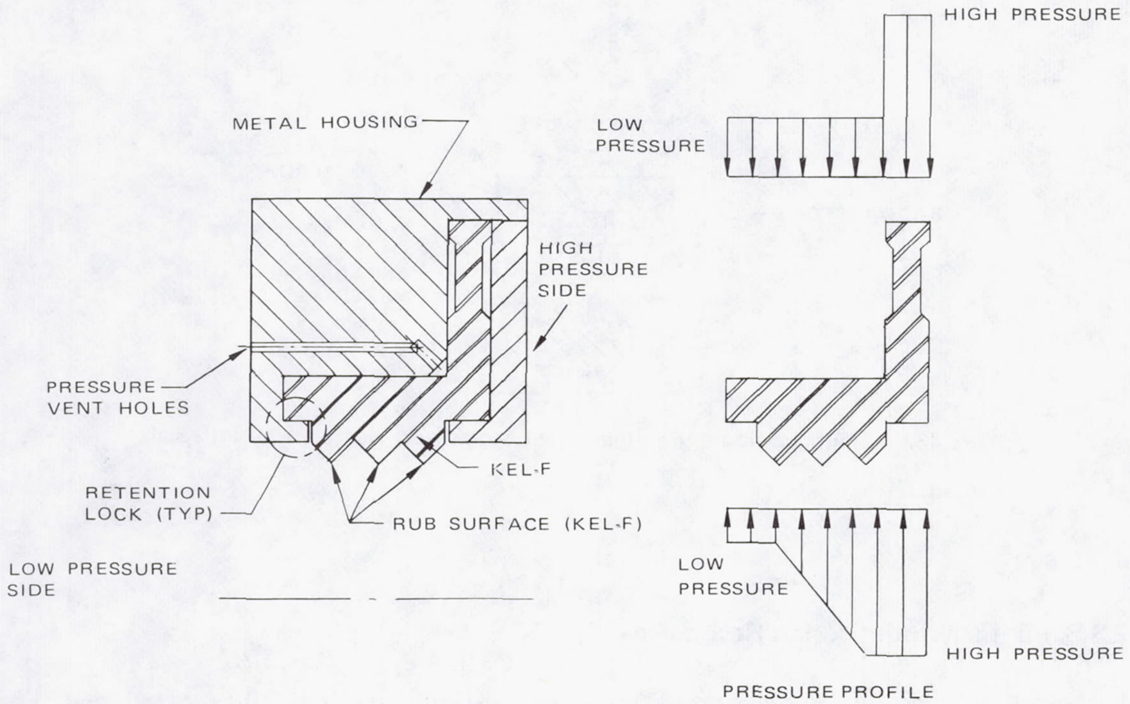


Figure 47. – Configuration and pressure profile for a restrained pressure-balanced plastic wear-in labyrinth seal.

stress on and deflection of the Kel-F caused by the thermally induced load. The wear-in load is minimized by locating the labyrinth teeth in the softer plastic material. Improved resistance to erosion and cavitation damage is obtained by locating the teeth in the metal rotor.

2.3.5.1.4 Segmented Labyrinths

The outer ring of wear-in labyrinths can be segmented and spring loaded to provide additional allowance for radial misalignment. The labyrinth segments are pushed radially inward by the spring and pressure load to a diameter smaller than the rotor. The additional wear-in depth provides minimum effective clearance for large eccentricities. The segments are free to be pushed radially outward by the rotor to prevent excessive contact loads during wear-in or transient operation. High-pressure seals may require pressure balancing to prevent excessive contact loads during wear-in. The segment diameter after wear-in is controlled by limiting the radially inward travel with a T-slot arrangement (fig. 48) or by allowing the segments to become arch bound. A lapped joint for a static seal is provided between the segments and housing. Antirotation devices are provided to prevent segment rotation.

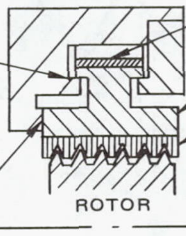


Figure 48. – Configuration of a spring-loaded, segmented, wear-in, labyrinth seal.

2.3.5.1.5 Labyrinth Erosion Resistance

Labyrinth devices, in general, provide the greatest life and reliability of all seal types. However, erosion or cavitation damage and fatigue cracking may limit the operating life. The high fluid velocities caused by large pressure differentials may result in erosion of the labyrinth surfaces. Fluid turbulence in the labyrinth can create localized low-pressure regions where the pressure is below the fluid vapor pressure. The energy released by the collapse of the vapor bubbles may erode the exposed surfaces. Surface erosion apparently attributable to surface fatigue failure is also caused by impingement of high-energy fluid.

The theory of erosion is controversial, and very little data for labyrinth devices are available. A general study of erosion (ref. 67) indicates that the erosion resistance of different materials increases approximately with the 2.5-power of hardness and linearly with material strength. The harder high-strength materials (tool steel, Stellite, maraging steel) are more resistant to erosion than the soft low-strength materials (aluminum, Monel, brass, bronze). The erosion rate generally starts slow, increases to a maximum value, and then decreases with increasing exposure time. Some hard and brittle materials (tungsten carbide, titanium carbide, ceramics) appear to have good initial erosion resistance, but erosion rates increase with exposure time. Additional discussion and references on erosion are given in reference 67.

Fatigue cracking of the labyrinth structure may occur if the flow frequencies correspond to the natural frequency of the structure. Flow turbulence may excite a resonant frequency vibration in the labyrinth components. Stiffening the structural members generally prevents vibration failures of labyrinths.

2.3.5.2 FLOATING-RING SEALS

Floating-ring elements (fig. 23) consist of an inner carbon ring for wear resistance and an outer steel ring for strength and thermal expansion/contraction control. The outer ring material usually is selected to provide the same thermal expansion and contraction rate as the shaft material, so that a constant clearance gap is maintained as the temperature changes. The outer ring is sufficiently strong, relative to the inner ring, to control the diameter of the composite ring.

Extreme temperature environments may result in a temperature differential between the floating ring and the shaft that will cause a variation of the clearance gap. If the ring temperature is lower than the shaft temperature, the clearance will decrease, and ring seizure may occur. This potential failure mode is minimized by allowing sufficient initial clearance to compensate for the maximum thermal gradient.

The inner ring is maintained in compressive hoop stress with an interference fit. The amount of interference is established to maintain the minimum contact unit load higher than the maximum fluid pressure. The fluid pressure acts on the interference joint and causes radial deflection of the inner ring if the interference unit load is lower than the fluid pressure. With a carbon inner ring and steel outer ring, the interference load will decrease at higher temperature because of the difference in thermal expansion rates. An extreme range of operating temperatures requires selection of an inner ring material that has a thermal expansion rate similar to that of the outer ring.

The load induced by unbalanced radial pressure (fig. 49) is supported by the composite ring in compressive hoop stress. The radial deflection caused by the compressive stress is proportional to ring rigidity. The radial section and modulus of elasticity are selected to minimize the deflection. The initial clearance is adjusted to allow for the deflection and provide the desired operating clearance.

The axial force induced by differential pressure (fig. 49) loads the floating ring against the stationary housing to provide a static seal. Low-pressure seals require a wave spring to provide sufficient contact load to maintain a static seal. High-pressure seals are pressure balanced by relieving the axial contact surface and minimizing the housing-to-shaft clearance to reduce the unbalanced load induced by axial pressure. Because of the increased friction force, high axial loads increase the radial load required to reposition the floating ring. The seal-ring wear rate is higher when the friction load is excessive because of the larger wiping force exerted by the shaft required to center the ring with the center of rotation.

The floating-ring element usually is restrained from rotation with two or more antirotation tangs or pins that engage slots. Unrestrained rings have been used; however, if the ring rotates with the shaft as a result of partial seizure or sticking, failure generally occurs. The centrifugal force on high-speed seals may cause excessive deflection and failure. In the

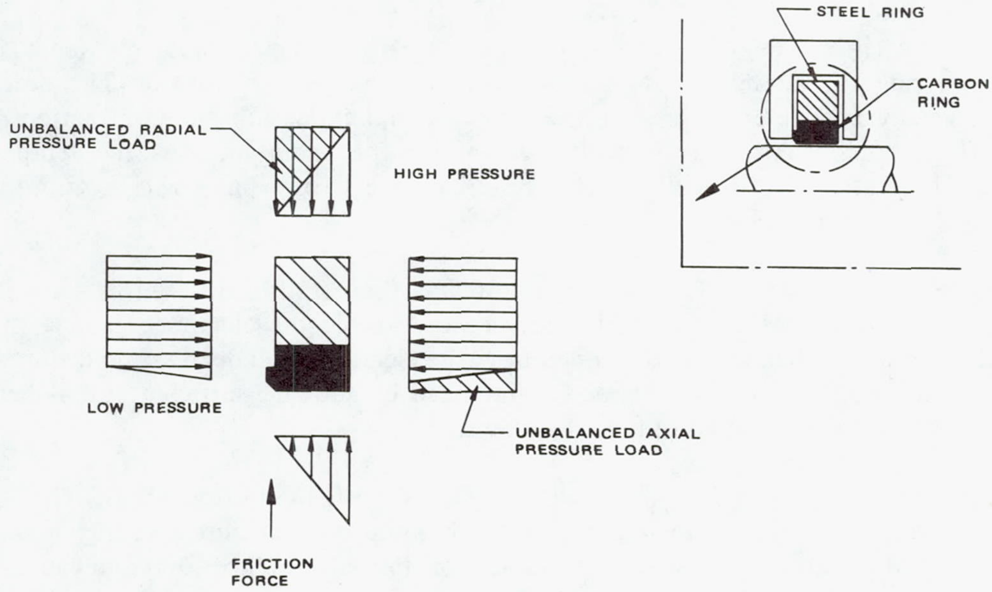


Figure 49. – Pressure forces on a floating-ring seal.

turbine on the H-1 engine, thermal decomposition of the lubrication additive led to shaft sticking that caused centrifugal failure of the floating ring; this problem was solved by adding anti-rotation tangs to the ring. Machined tangs generally are more reliable than press-fit pins because the pins tend to loosen with repeated impact loading.

Additional discussion on controlled-gap seals is given in reference 68.

2.3.5.3 ARCH-BOUND SEGMENTED SEALS

Arch-bound segmented shaft seals (fig. 21) are designed such that the sealing segments are butted together to form a solid ring at the operating diameter. The differential radial-pressure load (fig. 50) is supported by compressive hoop stress in the ring instead of bearing contact on the shaft surface. The reduced rubbing-contact load increases the pressure-speed-life limitations over those of conventional segmented shaft seals.

To allow the segments to wear in, the design diameter of the arch-bound segments at operating conditions is slightly smaller than the shaft operating diameter. To minimize the required wear-in, the segment design diameter is adjusted to compensate for the deflection caused by the load induced by radial pressure and differential thermal contraction or expansion. The bearing contact load decreases asymptotically during wear-in until most of

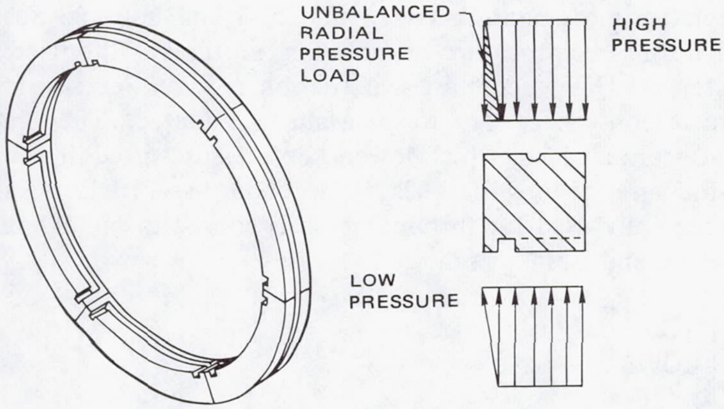


Figure 50. — Pressure-induced radial forces on an arch-bound segmented seal.

the load induced by radial pressure is supported by the butted segments in compressive hoop stress. The segment wear becomes negligible as the contact load approaches zero.

The seal construction is similar to that of the conventional circumferential segmented seals except for the butted joints. Since the sealing ring joints are butted together, additional backup rings to seal the joints are not required, and the design may be simplified by using a single ring similar to that in figure 21. A garter spring around the outside diameter is used to hold the segments together prior to pressure application. The segments are prevented from rotating relative to the housing with antirotation devices. The axial force induced by differential pressure is minimized by pressure balancing as in the floating-ring seal design. A steel washer and wave spring ordinarily are used to hold the segments against the axial seal surface prior to pressure application.

The contact load on high-pressure, high-speed seals may be excessive during wear-in, and the contact surfaces may be damaged before the wear-in is completed. Damage is prevented by conducting the wear-in at lower speed and pressure or by adjusting the design diameter until a line-to-line fit with the shaft diameter is obtained at operating conditions. The segment design diameter may be made large enough to provide a slight clearance at operating conditions to eliminate wear-in. Some trial-and-error adjustments generally are required because it is difficult to predict the exact diameter at operating conditions.

2.3.6 Face-Contact-Seal Secondary Elements

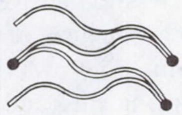
Face-seal secondary elements are devices that seal between the seal face ring and the stationary housing. The secondary element provides for the motion required for the seal face to track the rotating mating-ring surface and the axial travel necessary to compensate for face wear and relative movements between the shaft and housing. Secondary sealing must be maintained during reciprocating or wobbling motions without creating excessive friction drag or hysteresis. High secondary-seal friction requires increased spring force to provide the dynamic response necessary to maintain face contact. The larger spring force results in a higher face-contact load, which lowers the pressure-speed-life limits. Therefore, on high-speed, high-pressure, or long-life seals the secondary seal friction is minimized. The secondary elements generally used for turbopump seals consist of metal bellows, plastic lip seals, elastomeric O-rings, and metal piston rings.

2.3.6.1 METAL BELLOWS

Bellows elements are designed to provide the secondary-seal function, the spring force required to maintain face contact, and structural support for the seal face ring. The bellows element is extendible in the axial direction to compensate for relative axial motions, and sufficiently rigid in the torsional and lateral directions to act as the antirotation and piloting device for the seal face ring. The bellows element thus allows significant simplification of the seal design by eliminating the secondary seal, antirotation device, seal ring pilot, and loading springs. The all-metal construction eliminates periodic replacement of aged elastomer components; it also extends the minimum operating temperature from –65° to – 423°F and the maximum from 500° to 1500°F. Many turbopump applications require all-metal construction because of the extreme temperatures and reactive environments.

