

**NASA  
SPACE VEHICLE  
DESIGN CRITERIA  
(CHEMICAL PROPULSION)**

**NASA SP-8094**

**LIQUID ROCKET  
VALVE COMPONENTS**



**AUGUST 1973**

**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 on 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 Valve Components", was prepared under the direction of Howard W. Douglass, Chief, Design Criteria Office, Lewis Research Center; project management was by M. Murray Bailey. The monograph was written by H. J. Ellis and T. R. Spring 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, O. D. Goodman of Aerojet Liquid Rocket Company; D. K. Huzel of the Space Division of Rockwell International Corporation; T. M. Weathers of TRW Systems, TRW Inc.; and C. H. Kerrigan of the Lewis Research Center individually and collectively reviewed the monograph in detail.

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

August 1973



## 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 VALVE COMPONENTS

## 1. INTRODUCTION

The advent of the liquid rocket engine with requirements for valves of low weight, small size, high pressure, high flow, low leakage, high cycle life, and high reliability necessitated technological advances in the design of valve components. The discussion of these components will be better understood in the light of the following definitions:

A valve is a mechanical device by which the flow of fluid may be started, stopped, or regulated by a movable part that opens, shuts, or partially obstructs a passageway or port in a containing structure, the housing. A valving element is the movable part (e.g., a poppet or "butterfly" disk) that contacts or approaches a surface in the valve housing, the seat, to shut off or throttle flow. The shaft moves and positions the valving element. A seal is a member of either housing or valving element that limits leakage: a primary seal limits leakage from the upstream to downstream side of a valving unit; a secondary seal limits external leakage from the valve shaft or its equivalent. A valving unit is the combination of the valving element and seat or seal; in the case of a throttling valve, the valving unit is the combination of valving element and housing or outlet orifice that regulates flow.

As rocket engine work progressed, valve design technology advanced, but not without attendant problems. Unacceptable leakage has occurred because of contamination, fracture, wear, or overstress of the sealing surfaces; thermal expansion of seals installed by pressfitting; excessive thermal contraction of critical parts due to cryogenic propellant; attack of the seat by a propellant; failure of the seal retainer; excessive shaft deflection; careless assembly or handling; corrosion; and vibration. Failures to actuate have occurred because of differential thermal contractions; galling or binding of shafts; contamination in close clearances; change in pressure balance due to yielding of a soft seal; lubricant failure due to an incompatible propellant; inadequate housing rigidity; poor alignment of fixed and moving parts; dimensional changes in castings and forgings; and overloaded bearings. Housings have cracked because of stress corrosion, vibration loads, stress concentrations, and thermally induced stresses. Shafts have failed because of low-temperature brittleness, improper heat treatment, and inertial loads.

This monograph treats these and other problems as they were solved in the successful development of lightweight operational valves for liquid rocket systems. Practices and procedures for the design of valving elements, seats, seals, housings, shafts, shaft seals, and bearings are discussed in detail; general practices for cleaning and contamination prevention are summarized. The information on valving element and seat or seal interfaces is arranged by specific valve type (poppet, butterfly disk, ball, blade, sleeve, and cylindrical slide), because the detail design requirements for each type of valving element and its seat or seal vary widely. The balance of information is arranged by topic, because the detail design requirements for housings, bearings, shafts, and shaft seals and the problems of contamination and vibration apply to most types of valves. Specific design requirements arising from the particular use of a valve are discussed. Lapping techniques are treated briefly, because lapping is a critical manufacturing operation that is not always well understood and is not well documented.

Detailed discussions of the bases for selecting the best type of valve for specific rocket engine or vehicle application, as well as guidance for selection of the preferred actuator type and integration of the actuator with the valve, are presented in the design criteria monograph "Liquid Rocket Valve Assemblies" (ref. 1). Design of explosive-actuated valves is presented in the design criteria monograph "Liquid Rocket Pressure Regulators, Relief Valves, Check Valves, Burst Disks, and Explosive Valves" (ref. 2). Spring design is treated in the design criteria monograph "Liquid Rocket Actuators and Operators" (ref. 3).

This monograph on valve components is intended to be used together with the companion monograph, reference 1. A typical design sequence requires that the designer first consult reference 1 to select the type of valve assembly that best meets the system requirements and then use this monograph to guide the design of components to fulfill the assembly requirements. There is, of course, considerable cross-feed between these two design efforts.

## 2. STATE OF THE ART

Poppet (fig. 1), butterfly (fig. 2), and ball valves (fig. 3) are the types of valves most commonly used in liquid rocket systems. Blade valves (fig. 4) were used in the rocket engines for the Atlas<sup>1</sup> vehicle and continue to be a candidate type; they have not been selected for recent engines because of the seat wear problems associated with high loads on the soft seat. A linear sleeve valve (fig. 5) was used on the M-1 development engine, but current applications of this type of valve are limited because the valve is difficult to seal reliably. Rotary sleeve valves (fig. 6) are used in throttling applications. Cylindrical slide or spool valves (fig. 7) are widely used as servovalves and flow-routing valves, but other applications of this type of valve are limited because of the high leakage rates. Table I displays the chief features of valve components used in typical applications in operational vehicles.

Applications of the poppet valve to the liquid rocket engine brought about advances in the state of the art of control of geometry, surface finish, and alignment of the poppet and seat that contributed to decreased leakage and increased cycle life at high pressure. The designs evolved from the comparatively-high-leakage, conical-seat, poppet valve designs used in aircraft. The all-metal poppet valve is very reliable except that it is still susceptible to leakage caused by contamination. Advances have resulted also from use of plastics (primarily fluorocarbons like Kel-F and Teflon) as seal materials that are compatible with and perform well with either cryogenic or storable propellant systems. Initial use of Teflon as a seal material resulted in many instances of leakage until the evolution of designs that minimized detrimental seal yielding due to creep under load or to thermal relief of residual stresses.

The butterfly valve used on the present day liquid rocket engine evolved from application of the plastic disk seal and shaft seal to the low-pressure butterfly