Three different basic types of metal bellows are used (fig. 51): welded, formed, and machined. Welded bellows (fig. 51(a)) generally are used because they require significantly smaller space for a given spring rate, compression range, and pressure capacity. The welded-bellows element usually can be designed to require less space than conventional elastomeric seals. Since the available space on turbopumps usually is limited, the smaller space requirement and lower spring rate of the welded bellows are significant considerations and may be determining factors.

The configuration of welded bellows is varied to satisfy specific requirements. Five designs are shown in figure 51(a). The nested ripple design is used to minimize the spring rate and compressed length; the pressure capacity of the nested ripple design is significantly increased relative to the spring rate by the use of double-ply bellows. The single-sweep design provides higher pressure capability and less change of bellows mean effective diameter (MED) with pressure. The flat-plate design provides a constant MED at low pressure differentials. The toroidal design provides constant effective diameter and high-pressure capacity; however, because of the larger space and the higher spring rate required, toroidal bellows generally are not suitable for use on turbopump seals. The nested ripple design often is the best compromise for turbopump seals and is the most commonly used configuration.



NESTED RIPPLE

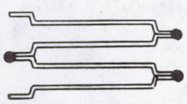
Minimum spring rate and compressed lengths. Large compression range. Requires MED (Mean Effective Diameter) calibration. Fatigue life difficult to predict.

SINGLE SWEEP

Higher pressure capability, less variation of MED, and higher spring rate than nested ripple.

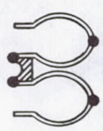
DOUBLE PLY

Higher pressure capability and lower spring rate than above. Requires MED calibration.



FLAT PLATE

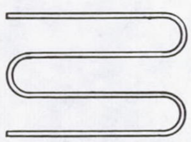
Constant MED for low pressures. Limited compression range and pressure capability.



TOROIDAL

Constant MED and high pressure capability. High spring rate and limited compression range. Large space required.

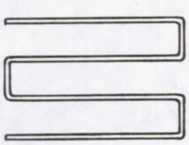
(a) Welded



U-SHAPED

Most reliable and least expensive. High spring rate; large space required. Requires MED calibration.

(b) Formed



RECTANGULAR

Constant MED and high pressure capability. Very high spring rate and limited compression range. Thickness variations due to machining tolerances.

(c) Machined

Figure 51. – Various configurations for metal-bellows secondary elements for face seals.

If space is not limited, formed bellows (fig. 51(b)) generally are used. The fatigue life of welded bellows is difficult to predict because of the variation of the stress-concentration factor at the weld joint. The formed bellows is more reliable and less expensive than the welded type; however, the stress level is also difficult to predict because of thickness variations and residual stresses caused by the forming process. Machined bellows (fig. 51(c)) generally are not used on turbopump seals because of the high spring rate and thickness variations caused by machining tolerances. Many variations of the basic types of bellows are available for special applications.

The analytical techniques for the design of metal-bellows diaphragms are discussed in reference 69. The theoretical analysis of bellows stresses, derivation of the mathematical formulas, and design criteria for some selected configurations are given in reference 70. Simplified formulas and curves for bellows analysis are presented in reference 71. The theoretical relations are based on specific conditions and generally require empirical coefficients for each bellows configuration. Most bellows suppliers have developed empirical relations for the specific configurations they manufacture. The supplier's data usually are more accurate than the values provided by the simplified theory.

The theoretical relation of the design variables for the bellows axial spring rate is given by the following equation (adptd. from ref. 70):

$$K_a = \frac{0.431\ \overline{R}\ E\ t^3\ n_p}{n_c\ s^3\ C_f} \tag{12}$$

where

K_a = axial spring rate, lbf/in.

$\overline{R}$ = bellows mean radius, in.

E = Young's modulus, psi

t = bellows plate thickness, in.

n_p = number of plies

n_c = number of convolutions

s = half of bellows span $\left(\text{i.e., } \dfrac{\text{radial OD} - \text{radial ID}}{2}\right)$, in.

C_f = constant for bellows configuration

The bellows stresses due to pressure and deflection are estimated by utilizing the referenced theory and the manufacturer's empirical data for the specific bellows configuration. The pressure-induced stresses increase as the bellows span, pitch, and diameter increase. Increasing the plate thickness and number of plies reduces the pressure-induced stress. The deflection stresses increase as the deflection and the plate thickness increase. Increasing the span and number of convolutions reduces the deflection stresses. The total stress in a bellows is the sum of the pressure and deflection stresses. Since the variables are interrelated, it is usually necessary to iterate the design to minimize the stresses.

The relative stress level or pressure capacity is evaluated by pressurizing the bellows and measuring the permanent deformation or change of free length. Welded bellows usually yield locally at the initial pressurization; this yielding causes the free length to increase or decrease, the change depending on the bellows design and direction of pressure. To prevent additional deformation during operation, the bellows usually are stabilized with a proof pressurization at the minimum operating compression, followed by stress-relief heat treatment. The effect of bellows free-length variation is minimized by utilizing a mechanical stop to establish the free length. Mechanical stops also are used to prevent excessive deflection stresses caused by overcompression of the bellows.

The total force applied to the seal face by the bellows is the sum of the pressure and spring loads. The pressure-induced load is equal to the product of the differential pressure and bellows effective area. Both spring load and effective area vary with pressure and pitch because of the variation of the effective span caused by deflection of the bellows plates. A portion of the bellows span is inactivated by the plates being pressed together by the force induced by differential pressure (fig. 52). A larger portion is inactivated as the pitch is

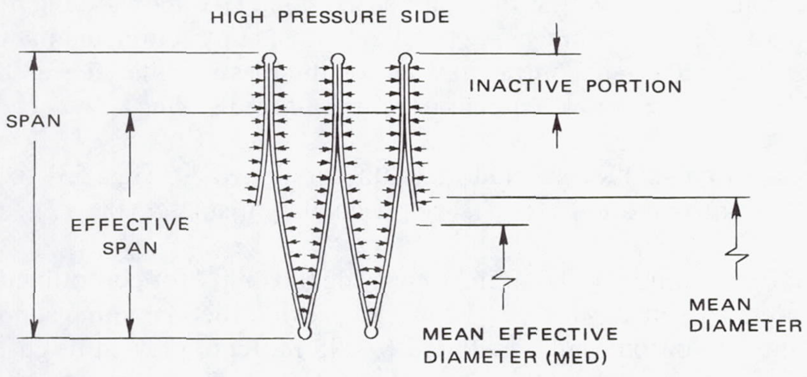


Figure 52. — Variation of bellows effective span and mean effective diameter as a result of pressure-induced deflection.

decreased because of the closer spacing of the plates. The spring load is minimized by using thin plates and the maximum number of convolutions; however, in some cases a lower total load (spring plus pressure loads) is obtained by using thicker plates and a larger pitch to minimize the change of effective span and diameter caused by pressure-induced deflection.

Accurate determination of the bellows load requires calibration testing to measure the total load (spring plus pressure) at the limits of operating pressure and compression. The calibration usually consists of installing the bellows at the nominal operating length in a test fixture and increasing the pressure through the required operating range to measure the load increase. The calibration is repeated for the maximum and minimum operating compression. The total load is converted to an equivalent mean effective diameter by calculating the bellows effective pressure area from the load increase due to pressure. The load increase is the result of the combination of the effective pressure area and the increased spring rate caused by the reduced effective span. Therefore, it is necessary to measure the load at the limits of operating compression to determine the total variation. The calculated effective diameter is used to establish the seal face dimensions for the desired pressure balance. On different bellows of the same configuration the variation of effective diameter usually remains within acceptable tolerances. The effective diameter may be predicted with sufficient accuracy for most applications by utilizing the manufacturer's test data for similar configurations.

2.3.6.2 PLASTIC LIP SEALS

Plastic lip seals are used as secondary seals for cryogenic fluids when the temperature is below the minimum (–65°F) for elastomeric O-rings. The plastic materials (Kel-F and Mylar) maintain adequate elasticity down to approximately –320°F; however, the resiliency is insufficient to compensate for the thermal contraction. The lip-seal design provides for pressure and spring loading to compensate for the thermal contraction and maintain sealing effectiveness. The higher thermal-contraction rate of the plastic material is used to increase the lip diametral interference at low temperature for improved sealing.

The lip seal is supported in the seal housing with a radiused lip (fig. 53) to provide the desired contour for conformation to the seal-ring secondary diameter (fig. 12).

The lip seal is made thin (0.005 to 0.010 in.) to provide flexibility for conformance. The edge of the radiused support is made thin (0.010 in.) to provide the maximum support without interfering with the lip contour and is radiused (0.005 in. R) to prevent its cutting into the plastic. The lip length (generally 0.050 to 0.060 in.) is established to provide sufficient sealing surface at the worst condition of misalignment; the lip length also determines the radial pressure load on the lip. A short lip may not provide sufficient conformance for effective sealing; a long lip results in increased radial load and a greater friction drag force.

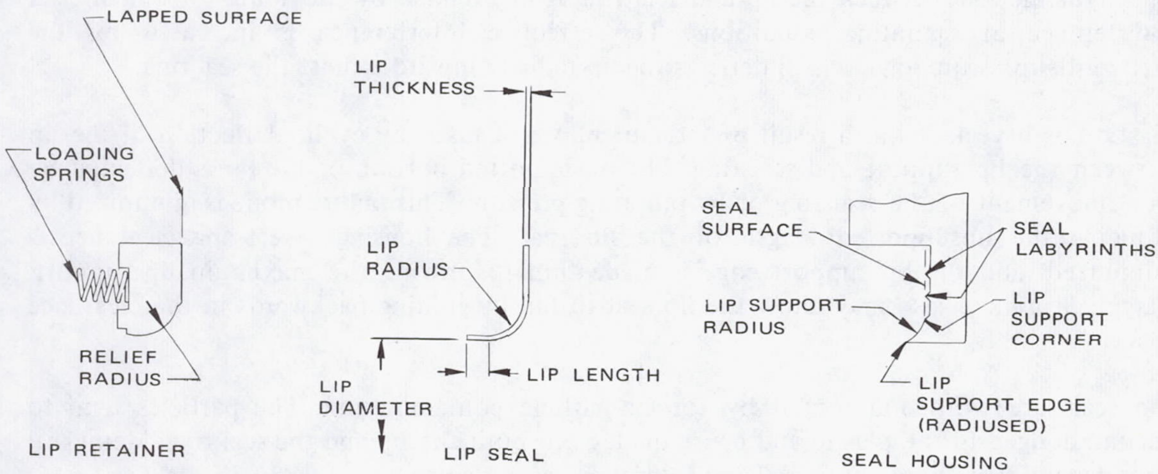


Figure 53. — Lip-seal elements.

The static seal between the lip and housing is maintained by spring loading the lip with a flat retainer ring against a labyrinth surface on the housing. The labyrinths are machined sharp (0.005 in.) and then lapped flat with a narrow (0.005- to 0.010-in.) land to provide a true surface. The retainer ring is relieved along the lip radius to allow pressure loading of the lip seal. The differential pressure force (fig. 54) increases the lip load against the housing for improved sealing.

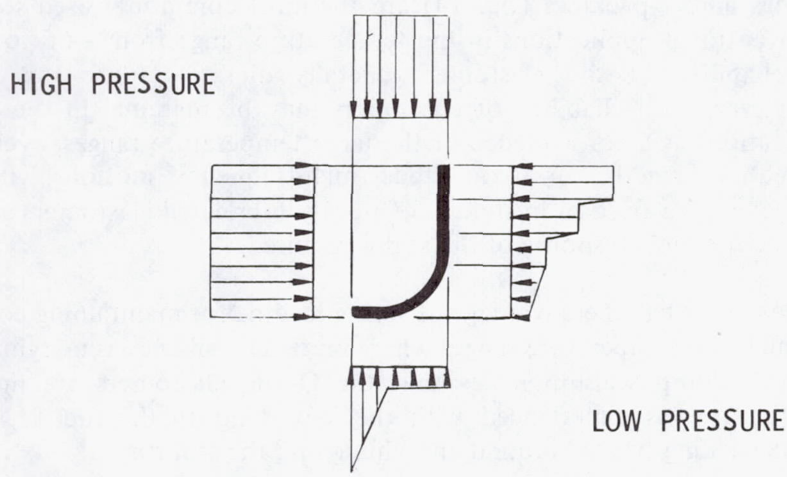


Figure 54. — Pressure forces on a lip seal.

The dynamic seal between the lip and seal ring is maintained by providing for a diametral interference at operating conditions. The effective interference is increased by the differential pressure load, which deflects the lip radially inward against the seal ring.

Plastic lips have failed as a result of fatigue cracking caused by cyclic deflection of the lip between the lip support and seal ring. The unsupported portion of the lip is deflected by axial movement of the seal ring or by pulsating pressure. This failure mode is minimized by reducing the unsupported length of the lip seal. The housing-to-seal-ring clearance is minimized, and the lip support edge is made thin to provide the maximum lip support. Larger clearances also may allow the lip seal to fail by folding backwards in the clearance space.

Lip seals are worn and scored by foreign-particle contamination. The particles tend to become lodged in the plastic and cause damage to both the lip and the seal ring. Metal seal rings generally are hard-chrome plated to minimize the damage.

Plastic lip seals are formed from flat washers that are heated in a contoured fixture to a temperature that allows plastic flow. Kel-F generally is formed in hot water at approximately 180°F; Mylar is formed at 325°F. The forming fixture is designed to prevent residual stresses in the lip, which may cause distortions and wrinkles after the material has relaxed. Antirotation devices were incorporated in the forming fixture for the lip seals for the H-1 engine after seal wrinkling was traced to the fixture being rotated during the forming operations.

2.3.6.3 ELASTOMERS

Elastomeric O-rings and V-packings (fig. 14) are the most commonly used secondary-seal elements for conventional applications in the temperature range from –65 to 500°F. The economy and reliability of the elastomer generally dictate its use for conventional applications. However, the reliability of elastomers may be marginal in the more severe turbopump applications as a consequence of the large temperature ranges, cyclic pressures, inadequate lubrication, reactive or toxic fluids, high-frequency motions, vibration, and shock loads. The excessive friction hysteresis of poorly lubricated elastomers also may be a problem when good dynamic response of the seal is required.

The V-packing elastomer has the advantage of spring loading for maintaining contact at low pressure and in the lower temperature ranges where material resiliency is marginal. However, experience on turbopump seals indicates that the O-ring elastomers are more reliable. Considerable difficulty was experienced with the V-packing on the fuel seal in the H-1 engine because the packing twisted around and "hung up" the seal ring.

The design of turbopump secondary elastomer seals generally follows the standard procedures developed for dynamic reciprocating O-rings. The military design standards for elastomers are given in reference 72. The design considerations for O-ring seals and the design standards for military and industrial elastomers are given in reference 73.

Design considerations for turbopump secondary elastomer seals may vary slightly from military or industrial standards to satisfy specific conditions. The high reliability requirements generally justify using precision tolerances to maintain better control of the O-ring squeeze, concentricity, and clearance gap. The squeeze is minimized to reduce O-ring friction hysteresis. The groove dimensions are adjusted to compensate for thermal expansion or contraction and elastomer shrinkage or swelling caused by incompatibility with the fluid. Floating backup rings are used in the O-ring groove to minimize the clearance gap and prevent O-ring extrusion.

The elastomer and all metal parts in contact with it are lubricated with a compatible lubricant to minimize friction and prevent spiral failures. Spiral failures occur when cyclic motion causes a portion of the O-ring to roll and twist (ref. 73). Rolling and twisting of the O-ring or V-packing also can cause the seal to hang open and leak. To minimize friction and increase wear life, the metal sealing surface generally is hard-chrome plated and polished to a finish of 5 to 10 μin. rms or coated with Teflon. Hardened steel also is used for the sliding surface to improve wear life. The soft metals (e.g., aluminum) generally are not satisfactory as a sliding surface for dynamic elastomers because of excessive friction and wear.

The most commonly used elastomers for turbopump secondary seals are Viton A and Buna N (table III). Viton A is generally preferred since it is not age limited and is resistant to deterioration by ozone. Various shapes and compositions of TFE in combination with metal and rubber spring-loading devices to compensate for lack of resiliency have been used to reduce friction and wear; however, the experience on turbopump seals is limited. The TFE compositions generally allow the operating temperature range of the elastomer to be extended to −100°F minimum and 600°F maximum.

2.3.6.4 PISTON RINGS

Piston-ring secondary seals (fig. 13) were commonly used in extreme temperatures or reactive fluids prior to the development of reliable metal bellows as secondary elements. Most current turbopump applications utilize metal bellows as secondary elements when all-metal construction is required. However, the piston-ring secondary seal has the advantage of a more nearly constant pressure-balance diameter and may be required for close control of face load in some applications. The piston ring is also more resistant to extreme oscillating pressure environments that may cause fatigue failure of metal bellows. The combination of higher pressure capacity and closer face-load control generally dictates the use of piston rings at pressures exceeding 500 psi.

The piston-ring secondary seal inherently has significantly higher leakage than the other types of secondary seals. The sealing effectiveness depends on very precise control of the mating surfaces to maintain sealing contact. In contrast to the elastomer and plastic lip, the metal ring will not conform to irregular surfaces and therefore is sensitive to machining tolerances, thermal distortions, and bending deflections. The piston ring is satisfactory for applications where a controlled leakage rate is acceptable.

The metal-to-metal sliding contact between the piston ring and the mating seal-ring surface also is the source of several problems. High-frequency motions cause wear and deterioration of the sealing surfaces, and vibration causes surface deterioration by fretting erosion. Excessive friction drag may result in poor dynamic response and seal hangup. The material combinations are selected to provide rubbing contact with minimum friction and wear. The effects of thermal contraction and expansion also are considered in the material selection.

The sealing surface that slides against the piston ring generally is hardened steel or hard-chrome-plated steel for wear resistance. The piston rings usually are constructed from cast-iron alloys. Carbonaceous materials and TFE compositions also have been used.

The design and material considerations for piston rings are discussed in reference 74.

2.3.7 Spring Load

The stationary face-contact seal ring is loaded against the rotating mating surface with a spring device to ensure sealing contact. The spring force generally is a compromise between the minimum load for effective sealing and the maximum load for the load-speed-life relationships.

The total seal-face load consists of the spring force plus the resultant pressure-induced forces; therefore, the spring force is determined in conjunction with the pressure balance to establish the desired total load. The spring force is significant on low-pressure seals because of the negligible pressure-induced force; conversely, the spring force generally is negligible on high-pressure seals because of the large pressure-induced forces.

The minimum spring load for sealing effectiveness is established by the dynamic response, secondary-seal or vibration-damper friction, and variations in interface pressure profile. The spring force for dynamic response is established by calculating the force required to accelerate the seal ring at the rate required for the seal ring to track the runout of the rotating mating surface. In some cases, the acceleration rate may be established by sudden movements of the turbopump shaft during transient operation. The resultant closing force (spring plus pressure) must be greater than the sum of the seal-ring inertia, secondary friction, and interface average pressure profile forces for effective sealing. Allowance for variation in interface pressure profile is provided by either additional spring load or a larger pressure-balance ratio.

The maximum spring load is limited by the load-speed-life relationships for specific fluids and materials. The load and speed limitations based on current practices are reasonably well defined for a wear life of approximately 3 hr. The life relationships, however, are not as well defined, because of the limited long-duration testing in rocket propellants. Therefore, the spring load is minimized on seals that have wear life requirements greater than approximately 4 hr.

The maximum spring load also may be established by the rubbing-friction power loss at the seal interface. In small turbomachinery, it is possible for the seal power loss to become a significant portion of the total turbopump power. The heat input caused by seal power loss may be significant on systems that utilize recirculating coolant fluid. Heat-balance calculations generally are performed to establish the allowable heat input.

The current practices for spring loading face seals in various applications in representative turbopumps are shown in figure 55. Most turbopump seals are designed for a spring load of

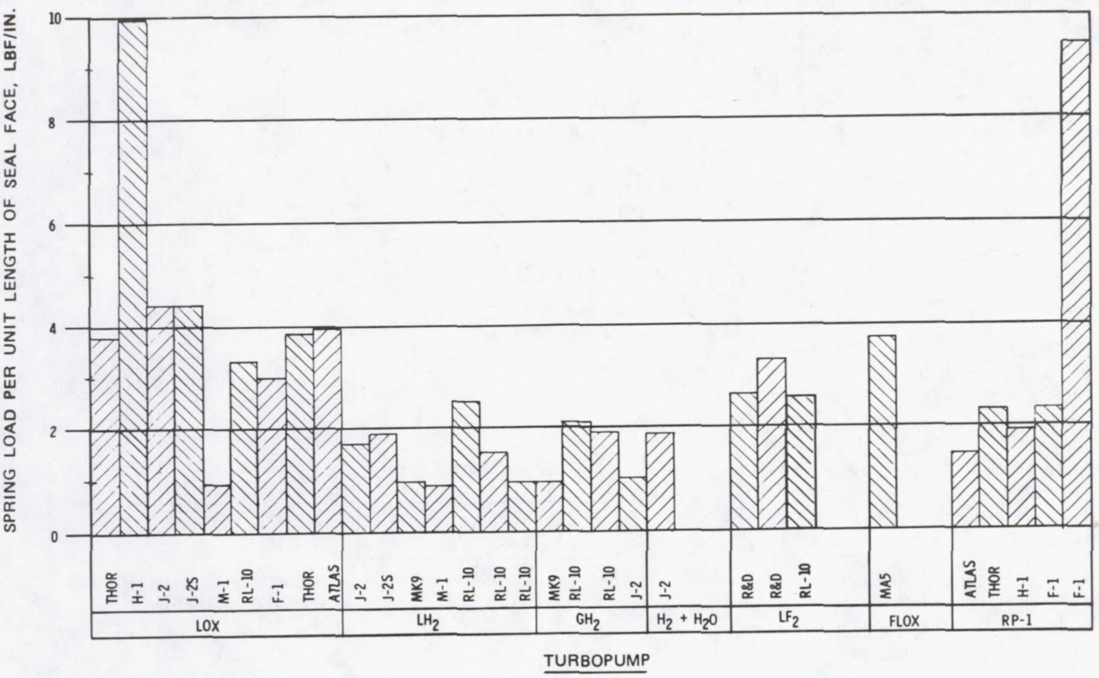


Figure 55. – Current practices for spring loading face-contact seals.

approximately 2 lbf/in. of seal face circumference. The minimum practical spring load is approximately 0.3 lbf/in. A maximum spring load of approximately 10 lbf/in. was used successfully to minimize leakage on the liquid-oxygen seals on the H-1 engine and the RP-1 seals on the F-1 engine; however, spring loads higher than approximately 4 lbf/in. are not used unless the cooling capacity of the sealed fluid is sufficient to dissipate the heat generated by the rubbing friction.

The spring loading devices consist principally of coil springs, wave springs, and metal bellows. Several small coil springs around the seal ring provide the most uniform face loading. The spring rate usually is minimized to reduce the load variation with operating compression. Low-rate springs and bellows are preloaded and restrained with a mechanical extension stop when higher operating loads are desired.

2.3.8 Pressure Balance

Face seals are pressure balanced as closely as possible to minimize the variation in face load caused by the forces induced by differential pressure (fig. 56). Pressure balance is particularly important when fluid pressures are greater than approximately 100 psi, because of the large pressure-induced forces on the seal face. It is not feasible to balance the

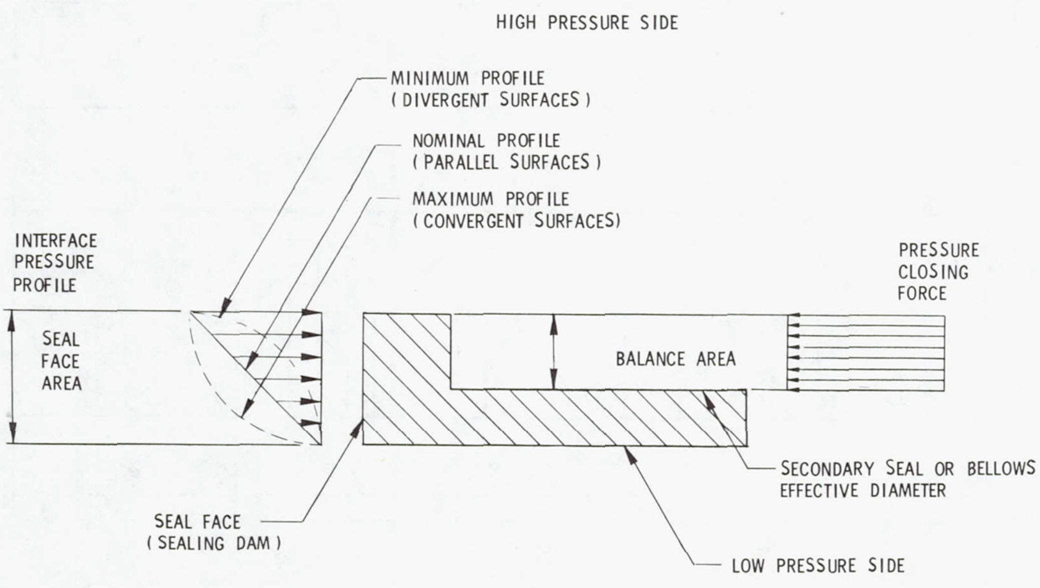


Figure 56. — Pressure-induced forces on face seals.

pressure-induced forces completely because of the variation of the seal interface pressure profile. Practical designs require a safety margin that allows for the maximum variation in interface pressure profile so that face separation can be prevented.

The average interface separating pressure is equal to the pressure profile factor α times the differential pressure:

$$\text{average interface separating pressure} = \alpha\,(P_1 - P_2) \tag{13}$$

The theoretical pressure profile factor for steady-state laminar or turbulent flow of an incompressible fluid between parallel sealing surfaces is approximately 0.50 (refs. 9, 75, and 76). This factor can increase to approximately 0.80 for convergent surfaces and can decrease to approximately 0.20 for divergent surfaces.

The theoretical pressure profile factor α for subsonic laminar flow of a compressible fluid between parallel sealing surfaces varies between approximately 0.50 and 0.67; the lower pressure ratios (P_1/P_2) give factors tending toward 0.50, and the higher pressure ratios produce factors tending toward 0.67, as shown by the following equation:

$$\alpha \approx \frac{1}{3}\left[1 + \frac{P_1/P_2}{1 + (P_1/P_2)}\right] \tag{14}$$

The theoretical pressure profile factor for flow of a compressible fluid between parallel sealing surfaces with isentropic entrance conditions and choking (sonic flow) at the exit varies between approximately 0.67 and 1.00; flow just barely choked gives factors tending toward 0.67, and very highly choked flow gives factors that approach 1.00. Leakage flow can become choked at the exit for pressure ratios greater than about 4:1 ($\Delta r/h > 100$). The experimental data indicate closer agreement to the theoretical anlaysis when an entrance-loss coefficient of 0.6 is used (ref. 12). Since entrance losses tend to reduce the average interface separating pressure, the pressure profile factor for choked flow is reduced when the entrance losses are considered. The theoretical pressure profile factor for choked flow with an entrance-loss coefficient of 0.6 varies between approximately 0.5 and 0.65 for pressure ratios from 1 to 10.

The pressure profile factor is affected by turbulence (both flow induced and rotation induced), change of fluid state, viscosity, seal clearance, and interface geometry. The effect of rotation on the pressure profile factor usually is not significant in sealing cryogenic or compressible fluids.

Pressure balance is achieved when the separating force on the seal induced by interface pressure is equal to the pressure-induced closing force. The pressure balance ratio is defined as the ratio of the effective closing or balance area (fig. 56) to the seal face or dam area:

$$\text{pressure balance ratio} = \frac{\text{closing area}}{\text{face area}} \tag{15}$$

A seal is pressure balanced when the pressure balance ratio is equal to the interface-pressure profile factor. Therefore, a balance ratio of 0.5 corresponds to the theoretical pressure balance for laminar incompressible flow across parallel sealing surfaces.

The balance ratio used on practical designs varies from 0.55 to 1.0, the value depending on the operating conditions and the desired safety margin. High-pressure rubbing-contact seals are relatively close-balanced to minimize the face contact load; therefore, the margin for variation in the interface pressure profile is reduced, and the seal is more sensitive to face separation caused by face distortion or fluid vaporization across the interface. The high pressure may cause additional deflections of the seal face and mating ring surfaces; these deflections increase the interface pressure profile and result in greater separating forces. The maximum sealing effectiveness is achieved with large balance ratios. A balance ratio of 0.8 or greater generally provides for the maximum variation of the interface pressure profile that results from converging surfaces, choked flow, and fluid vaporization. Low-pressure cryogenic seals usually are designed with a balance ratio of 0.85. Seals for high-pressure incompressible fluids generally have a balance ratio of 0.55 to 0.6; seals for high-pressure compressible fluids have a ratio of 0.7. Most conventional seal designs use a balance ratio of 0.65.

The pressure-balance ratio usually is selected in conjunction with the spring load to provide a total (spring plus pressure) face-contact load that is consistent with the load-speed-life relations and the minimum load requirements for effective sealing.

The variation in the face and balance diameters because of thermal contraction or expansion, pressure deflection, and dimensional tolerances must be considered for accurate determination of the balance ratio. Precision tolerances ordinarily are used on turbopump seals to minimize the balance-ratio variation. The face diameters on carbon inserts usually are machined after installation to reduce the tolerance accumulation. Face diameters on bellows seals usually are machined after the bellows effective diameter is measured.

3. DESIGN CRITERIA and Recommended Practices

3.1 SEAL SYSTEM

3.1.1 Pressure Environment

The system design shall minimize the severity of the seal pressure environment.

Reduce high pressure levels (> 500 psig) by utilizing upstream labyrinths or circumferential clearance seals in conjunction with low-pressure return bleeds that allow recirculation back to a low-pressure area (fig. 10). Reduce the differential pressure by maintaining an established pressure level downstream of the primary seal. Minimize the amplitude of pressure oscillation by utilizing labyrinths upstream of the seal as a damping device (fig. 4).

3.1.2 Thermal Environment

The system design shall minimize exposure of the seals to high-temperature environments and severe thermal gradients.

Minimize high-temperature environments at the seal location by designing the system to isolate the seal in a cooled area. Provide barriers or labyrinth devices to prevent the direct flow of high-temperature fluids onto the seal assembly or its components. The thermal stresses created by the differential thermal expansion/contraction should be held to a minimum by designing the seal housings and mating rings to provide minimum restraint to thermal deflections and by using materials with similar thermal expansion/contraction rates.

3.1.3 Vacuum Environment

The system design shall preclude exposure of rubbing seals to hard vacuum.

Route the seal drains to a safe disposal area internal to the turbopump or engine (e.g., the turbine exhaust or pump inlet); when the fluids are compatible, eliminate the drains and drain the seal leakage directly into the turbine (fig. 3).

3.1.4 Rubbing Speed

Seal rubbing speeds shall be at a minimum consistent with the turbopump design requirements.

Use the smallest possible seal diameter to minimize the rubbing speed.

3.1.5 Cooling and Lubrication

The system design shall provide for adequate cooling and lubrication of the seals.

Cooling and lubrication of the primary shaft seal and mating ring should be provided by allowing the sealed fluid to flow through the seal cavity.

Where possible, utilize bleed holes through the impeller from the seal cavity to the impeller inlet or through the shaft back to the pump inlet. Pumping vanes on the impeller rear shroud also may be utilized to ensure recirculation of the sealed fluid. The seal should not be installed in a deadend cavity that may allow accumulation of stagnant vapor.

Dry-running intermediate seals should be circumferential shaft seals with an inert-fluid purge for cooling.

Additional cooling and lubrication should be provided when required by purging with an inert fluid or by injecting a cooling fluid into the seal cavity or onto the seal mating ring. Separate cooling and lubrication systems isolated from the sealed fluid should be used when the cooling fluid is not compatible with the sealed fluid.

For maximum heat dissipation, locate mating rings with the pressure on the outer diameter of the sealing interface. The use of one mating ring with two rubbing contact seals should be avoided because of overheating and distortion problems. Separate mating rings for each seal should be used.

3.1.6 Leakage Drains

The system design shall provide drains for safe disposal of expected seal leakage.

Size the seal drains to accommodate the maximum anticipated leakage without building up a significant back pressure. Calculate the drain effective flow area with conventional flow equations by estimating the maximum anticipated seal leakage and establishing the allowable drain cavity pressure. The drain pressure differential and leakage rate will then establish the required drain size.

3.1.7 Fluid Separation

The system design shall provide safe separation of incompatible fluids.

Incompatible fluids on the same shaft should be separated by utilizing two face-contact seals to minimize the leakage; provide separate drains for each propellant to vent the leakage to a safe disposal area and either a purged double-circumferential seal (fig. 6) or two intermediate face-contact seals with a purge between to separate the drain cavities (fig. 10(a)). The purge pressure should be high enough to provide a pressure barrier between the drains to prevent mixing of the propellants by leakage through the intermediate seals.

3.1.8 Fail-Safe Provisions

The system design shall provide that a single seal failure will not cause failure of the turbopump.

Conduct a theoretical failure analysis to estimate the operating parameters and the possible results for each different failure condition; all different modes of operation and variations of performance should be considered. Size the seal drains to provide for the maximum leakage that would result if a single seal fails. The intermediate seal purge pressure should be established higher than the maximum drain back pressure. Redundant seals in series should be used when failure of a single seal would cause turbopump failure.

3.1.9 Purge Requirements

The system design shall provide seal cavity purges adequate to remove trapped fluids and to dry accumulated moisture.

Provide purged cavities with an inlet and outlet port to allow the purge gas to flow through the cavity. The purge flowrate and length of time should be established to ensure complete removal of air and moisture prior to system chilldown. After the test, the purge should remain on or air should be prevented from re-entering the cavity until the hardware returns to ambient temperature.

3.2 SEAL ASSEMBLY

3.2.1 Pressure Capability

The seal shall perform satisfactorily at the maximum expected operating pressure.

Select the seal type (table II) that will satisfy the pressure-speed-life relation with the minimum leakage and maximum reliability. The recommended limits of the fluid pressure-speed relation for turbopump face-contact seals (3-hr life) in liquid oxygen, liquid hydrogen, liquid fluorine, gaseous hydrogen, RP-1, and hot gas are shown by the solid curves in figure 57. The current practices are represented by the circular data points.

The recommended limits are estimated by assuming that the face-contact load increases in proportion to the fluid pressure and that the limit is established by the seal interface heat buildup, which is a function of a constant fluid-pressure/speed relation for each fluid. The limiting pressure-speed relation was established by the relative success of the current applications. The limits shown are to be used as approximate guides before the seal detail design is established. The limits are based on commonly used seal materials (table III) and a wear life of approximately 3 hr. Long-life seals will require a more conservative pressure-speed factor or noncontact seal interfaces. The following recommendations are based on current turbopump practices:

- Face-contact metal-bellows seals (fig. 11) should be used for cryogenic or reactive fluids at pressures up to approximately 500 psig.

- Face-contact piston-ring seals (fig. 13) should be considered for high-pressure (> 500 psig) cryogenic or reactive propellants when face-load control is critical.

- Face-contact elastomeric seals (fig. 14) should be used at pressures up to approximately 1000 psig for conventional lubricated applications in the temperature range of –65° to 500°F.

- Circumferential shaft-riding segmented carbon seals (fig. 20) should be used for low-pressure (< 100 psig) purged intermediate or hot-gas applications.

- Circumferential floating-ring controlled-gap shaft seals (fig. 23) should be used for high-pressure (> 100 psig) hot-gas, purge-gas, or long-life (> 4 hr) applications when the increased leakage is acceptable.

- Circumferential labyrinth seals (fig. 24) should be used for high-pressure and long-life applications when reliability and economy are the primary considerations and the increased leakage is acceptable.

- Face-type hydrostatic/hydrodynamic seals (figs. 15 through 18) should be considered for combined high-pressure, high-speed, and long-life applications when minimum leakage is desired.

- Variations or combinations of the basic seal types in table II should be considered for special problems.

3.2.2 Temperature Capability

The seal shall perform satisfactorily at the operating temperature of the application.

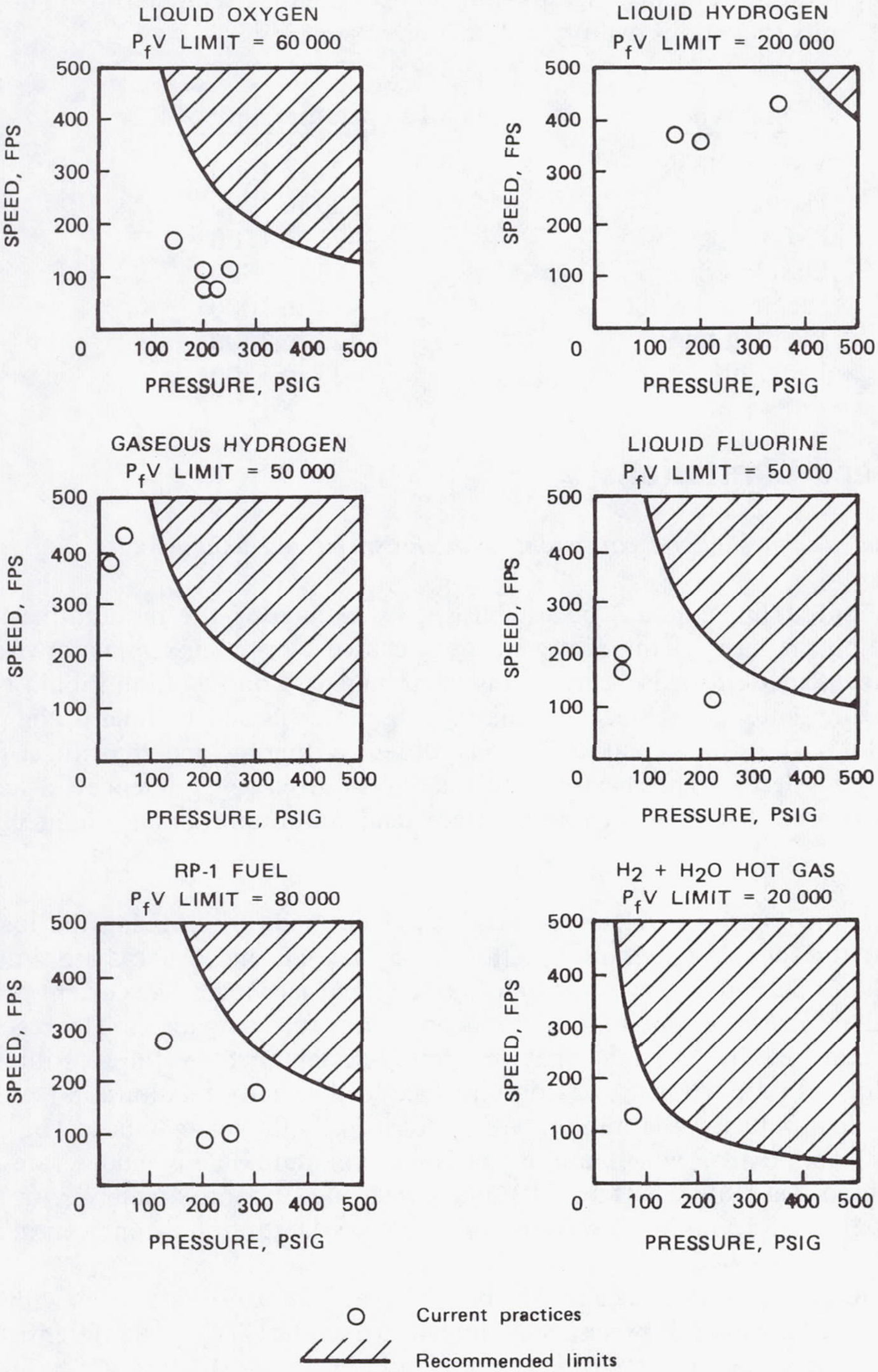


Figure 57. – Current practices and recommended limits for P_fV factor for face-contact seals (3-hr life).

The seal assembly configuration should allow utilization of materials that have satisfactory properties over the maximum range of anticipated operating temperatures. The following temperature limits for various seal types should be observed:

Seal type	Temperature limits, °F
Metal bellows	–423 to 1500
Plastic lip	–320 to 200
Piston ring	–423 to 1500
Elastomeric	–65 to 500
Segmented carbon	–423 to 1000
Floating ring	–423 to 1500
Labyrinth	–423 to 1800

3.2.3 Speed Limitations

The seal design shall be consistent with the speed and load-velocity limitations.

The surface speed limit should be established by estimating the resultant seal interface temperature on the basis of the heat generation caused by rubbing contact or viscous shear and the heat dissipation to the surrounding environment. Consideration should be given to the temperature limits of seal-face and mating-ring materials and thermal decompositon of the sealed fluid. The stresses and deflections caused by thermal and centrifugal forces also should be considered. The heat-transfer methods given in references 6 and 7 are recommended for calculating the seal interface temperature profile and thermally induced deflections.

Estimates of the face load-velocity limits should be made by utilizing the load-velocity relationships for specific materials and fluids. The recommended limits are shown by the curves in figures 58 and 59 as a function of velocity and face load. The current practices are shown by the circled data points on the same figures for comparison. The recommended limits are estimated by assuming that the limit is established by the seal interface heat buildup, which is proportional to a constant face load-velocity relationship for each fluid. The limiting load-velocity relationship was established by the relative success of the current applications. The limits, which are to be used as approximate guides, are based on commonly used seal materials (table III) and a wear life of approximately 3 hr. For longer life, a seal will require a more conservative load-velocity factor or noncontact seal interfaces.

The face load on high-speed seals should be minimized by utilizing designs with minimum spring-load requirements and precise pressure-balance control. The circumferential clearance seals (e.g., floating-ring and labyrinth) or the fluid-film seals (hydrostatic/hydrodynamic) should be considered when the surface speed is higher than approximately 500 ft/sec.

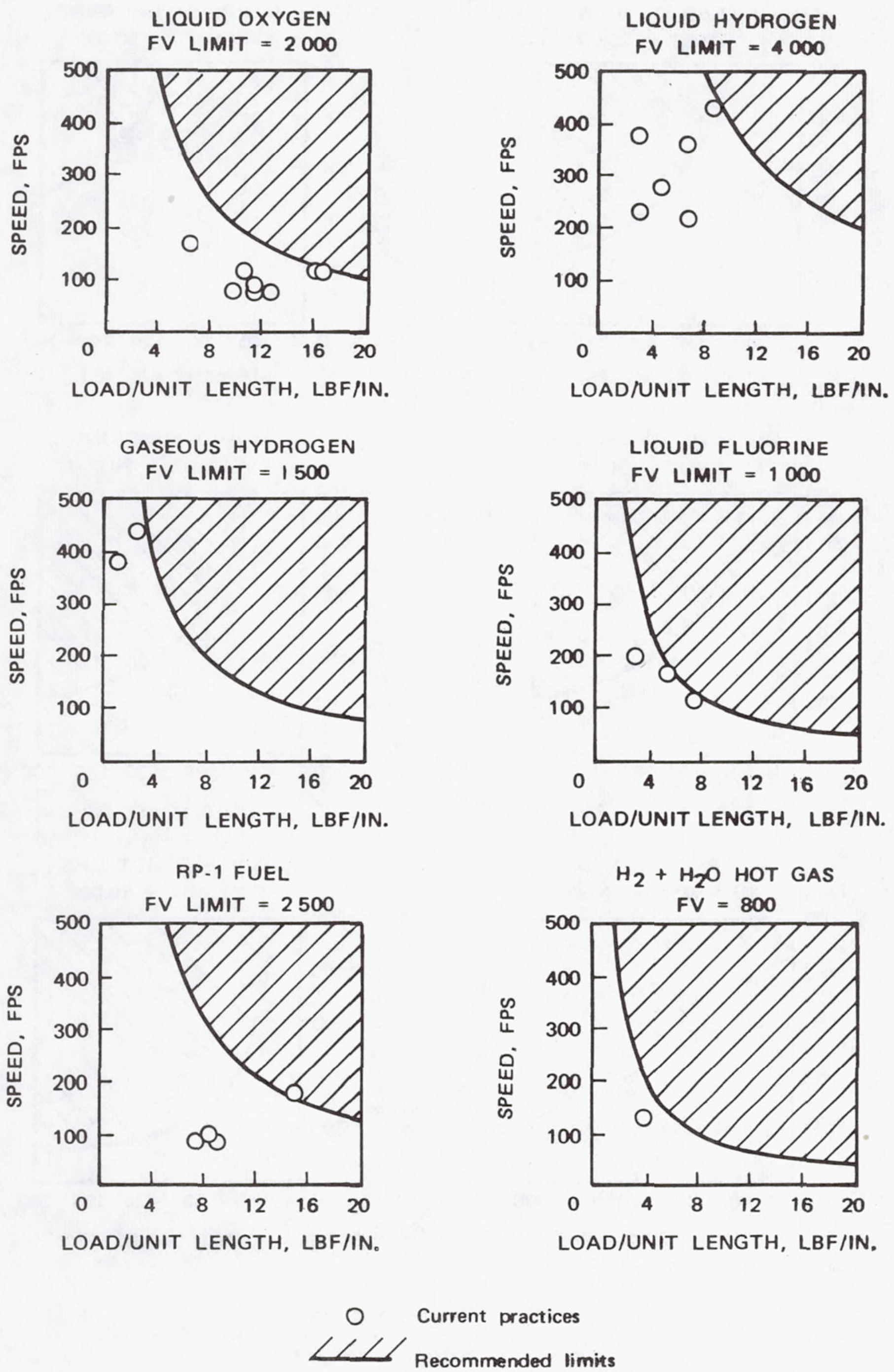


Figure 58. – Current practices and recommended limits for FV factor for face-contact seals (3-hr life).

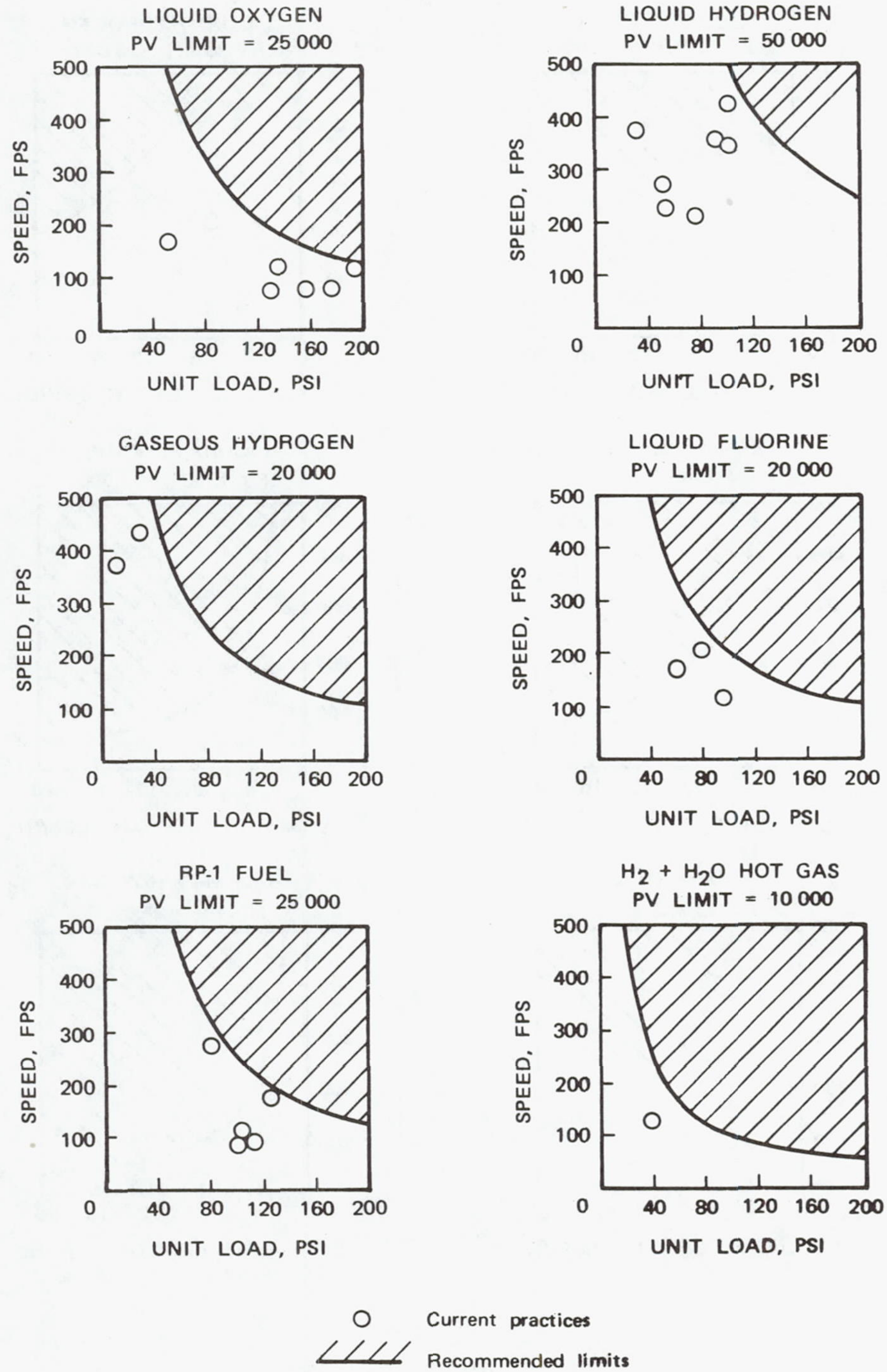


Figure 59. — Current practices and recommended limits for PV factor for face-contact seals (3-hr life).

3.2.4 Wear Life

The seal wear life shall satisfy the turbopump life requirements.

Provide sufficient height on the seal contact face to allow for the maximum anticipated wear rate. The wear rate should be minimized by providing some form of lubrication (e.g., hydrodynamic, boundary, deposited film) and by using the lowest possible face-contact load. Except when the PV factor is low and good lubrication is available for contact seals, the clearance or fluid-film seals should be considered when wear-life requirements exceed 4 hr.

3.2.5 Leakage

The seal leakage shall be the minimum possible consistent with the required operating conditions.

Face-contact metal-bellows or elastomeric seals with the maximum face contact load allowed by the load-velocity relations for specific materials and fluids should be used to minimize seal leakage. The face-contact load should be decreased as the speed and life requirements increase. The hydrostatic/hydrodynamic concepts should be considered for effective sealing at load-velocity-life relations greater than those allowed by the current state of the art for rubbing-contact seals.

Conduct a thorough heat-transfer and stress analysis of the seal face and mating surface so that seal interface distortions that can cause excessive leakage (refs. 6 and 7) can be anticipated and minimized.

To size the drains and predict propellant losses, estimate the seal design leakage by using the available theoretical methods and empirical relations (refs. 8 through 15). It is recommended that the dynamic leakage for cryogenic face-contact seals be estimated by utilizing either the laminar-flow-with-inertia or the turbulent-flow-without-inertia theory (the choice depending on the value of the Reynolds number), with the assumptions of liquid conditions and a leakage-path effective gap of approximately 200 μin. The static leakage should be estimated on the basis of a leakage-path effective gap of approximately 50 μin.

3.2.6 Misalignment Tolerances

3.2.6.1 AXIAL OPERATING LENGTH

The seal axial operating length shall provide for installation tolerances, thermal expansion/contraction differentials, dynamic deflections, and face wear.

Design the seal bellows or loading springs and the internal clearances to allow for the maximum tolerance variations. Tolerance stackups with consideration of thermal effects and stress analysis with consideration of dynamic deflections should be performed.

3.2.6.2 RADIAL LOCATION

The radial location of the seal face relative to the center of rotation shall prevent excessive wiping action.

It is recommended that the radial location of face-contact seals be held within approximately 0.003 in./in. of diameter on high-speed (> 200 ft/sec) seals. Shaft-riding seals should be held within approximately 0.005 in./in. of diameter.

3.2.6.3 MATING SURFACE NORMALITY

The nomality of the seal mating surface shall be such as to prevent excessive mating ring wobble.

It is recommended that the normality of the seal mating surface (as installed) to the center of rotation be held within approximately 0.0001 in./in. of diameter (T.I.R.) on high-speed seals.

3.2.6.4 ROTATIONAL ECCENTRICITY

Shaft-riding-seal designs shall provide for radial location of the mating surface that prevents excessive rotational eccentricity.

The mating surface diameter on high-speed shaft-riding seals should be held concentric to the center of rotation within 0.0005 in./in. of diameter (T.I.R.).

3.2.7 Vibration Control

The seal vibration damping shall prevent resonant vibrations that result in excessive leakage or fatigue failure.

Control seal vibration by designing the seal for a natural frequency that is higher than any expected on the turbopump or by providing mechanical-friction dampers (fig. 11) at the seal-face outside diameter or at the bellows component. The friction drag force for effective damping of bellows seals should be approximately 5 to 10 percent of the seal spring load. The secondary-seal friction on elastomeric, piston-ring, and lip seals usually is adequate for effective damping.

To eliminate vibration caused by stick-slip conditions, select rubbing material combinations to provide self-lubrication or provide lubrication at the rubbing seal face.

3.2.8 Contamination Allowances

The seal shall tolerate the foreign-particle contamination allowable in the application.

Use rubbing-contact face seals with tungsten carbide, titanium carbide, or aluminum oxide face materials and metal-bellows secondary element for maximum resistance to abrasive environments in those applications in which the materials are compatible. Special protection systems (neutral fluid injection, buffer zones, centrifugal separators, slingers) should be used if the contamination is severe. Where possible, use system filters in the range 10 to 100μ to clean the sealed fluid.

Avoid installing the seal in a sump area that would tend to collect foreign-particle settlings.

3.2.9 Seal Mounting Requirements

3.2.9.1 SEAL PILOT

The seal pilot shall provide precise radial location with allowance for thermal differentials.

Use seal housing materials that have thermal contraction/expansion rates similar to that of the pump housing, or adjust the ambient fit to allow for the thermal differential.

Use three equally spaced radial pins engaging radial slots or radial splines to compensate for temperature gradients larger than allowable for diametral pilots.

3.2.9.2 FLANGE SEAL

The flange static seal shall compensate for thermal contraction.

Use metallic, spring-loaded, pressure-actuated static seals to compensate for the dimensional variations caused by thermal contraction at the sealing joint.

3.2.9.3 FLANGE LOADING

The flange loading shall maintain the seating stress required for effective sealing.

To maintain flange loading, use several small bolts around the flange with a sufficient preload to cause elastic elongation. Avoid using large ring nuts, because of the low elastic elongation and nonuniform loading.

3.2.9.4 SHIMS

The shimming provision shall be adequate to maintain the required installation tolerance.

To adjust the seal axial installed length, use ground metal gasket-shims coated with a softer material (copper, silver, Teflon); use spacers with grooves for spring-loaded static seals.

3.2.9.5 PROVISION FOR SEAL REMOVAL

The seal design shall provide for removal of stuck or wedged seals.

Include threaded holes in the flange or recessed grooves that can be engaged by special pulling tools.

3.2.9.6 LOCKING DEVICES

Bolts and nuts used to mount the seal shall be securely retained.

Use positive locking devices to prevent loosening of bolts and nuts. When possible, bolts and nuts should be trapped in position by the adjacent hardware.

3.2.9.7 MATING-RING MOUNTING

The method for mounting the mating ring shall provide for minimum distortion, static sealing, and antirotation.

Use free-floating nonloaded mating rings to minimize distortion caused by clamping loads. The shaft mating surfaces for clamped rings should be flat (3 helium light bands) and normal to the direction of applied load. The mating ring should be isolated from all structural loads.

Use gaskets, dispersion coating, plating, or elastomers to seal the joints between the mating ring and shaft or spacers exposed to pressure.

Use antirotation pins, radial splines, or clamping friction to prevent mating-ring rotation relative to the shaft.

3.3 SEAL COMPONENTS

3.3.1 Seal Materials

3.3.1.1 COMPATIBILITY

Seal materials shall not be adversely affected by the sealed fluid or by the operating temperature.

The seal materials based on current practices given in table III are recommended. Additional material recommendations are given in references 17 through 20.

For structural components, use metals that maintain adequate ductility at cryogenic temperatures (e.g., Monel, nickel-base alloys, and stainless steels). Avoid using the martensitic steels (e.g., 17-7 PH, AM-350, 4130, 4340) for flexing elements at low temperature because they become brittle.

Avoid metal-to-metal rubbing contact in liquid oxidizers because of the explosion hazard.

Metals resistant to impact detonation and oxidation should be used for seal-ring pilots, antirotation tangs, and friction-type vibration dampers.

Structural metals subject to surface reaction with the fluid should be protected with chromium or cadmium.

Fasteners that are subject to thread galling should be plated with copper, silver, or cadmium for applications in the temperature range of -423 to 1000°F.

Static seals and gaskets should be plated with copper or silver for improved sealing at temperatures from -423 to 1000°F.

Gold or copper plating is recommended for seals used in liquid fluorine.

For rubbing-contact seals, use carbon (P5N, P692, P03N, P5AG, EY105, CDJ83, CCA72, G84, G39) seal faces rubbing against hard-chrome-plated steel and tungsten/chromium carbide-coated mating rings. Avoid using carbonaceous materials in oxidizing atmospheres at temperatures higher than approximately 1000°F because of oxidation and chemical erosion. Special high-temperature carbons (e.g., P2003, G84, CDJ83) should be used when temperatures exceed 800°F.

Plastic materials (e.g., Kel-F, Teflon, Mylar) should be used for gaskets, static seals, and secondary seals in most cryogenic or reactive propellants (except fluorine). Avoid using plastic materials at temperatures below –320° or above 600°F. Plastic materials should not be used for stressed components above approximately 200°F because plastics are subject to cold flow or creep.

Avoid using plastic materials in liquid oxygen when impact energy levels exceed 70 ft-lbf or when pressures are extremely high ($\geqslant$ 5000 psi). Nonmetallic materials intended for use in oxygen should be tested in accordance with test procedures outlined in reference 77.

Elastomeric materials (Viton A, Buna N) should not be used at temperatures below approximately –65°F or above approximately 500°F. Use Viton A for resistance to deterioration by ozone.

Epoxy adhesive Epon 901/B3 is recommended for bonding carbon inserts to be used in liquid oxygen. Avoid using epoxy adhesive if the material will be exposed to impact energy levels in excess of 2 kg-m (14.46 ft-lbf). Do not use epoxy in liquid fluorine.

Use ceramic (aluminum oxide, titanium carbide, tungsten carbide) and cermet materials for maximum oxidation resistance at high temperature (up to 2000°F). The ceramic and cermet materials are recommended for use with liquid fluorine and other highly reactive propellants.

3.3.1.2 CORROSION

Seal materials shall not be subject to corrosion that could adversely affect performance.

Use corrosion-resistant stainless steels and nickel-base alloys for seal construction. Protect ferrous metals (e.g., 4130 and 4340) with chromium or cadmium plating.

Use nickel-base alloys for metallic elements exposed to hydrogen-rich steam (H_2 + H_2O).

3.3.1.3 Hydrogen Embrittlement

Seal structural components in hydrogen environments shall be resistant to hydrogen embrittlement.

The stable stainless steels (e.g., 300 series and A-286) or aluminum alloys should be used for hydrogen environments above −200°F. The nickel-base alloys should be used at temperatures below −200°F. Materials subject to hydrogen embrittlement should be protected with copper plating (refs. 21 and 22).

3.3.1.4 MATERIAL PROPERTIES AND TEMPERATURE EFFECTS

Seal material properties and their variation with temperature shall be suitable for the application.

Select the seal materials on the basis of the requirements established by a heat-transfer and stress analysis and the material properties given in references 23 and 24. Consider the variation of material physical properties with temperature (figs. 33, 34, and 35).

3.3.1.5 WEAR AND FRICTION

Rubbing-contact seal materials to be used in cryogenic or dry environments shall form a self-lubricating film when exposed to the fluid medium.

Use carbonaceous materials with inorganic resins, metal fluorides, silver, and various other additives to assist in the formation of a self-lubricating film on the mating surface.

Use materials containing titanium carbide or aluminum oxide to form a lubricating fluoride film in liquid fluorine.

Fused fluoride coatings should be applied to the mating surface of ceramic and cermet materials to reduce the wear and friction in liquid oxygen, liquid fluorine, liquid sodium, hydrogen, vacuum, and air (refs. 25 and 26).

Additional recommendations for seal rubbing materials based on current practices are given in table III.

3.3.1.6 CLEANING

Parts exposed to liquid oxidizers shall be free of all hydrocarbon contamination.

Metallic parts should be cleaned by (1) brushing in trichloroethylene, (2) hot vapor degreasing, or (3) ultrasonic cleaning (ref. 19). Non-metallic materials (plastics, elastomers, and carbons) should be cleaned by hand wiping with a lint-free cloth moistened with liquid Freon. Avoid flushing with cleaning solvents. All parts should be cleaned as individual components prior to assembly. Protect the parts after cleaning by sealing in a plastic bag until ready for assembly.

Oxidizer seals should be designed to allow cleaning of the separate components. Avoid trapped areas which could prevent adequate cleaning.

Carbon materials for oxidizer service should be machined and lapped either dry or in a compatible fluid. Avoid using diamond or silicon lapping compounds suspended in mineral oil bases, because of the difficulty of cleaning soft or porous seal materials.

3.3.1.7 PASSIVATION

Materials exposed to liquid fluorine shall be passive in contact with fluorine.

Build up a protective fluoride film and react surface contamination by passivating the material in gaseous fluorine prior to liquid exposure. Avoid exposure to moisture after passivation (ref. 19).

3.3.2 Face-Contact-Seal Rubbing Elements

3.3.2.1 SEAL FACE WIDTH

The seal face width shall provide maximum sealing effectiveness consistent with wear-life and heat-transfer requirements.

The seal face width should be a minimum of 0.030 in. to provide sufficient margin for machining tolerances and edge chipping. The minimum width for larger seals increases as a result of the structural load and unit contact load requirements. The 0.030-in. width may be used for larger seals if pressure-vented support pads (fig. 18(b)) are used on the seal face to assist in load support.

Use the minimum practical face width for precise pressure balance and maximum heat transfer to the sealed fluid.

The maximum face width should be established by the load variation due to the interface pressure profile or by the effect of face distortion on leakage and the pressure profile. The theoretical leakage is reduced in direct proportion to the face-width increase; however, the adverse effect of additional face distortion usually is greater on turbopump seals.

Use the maximum practical face width to reduce the unit contact load for minimum wear and to achieve maximum heat transfer by conduction into the seal ring and mating ring.

The recommended face width in relation to face diameter for turbopump face-contact seals is given in figure 37.

3.3.2.2 SEAL NOSE HEIGHT

The seal nose height shall allow for the maximum face wear and shall be consistent with the structural requirements.

Establish the minimum nose height (fig. 36(a)) on the basis of predicted maximum wear rate and life requirements.

Establish the maximum nose height on the basis of (1) the structural limitations required to prevent failure at the junction of the nose and seal-ring base and (2) the potential face distortion caused by thermal and pressure stresses.

The nose height on turbopump seals should be from 0.030 to 0.060 in. The height-to-width ratio should be approximately 0.3 to 0.8.

Use a minimum fillet radius of 0.020 to 0.030 in. at the junction of the nose and seal-ring base on brittle materials (carbon).

Allow a minimum of 0.005 in. carbon base above the metal carrier (fig. 36(b)) on insert designs for oxidizer service.

3.3.2.3 SEAL-RING ANTIROTATION DEVICE

The antirotation device shall prevent the seal ring from rotating but shall not restrict axial movement.

Use two or more tangs or blocks on the stationary housing that engage slots on the seal ring flange or radial splines. The tang contact area should be sufficient to prevent indentations in the slots from the impact loads. Minimize the tang-to-slot clearance. The tang material should be softer than the slot material.

Avoid using antirotation pins because they result in a higher contact load and tend to wear indentations in the slot.

3.3.2.4 SEAL-RING PILOT

The seal-ring pilot shall maintain precise radial location of the seal and allow the required face angular misalignment.

Locate the seal-ring pilot (fig. 12) at the secondary seal and maintain a length-to-diameter ratio of approximately 0.05 to 0.1. The pilot diametral clearance should generally be about 0.003 in./in. of diameter at operating conditions. Adjust the ambient clearance to provide for thermal differentials and pressure deflections. Radial splines may also be used to maintain radial alignment and compensate for thermal differentials. When necessary, use hard-chrome plating on metallic seal rings at the pilot diameter to eliminate fretting damage.

3.3.2.5 SEAL INSERT RETENTION

The seal-ring insert shall be retained at operating conditions.

Seal-ring inserts should be retained in the carrier with an interference fit at the OD of the insert and with adhesive bonding applied to the bottom of the insert (fig. 36(b)). The diametral interference for carbon inserts at operating conditions should be a minimum of approximately 0.001 in./in. of diameter. Use special steels with low expansion and contraction rates (e.g., Invar 36, Carpenter 42, molybdenum steel) to minimize the interference change with temperature.

The epoxy adhesives recommended for bonding are listed in table III. Avoid using adhesive bonding as the only retention method when the insert temperature may exceed approximately 300°F.

3.3.2.6 INSERT/CARRIER SEPARATION

The interference fit on inserts with inside pressure shall be sufficient to ensure that the retention force is greater than the separating force.

Establish the required interference on the basis of the relative values of the pressure separating force between the insert and carrier and the retention force caused by friction at the insert OD. The friction force is determined by the unit contact load at the insert OD required to deflect the insert radially inward by the amount of the interference. The insert rigidity and the coefficient of friction determine the necessary interference. The effects of thermal differentials and pressure deflections on the interference must be considered.

Consider mechanical locks for insert retention when it is not feasible to provide sufficient retention force with an interference fit. The mechanical lock should be spring loaded against the insert with a load greater than the separating force to prevent slight movements that can cause seal-face distortion.

Use epoxy adhesives to exclude the high-pressure fluid from the insert-to-carrier joint.

The recommended method for reducing the pressure separating force for seals with higher pressure at the inside diameter is shown in figure 38(b).

3.3.2.7 INSERT DISTORTION

Inserts retained in a metal carrier with an interference fit shall not be distorted excessively when chilled to cryogenic temperature.

To minimize the interference change with temperature, use the special steels (Invar 36, Carpenter 42, molybdenum steel) with a thermal-contraction rate similar to that of the insert material.

Align the insert and carrier cross-section centroids (fig. 39(b)) as closely as possible to minimize the bending moment. The bending force should be minimized by reducing the cross-sectional area of the insert.

3.3.2.8 LAPPED-JOINT SEAL-RING/CARRIER ASSEMBLY

The lapped joint shall (1) allow relative movement between the seal ring and carrier plate to minimize distortion and (2) maintain sufficient closing force to effect a static seal.

The seal ring and carrier plate surfaces should be lapped flat (3 helium light bands) and relieved to provide a resultant pressure closing force (fig. 40). The relieved surface should be provided with bearing support pads, except where distortion can cause the bearing support pads to separate the seal-ring static seal.

3.3.2.9 SPRAY-COATED SEAL RINGS

The spray coating shall maintain a bond to the seal-ring base adequate to prevent chipping and flaking.

The plasma-spray process is recommended for aluminum oxide coatings. The thermally induced stress should be minimized by using a thin (0.010 to 0.020 in.) coating of aluminum oxide sprayed on a transition layer of Nichrome. Use pure Nichrome on the base metal, then a mixture of 50-percent Nichrome/50-percent aluminum oxide, and finally pure aluminum oxide.

Use a raised tapered structure similar to that shown in figure 36(d) for the coating base. The coating should be applied to both sides of the seal ring to equalize the thermal load and minimize the thermal distortion.

3.3.3 Circumferential-Seal Rubbing Elements

3.3.3.1 SEGMENTED SHAFT SEALS

3.3.3.1.1 Segment Loading

The radial and axial loading on a segment shall be sufficient for effective sealing without exceeding the load-speed-life limitations for specific seal materials and sealed fluids.

To maintain shaft contact, the segments should be loaded in the radial direction with a garter spring (fig. 20). The garter-spring load should be approximately 0.1 to 0.2 lbf/in. of circumference.

Axial loading should be maintained with a wave spring (fig. 20) providing approximately 0.5 to 1.0 lbf/in. of circumference. The garter-spring load should be higher than the friction force caused by the wave spring.

The differential axial and radial pressure forces should be partially balanced by relieving the contact surfaces (fig. 41). The resultant shaft contact load should be consistent with the load-speed-life limitations.

3.3.3.1.2 Segment Joints

The segment joints shall provide for static sealing.

The segment joints should be overlapped with backup segmented rings and lapped flat or machined to the same radius for conformance (fig. 20).

3.3.3.1.3 Segment Antirotation Device

The antirotation device shall prevent the segments from rotating but shall not restrict the required radial movement.

Use antirotation pins at the segment gaps to prevent rotation. Pin blocks (fig. 21) may be used to distribute the load over a larger contact area. Provide sufficient clearance between the pins and segments to allow the required radial movement. Consider dimensional tolerances and thermal differentials.

3.3.3.1.4 Segment Diameter

> *The segment diameter shall conform to the shaft diameter at operating conditions.*

The segment design diameter should be established with consideration for the thermal differential so that the segment has the same radius as the shaft under operating conditions.

3.3.4 Hydrostatic/Hydrodynamic Face Seal Elements

3.3.4.1 SELF-ENERGIZED HYDROSTATIC SEALS

> *The face geometry shall provide a self-energized hydrostatic force that is balanced with the pressure and spring closing forces at the design clearance.*

Use the recessed pads with orifice compensation or the recessed step or the convergent tapered face (fig. 15), the choice depending on the specific design requirements.

The recessed-pad design should utilize a minimum of three separate pads around the seal face to provide for face misalignment stability. The volume of the recessed pads should be minimized to prevent dynamic instabilities caused by slow response to pressure changes. The orifice size should be large enough to prevent clogging from contaminants in the sealed fluid.

The depth of the recessed step should be determined by calculating the required pressure drop across the recessed area to provide the restoring force necessary to maintain the design face clearance.

The convergent-taper design also should be established by the relative pressure drop across the tapered area. The average pressure force on the seal face should be equal to the pressure and spring closing forces at the design clearance.

Conduct a thorough stress and thermal analysis so that the seal face and mating ring distortions can be minimized (refs. 6 and 7). Conditions that result in a divergent interface surface must be avoided because the restoring force is inherently unstable.

Reference 35 is recommended for a discussion of the detail design procedures.

3.3.4.2 EXTERNALLY PRESSURIZED HYDROSTATIC SEALS

The interface hydrostatic force resulting from the external pressurization shall be equal to the pressure and spring closing forces at the design clearance.

Utilize a minimum of three separate pads around the seal face to provide for face misalignment stability. The volume of the recessed pads should be minimized to prevent dynamic instabilities. The orifice size should be established to provide the necessary pressure drop at the required purge flow to maintain the design clearance.

The required orifice pressure drop should be determined in conjunction with the flow analysis of the face/pad region and the supply system.

Detail design procedures are discussed in references 35 and 46.

3.3.4.3 HYDRODYNAMIC SEALS

The hydrodynamic lift force shall be sufficient to balance the net pressure and spring closing forces at the design clearance.

The face geometry should be established with consideration for the rotational speed and fluid viscosity. The secondary-seal diameter or the bellows effective diameter and the spring load should be established to prevent the closing force from exceeding the available hydrodynamic lift force. The hydrodynamic design should provide that the seal be pressure balanced to obtain the desired operating film thickness at the seal interface (ref. 48).

Use materials that are dimensionally stable and suitable for intermittent rubbing contact in the fluid medium to be encountered.

Conduct a thorough stress and thermal anlaysis so that distortions of the seal face and mating ring can be minimized (refs. 6 and 7).

Detail design procedures for various applications are discussed in references 31 through 53.

3.3.4.4 HYBRID SEALS

The hydrostatic action shall provide the lift force required to maintain the design clearance at low rotational speeds, and the hydrodynamic action shall provide additional lift and dynamic stability at high speeds.

The design procedures for the hydrostatic and hydrodynamic concepts should be integrated to optimize the hybrid-design configuration. Design methods are discussed in reference 45.

3.3.5 Circumferential-Seal Clearance Elements

3.3.5.1 LABYRINTH SEALS

3.3.5.1.1 Geometry

The labyrinth geometry shall provide maximum sealing effectiveness.

The labyrinth clearance should be a minimum consistent with the installed and operating tolerances. Step or staggered labyrinths should be used where possible. The teeth tips should be sharp (0.005 to 0.015 in. R). The number of teeth should be maximized (fig. 44) consistent with the optimum tooth pitch of approximately 0.1 in. per 0.01 in. diametral clearance (fig. 45) for straight labyrinths. The cavity depth should be approximately equal to the tooth pitch.

Detail design procedures for various applications are discussed in references 54 through 65.

3.3.5.1.2 Wear-In Labyrinths

Wear-in labyrinth materials shall be compatible with the fluid environment and shall be easily deformable or worn away at initial contact.

Metal honeycomb or sintered-metal-fiber abradable materials should be used in nonoxidizing fluids. The honeycomb foil thickness and cell size should be based on a tradeoff of ease of deformation and erosion resistance.

The thickness generally should be 0.002 to 0.005 in., the cell width approximately 1/16 in., and the cell depth one to two times the width. Cells are oriented normal to the direction of rotation. The honeycomb strip should be brazed to a support ring for attachment. Abradable materials should not be used in applications where the wear debris will contaminate the system. Carbonaceous materials may be used if the interference is not too large.

Kel-F is the recommended material for wear-in labyrinths in liquid oxygen.

3.3.5.1.3 Plastic Labyrinths

The housing for plastic labyrinths shall provide structural support adequate to maintain dimensional stability.

The plastic material should be restrained from its natural contraction rate by locking relatively thin sections into a metal housing (fig. 47). The housing should be strong enough to control the thermal contraction and provide dimensional stability. A static seal flange and pressure vents should be provided on high-pressure seals. The length of the plastic span between retention locks should be approximately 0.5 to 1.0 in.

3.3.5.1.4 Segmented Labyrinths

Segmented labyrinths shall provide for radial movement with positive stops to control the wear-in diameter.

Segmented labyrinths should be spring loaded to allow wear-in to a diameter smaller than the rotor. The segment diameter after wear-in should be controlled by limiting the radially inward travel with a T-slot arrangement (fig. 48) or by allowing the segments to become arch-bound. A lapped joint should be provided between the segments and housing for a static seal. Antirotation devices should be provided to prevent segment rotation. High-pressure seals should be partially pressure balanced to prevent excessive contact loads during wear-in.

3.3.5.1.5 Erosion Resistance of Labyrinth Materials

Labyrinth materials shall be resistant to surface erosion caused by cavitation or impingement of high-energy fluid.

The hard high-strength materials (tool steel, Stellite, maraging steels) are recommended for labyrinths that may be exposed to severe cavitation or to high-energy-fluid impingement. The soft low-strength materials (aluminum, Monel, brass, bronze, plastic) should not be used when surface erosion may be a problem.

3.3.5.2 FLOATING-RING SEALS

3.3.5.2.1 Clearance

The floating-ring element shall maintain a controlled clearance to the shaft under all extremes of operating conditions.

Use an outer-ring material that has the same thermal expansion and contraction rate as the shaft material, so that a constant clearance gap is maintained as the temperature changes. The ring radial section should be sized to provide sufficient rigidity to minimize the radial deflection due to the unbalanced pressure while minimizing the seal ring inertia to achieve maximum response to eccentric shaft rotation. The seal ring-to-shaft operating clearance should provide adequate leakage control and sufficient margin to allow for the thermal and pressure deflections; an operating diametral clearance of 0.0005 to 0.001 in./in. of diameter is recommended.

3.3.5.2.2 Wear

The inner ring and shaft surface shall withstand the wear resulting from the intermittent contact required to center the ring.

The inner ring materials should be selected for minimum friction and wear. The material should be compatible with the sealed fluid and self-lubricating if the fluid is not a lubricant. Carbonaceous materials are recommended for most applications. The shaft surface should be plated with a hard wear-resistant material (hard-chrome plate, chromium carbide, or tungsten carbide). The recommended seal materials based on current practices are given in table III.

3.3.5.2.3 Axial Loading

The floating-ring axial load shall be adequate for static sealing but shall not restrict radial movement.

The differential axial pressure force (fig. 49) on high-pressure seals should be partially balanced by relieving the axial contact surface and minimizing the housing-to-shaft clearance. Use a wave spring on low-pressure seals to maintain axial contact.

3.3.5.2.4 Antirotation Device

The antirotation device shall restrain the floating ring from rotating without restricting its free radial or axial movement.

The outer steel ring should have two or more machined tangs that engage slots in the housing to prevent rotation.

3.3.5.3 ARCH-BOUND SEGMENTED SEALS

The segments shall become arch-bound after wear-in at the operating conditions to form a solid floating ring with minimum shaft contact load.

Adjust the segment design diameter to compensate for the deflection caused by the radial pressure load and the thermal contraction or expansion differential. The diameter should be approximately 0.001 in. smaller than the shaft at operating conditions to allow wear-in. The recommended materials based on current practices are given in table III.

3.3.6 Face-Contact-Seal Secondary Elements

3.3.6.1 METAL BELLOWS

3.3.6.1.1 Function

The bellows element shall act as the secondary seal and provide structural support for the seal ring.

The bellows element should be leak proof and allow for motion in the axial direction. It should be sufficiently rigid in the torsional and lateral directions to act as the antirotation and piloting device. Mechanical stops should be used to establish the free length and prevent overcompression.

3.3.6.1.2 Spring Force

The bellows elements shall provide the spring force necessary to maintain face contact.

The bellows type, diameter, plate thickness, span, convolutions, and number of plies should be consistent with the space available and with the stress requirements, so that the desired spring force and compression range are provided. The bellows stress and spring rate should be estimated with the use of available analytical techniques (refs. 69 through 71) and the supplier's empirical data for specific configurations.

Minimize the spring rate by using the maximum number of convolutions that can be fitted into the available space with the convolution pitch necessary to provide the required compression range and the plate thickness necessary to satisfy the pressure and deflection stress requirements. Double-ply bellows may be used to increase the pressure capacity without significantly increasing the spring rate.

3.3.6.1.3 Pressure Capacity

The bellows element shall withstand the maximum differential pressure without significant permanent deformation.

The bellows pressure capacity should be estimated with the available analytical techniques (refs. 69 through 71) and the supplier's empirical data for specific configurations. The bellows should be stabilized with a proof pressurization at the minimum operating compression followed by a stress-relief heat treatment. The change in free length as a result of pressurization should not exceed ±0.005 in. after the bellows have been stabilized.

3.3.6.1.4 Effective Diameter

The bellows effective diameter shall be sufficiently constant to allow adequate pressure balance.

When accurate pressure balance is required, the bellows should have minimum variation of effective diameter with pressure and pitch. Thicker plates, smaller span, and a larger pitch should be used to minimize the effective diameter change.

The bellows should be calibrated to measure the effective diameter or total load (spring plus pressure) at the limits of operating pressure and compression before the seal face diameters are established for force balance.

3.3.6.2 PLASTIC LIP SEALS

3.3.6.2.1 Loading

The lip-seal elements shall provide for pressure and spring loading of the lip.

Establish the lip-seal length and housing support geometry (fig. 53) to provide a differential pressure force that will increase the seal seating stress at higher pressure. The seal should be spring loaded against the housing with a flat retainer ring to compensate for the thermal contraction.

3.3.6.2.2 Geometry

The lip contour and thickness shall provide conformance to the seal ring and flexibility for misalignment.

Dimension the lip-seal radius and housing support radius to provide a smooth transition with the maximum lip support between the housing and seal ring. The lip-seal thickness should be approximately 0.005 to 0.010 in. to maintain flexibility for conformance and to minimize the thermal loads.

3.3.6.2.3 Interference

The lip seal shall maintain a diametral interference with the seal ring at extreme operating conditions.

Dimension the lip-seal diameter to provide a diametral interference of approximately 0.005 in./in. of diameter at room temperature. The materials should be selected to allow the interference to increase at operating temperature. Size the seal ring-to-housing clearance and the lip length to maintain an interference under the worst conditions of misalignment.

3.3.6.3 ELASTOMERS

3.3.6.3.1 Squeeze

The groove dimensions shall provide squeeze adequate to maintain sealing contact and prevent excessive friction at extreme conditions.

Control the groove dimensions with precision tolerances. A tolerance stackup to establish the dimensional limits should be performed. The effects of thermal contraction or expansion and shrinkage or swelling should be considered to establish the squeeze. An operating squeeze of 5 to 10 percent is recommended for secondary elastomer seals.

Detail design requirements are given in references 72 and 73.

3.3.6.3.2 Sealing Surface

The elastomer sealing surface shall minimize friction and wear.

The sealing surface should be either hardened steel or hard-chrome plating, finished to 5 to 10 μin. rms. The metal surface may also be coated with 0.001- to 0.003-in.-thick Teflon to minimize friction and wear.

3.3.6.3.3 Lubrication

The lubrication shall be sufficient to prevent rolling, twisting, or sticking of the elastomer.

The elastomer and all metal parts in contact with the elastomer should be lubricated with a compatible lubricant (ref. 73). Silicone oil or grease is recommended for Buna N and Viton; do not, however, use Buna N or silicone oil or grease in liquid or gaseous oxygen.

3.3.6.4 PISTON RINGS

The ring and cylinder sealing surfaces shall provide line-to-line contact with minimum friction and wear.

The piston ring sliding surface should be hardened steel or hard chrome plate, finished to 8 to 16μ in. rms, and round within 0.0003 in./in. of diameter. The cylindrical taper of the sealing diameter should not exceed 0.0003 in./in. of length, and the squareness of the diameter should not exceed 0.0005 in./in. of groove depth (ref. 74) at all operating conditions. The effects of axial thermal gradients on the angular displacement of the groove seal surface should be considered.

3.3.7 Spring Load

The spring load shall provide effective sealing and shall be consistent with the load-speed-life relations.

Establish the minimum load for effective sealing by calculating the force required to accelerate the seal ring at the rate required to track the rotating mating surface runout or to follow sudden shaft movements. The resultant closing force should be greater than the sum of the seal-ring inertia, secondary friction, and interface pressure profile forces.

Establish the maximum load by the load-speed-life relationships for specific fluids and materials or by the seal power-loss limitations. Calculate the allowable heat input by heat-balance analysis.

The recommended spring loads per unit circumferential length of seal face are as follows:

Sealed fluid	Recommended spring load, lbf/in.
Liquid oxygen	2 to 4
Liquid hydrogen	1 to 2
Gaseous hydrogen	1 to 2

(continued)

Hot gas (H_2 + H_2O)	1 to 2
RP-1	1 to 4
Liquid fluorine	2 to 3

3.3.8 Pressure Balance

The pressure balance shall prevent excessive face load at high pressure and provide adequate margin for variation in the interface pressure profile.

Seals with differential pressures in excess of 100 psi should be pressure balanced. The pressure-balance ratio should be selected with consideration for the maximum variation of interface pressure profile and the total (spring plus pressure) face-load requirements. The following pressure-balance ratios are recommended (table IV):

Table IV. – Recommended Pressure-Balance Ratios for Turbopump Rotating-Shaft Seals

Sealed fluid condition	Seal requirement	Recommended balance ratio
High-pressure incompressible	Minimum face load	0.55 to 0.6
Low-pressure incompressible (conventional designs)	Minimum load and adequate margin for interface pressure variation	0.6 to 0.7
High-pressure cryogenic*	Minimum face load	0.65 to 0.7
Low-pressure cryogenic*	Maximum sealing effectiveness	0.8 to 0.9
High-pressure, sonic (choked) flow, compressible	Minimum face load	0.67 to 0.75 as determined by analysis (ref. 12)
Low-pressure, subsonic laminar flow, compressible	Minimum load and adequate margin	0.6 to 0.7 as determined by the following equation (eq. (14)) plus required design margin: $\alpha \approx \frac{1}{3}\left[1 + \frac{P_1/P_2}{1+(P_1/P_2)}\right]$

*Listed as a separate category because fluid can change from liquid to vapor as it crosses seal interface.

APPENDIX A

Conversion of U.S. Customary Units to SI Units

Physical quantity	U.S. customary unit	SI unit	Conversion factor*
Angle	deg	rad	1.745×10^{-2}
Angular velocity	rad/sec	rad/sec	1.
Area	in.^{2}	cm^2	6.452
Density	lbm/in.^{3}	kg/m^3	2.768×10^{4}
Energy	Btu	J	1.054×10^{3}
Force	lbf	N	4.448
FV factor	lbf/in. × ft/sec	N/cm × m/sec	5.338×10^{-1}
Gas constant (specific)	in.-lbf/(lbm-°R)	J/(kg-K)	4.483×10^{-1}
Heat generation rate	Btu/sec	J/sec	1.054×10^{3}
Heat generation rate per unit length	Btu/(sec-in.)	J/(sec-cm)	4.150×10^{2}
Heat generation rate per unit area	Btu/(sec-in.^{2})	J/(sec-cm^2)	1.634×10^{2}
Impact energy level	ft-lbf	J	1.356
Leakage rate	gpm (gal/min)	cm^3/min	3.785×10^{3}
	scfm (std cu ft/min)	cm^3/min	2.832×10^{4}
Length	ft	m	3.048×10^{-1}
	in.	cm	2.54
	μin.	μm	2.54×10^{-2}
Load/unit length	lbf/in.	N/cm	1.751

(continued)

Conversion of U.S. Customary Units to SI Units (concluded)

Physical quantity	U.S. customary unit	SI unit	Conversion factor*
Mass	lbm	kg	4.536×10^{-1}
Mass density	lbm-sec^2/in.^{4}	kg-sec^2/cm^4	1.090×10^{-2}
Modulus of elasticity	psi (lbf/in.^{2})	N/cm^2	6.895×10^{-1}
Pressure	psi	N/cm^2	6.895×10^{-1}
PV factor	lbf/in.^{2} X ft/sec	N/cm^2 X m/sec	2.102×10^{-1}
Rotational speed	rev/sec	rad/sec	6.283
	rpm (rev/min)	rad/sec	1.047×10^{-1}
Spring rate	lbf/in.	N/cm	1.751
Surface speed (rubbing speed)	ft/sec (fps)	m/sec	3.048×10^{-1}
Temperature	°F	K	$K = \frac{5}{9}(°F + 459.67)$
	°R	K	$K = \frac{5}{9}(°R)$
Unit load	psi	N/cm^2	6.895×10^{-1}
Viscosity, absolute	lbf-sec/in.^{2}	N-sec/cm^2	6.895×10^{-1}
Viscosity, dynamic	lbm/(in.-sec)	N-sec/cm^2	1.786×10^{-3}
	lbm/(ft-hr)	N-sec/cm^2	4.134×10^{-8}
Viscosity, kinematic	in.^{2}/sec	cm^2/sec	6.452
Wear rate	in./hr	mm/hr	2.54×10^{1}
Yield strength	psi	N/cm^2	6.895×10^{-1}
	ksi (1000 psi)	N/cm^2	6.895×10^{2}
Young's modulus	psi	N/cm^2	6.895×10^{-1}

*Except for temperature conversions, which are to be made as shown, multiply value given in U.S. customary unit by conversion factor to obtain equivalent value in SI unit. For a complete listing of conversion factors for basic physical quantities, see Mechtly, E. A.: The International System of Units. Physical Constants and Conversion Factors. Second Revision, NASA SP-7012, 1973.

APPENDIX B

GLOSSARY

Term* or Symbol	Definition
$\bar{b}$	seal face average circumferential width, in.
balance ratio	$\dfrac{\text{seal face effective closing area}}{\text{sealing face (dam) area}}$
C_f	constant for bellows configuration
$\bar{d}$	seal face average diameter, in.
E	Young's modulus, psi
F	resultant total contact load per unit circumferential length, lbf/in.
F_f	friction force, lbf
F_W	wave spring load, lbf
FV factor	total contact load per unit circumferential length times rubbing speed, lbf/in. X ft/sec
f	coefficient of friction
fps	feet per second
g	acceleration due to gravity, ft/sec^2
h	effective (average) leakage gap, in.
ID	inside diameter, in.
J	mechanical equivalent of heat, ft-lbf/Btu
K_a	axial spring rate, lbf/in.
L	seal face radial length, in.
MED (or M.E.D.)	mean effective diameter, in.
N	rotational speed, rev/sec

*As noted, seal terminology used in the preceding text basically is that presented in ASLE SP-1 (ref. 1), which should be reviewed for complete coverage of terms.

Term or Symbol	Definition
n_c	number of convolutions
n_p	number of plies
n_t	number of throttlings in a labyrinth seal
OD	outside diameter, in.
P	face contact unit load, psi
$\overline{P}$	average pressure, $(P_2 + P_1)/2$, psia
P_1	downstream pressure, psia
P_2	upstream pressure, psia
P_{r1}	pressure at inner radius r_1, psia
P_{r2}	pressure at outer radius r_2,psia
PV factor	total contact unit load times rubbing speed, lbf/in.^{2} $\times$ ft/sec
P_fV factor	fluid pressure times rubbing speed, lbf/in.^{2} $\times$ ft/sec
Q	volume flow, in.^{3}/sec
$\dot{q}$	heat generation rate per unit circumferential length, Btu/(sec-in.)
$\dot{q}'$	heat generation rate per unit contact area, Btu/(sec-in.^{2})
R	specific gas constant, in.-lbf/(lbm-°R)
$\overline{R}$	bellows mean radius, in.
Re	Reynolds number
R&D	research and development
$\overline{r}$	seal face average radius, in.
rms	root mean square
Δr	$r_2 - r_1$, in.

Term or Symbol	Definition
r_1	seal face inner radius, in.
r_2	seal face outer radius, in.
SSME	space shuttle main engine
s	half of bellows space (i.e.; $\frac{\text{radial OD} - \text{radial ID}}{2}$), in.
scfm	standard cubic feet per minute
T	absolute temperature, °R
T.I.R.	total indicated runout
t	bellows plate thickness, in.
total load	spring load + hydraulic load (based on linear profile), lbf
U	leakage flow velocity, in./sec
unit load	total contact load ÷ nose contact area, lbf/in.^{2}
V	surface speed, ft/sec
$\dot{w}$	weight flowrate, lbm/sec
α	pressure profile factor
λ	molecular mean free path at average pressure, in.
Δ	incremental change in a quantity, or difference between two quantities
μ	viscosity, lbm/in.-sec
μ_a	absolute viscosity, lbf-sec/in.^{2}
ν	kinematic viscosity, in.^{2}/sec
ρ	density, lbm/in.^{3}
ρ_m	mass density ρ/g, lbm-sec^2/in.^{4}
ϕ	flow coefficient

Term or Symbol	Definition
ω	angular velocity, rad/sec
ψ	leakage function, $\psi = \sqrt{\dfrac{1-(P_1/P_2)^2}{n_t + \ln (P_1/P_2)}}$

Material*	Identification
Aerozine 50 or A-50	50/50 blend of N_2H_4 and UDMH, propellant grade per MIL-P-27402
A-286	austenitic heat-resistant iron-base alloy; same as AMS 5735
AM-350, AM-355	semi-austenitic heat-resistant precipitation-hardening stainless steels
AmCerMet 701-65	sintered Ni-Cr alloy filled with fluoride eutectic; manufactured by Astro Met Associates, Inc.
AMS 4530, AMS 4650	Cu-Be alloys
AMS 5646	wrought Ni-Cr corrosion- and heat-resistant steel
AMS 5665	nickel-base Cr-Fe alloy; same as Inconel 600
AMS 5735	austenitic iron-base alloy containing 15 Cr, 26 Ni, and 1.3 Mo; same as A-286
Bearium B-10	trade designation of Bearium Metals Corp. (Rochester, NY) for leaded bronze
Buna N	trade designation for copolymer of butadiene and acrylonitrile
Carpenter 42	trade designation for low-expansion steel alloy (42% Ni, balance Fe) manufactured by Carpenter Steel Div., Carpenter Technology Corp. (Reading, PA)
CCA72, CCA-72	trade designation for carbon (graphite) manufactured by Union Carbide Corp.
CDJ83	trade designation for carbon (graphite) manufactured by Union Carbide Corp.
cermet	material composed of a ceramic with a metal binder; e.g., tungsten carbide with a cobalt binder

*Additional information on metallic materials herein can be found in the 1972 SAE Handbook, SAE, Two Pennsylvania Plaza, New York, NY; in MIL-HDBK-5B, Metallic Materials and Elements for Aerospace Vehicle Structures, Dept. of Defense, Washington, DC, Sept. 1971; and in Metals Handbook (8th ed.), Vol. 1: Properties and Selection of Metals, Am. Society for Metals (Metals Park, OH), 1961.

Material	Identification
chrome	chromium
Coor's Al_2O_3	designation for a fused alumina of great hardness
CRES	corrosion-resistant steel
elastomer	polymeric material that at room temperature can be stretched to approximately twice its length and on release return quickly to its original length
Epon 422, Epon 901/B3	trade designations for epoxy adhesives manufactured by Shell Chemical Co.
epoxy	thermosetting resin widely utilized as a binder in the fabrication of glass-filament/resin composites and as an adhesive
EY-105	trade designation for carbon (graphite) manufactured by Morganite, Inc.
FLOX	mixture of 30% LF_2 and 70% LOX
GH_2	gaseous hydrogen
GN_2	gaseous nitrogen per MIL-P-27401
G39, G84, G84SC	trade designations for carbon (graphite) manufactured by U.S. Graphite
Hastelloy	trade designation for a series of high-temperature nickel-base alloys manufactured by Stellite Div., Cabot Corp.
helium	pressurant helium (He) per MIL-P-27407
HF	hydrofluoric acid
$H_2 + H_2O$	hydrogen-rich steam (combustion product of $LO_2 + LH_2$)
hot gas	combustion products or gaseous discharge from a heat exchanger; hot-gas temperature can reach 1300°F, depending on the process, in state-of-the-art propulsion systems
HT 424	trade designation for epoxy adhesive manufactured by American Cyanamid Co.
hydrazine	N_2H_4, propellant grade per MIL-P-26536

Material	Identification
Inconel 600, 718, X (now X-750)	trade designations of International Nickel Co. for Ni-Cr-Fe alloys
Inconel 5665	same as AMS 5665 and Inconel 600
Inconel 5667	nickel-base Cr-Fe alloy; same as Inconel X-750
Invar 36	trade designation of Carpenter Steel Division, Carpenter Technology Corporation, for a nickel-base alloy (36% Ni, balance Fe) with a low coefficient of expansion
Kel-F	trade designation of 3M Corp. for a high-molecular-weight polymer of chlorotrifluoroethylene
Kentanium K162B	trade designation of Kennametal, Inc. (Latrobe, PA) for titanium carbide bonded with a Ni-Mo alloy
K-Monel	trade designation of International Nickel Co. for a wrought age-hardenable alloy containing Ni, Cu, and Al
LF_2	liquid fluorine, propellant grade per MIL-P-27405
LH_2	liquid hydrogen (H_2), propellant grade per MIL-P-27201
LOX, LO_2	liquid oxygen (O_2), propellant grade per MIL-P-25508
LW1	trade designation for a flame-plating form of tungsten carbide manufactured by Linde Co.
LW5, LW-5	trade designation for a flame-plating form of chromium carbide manufactured by Linde Co.
maraging steel	martensite- and age-hardening Ni-Fe alloy
molybdenum steel	a type of tool steel whose hardness is enhanced by its molybdenum content
Monel	trade designation of International Nickel Co. for an alloy containing mainly Ni and Cu
Mylar	trade designation of E.I. duPont de Nemours and Co. for a polyethylene terephthalate film
N_2H_4	hydrazine, propellant grade per MIL-P-26536

Material	Identification
N_2O_4	nitrogen tetroxide (oxidizer), propellant grade per MIL-P-26539
Nichrome	trade designation of Driver-Harris Co. (Harrison, NJ) for a high-temperature corrosion-resistant Ni-Cr alloy
para-hydrogen	type of molecular hydrogen having an antiparallel nuclear spin; preferred over ortho-hydrogen for rocket fuels
plastic	high-molecular-weight material that while usually firm and hard in its finished state is at some stage in its manufacture soft enough to be formed into a desired shape by application of heat or pressure or both
P03N, P2003	trade designations for carbon (graphite) manufactured by Pure Carbon Co.
P5AG	trade designation of Pure Carbon Co. for carbon (graphite) impregnated with silver (Ag)
P5N	trade designation of Pure Carbon Co. for carbon (graphite) impregnated with metal-fluoride salt
P692	trade designation of Pure Carbon Co. for carbon (graphite) impregnated with resin
RP-1	kerosene-base high-energy hydrocarbon fuel, propellant grade per MIL-P-25576
Stellite	designation for a series of hard Co-W-Cr-C alloys manufactured by Stellite Div., Cabot Corp.
synthetic rubber	any of a group of man-made elastomers whose properties approximate one or more of the properties of natural rubber
T6, T73, T86, T351	designations for heat-treating and tempering processes for aluminum alloys
Teflon (TFE)	trade designation of E.I. duPont de Nemours and Co. for polymer of tetrafluoroethylene
tool steel	hard steel intended for use in tools; contains a high percentage of C or other hardness-imparting elements (e.g., Mo)
UDMH	unsymmetrical dimethylhydrazine, propellant grade per MIL-P-25604

Material	Identification
Viton A	trade designation of E.I. duPont de Nemours and Co. for copolymer of vinylidene fluoride and hexafluoropropylene
17-7PH	semi-austenitic precipitation-hardening stainless steel
300 series (e.g., 304, 310, 347)	series of austenitic stainless steels
400 series (e.g., 416, 440C)	series of martensitic and ferritic stainless steels
2014, 2024	wrought aluminum alloys with Cu as the principal alloying element
4130, 4340	high-strength, martensite-hardening, low-alloy steels
6061, 6151	wrought aluminum alloys with Mg and Si as the principal alloying elements
7075, 7079	wrought aluminum alloys with Zn as the principal alloying element

ORGANIZATION

Abbreviation	Identification
AEC	Atomic Energy Commission
AFML	Air Force Materials Laboratory
AFRPL	Air Force Rocket Propulsion Laboratory
AISI	American Iron & Steel Institute
AMS	Aerospace Materials Specification
ASLE	American Society of Lubrication Engineers
ASM	American Society for Metals
ASME	American Society of Mechanical Engineers
ASTM	American Society for Testing and Materials
DMIC	Defense Metals Information Center

Abbreviation	Identification
IIT	Illinois Institute of Technology
SAE	Society of Automotive Engineers
WADD	Wright Air Development Division
WPAFB	Wright Patterson Air Force Base

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SP-8048	Liquid Rocket Engine Turbopump Bearings, March 1971

SP-8101	Liquid Rocket Engine Turbopump Shafts and Couplings, September 1972
SP-8100	Liquid Rocket Engine Turbopump Gears, March 1974
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SP-8097	Liquid Rocket Valve Assemblies, November 1973
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SP-8119	Liquid Rocket Disconnects, Couplings, Fittings, Fixed Joints, and Seals, September 1976
SP-8123	Liquid Rocket Lines, Bellows, Flexible Hoses, and Filters, April 1977
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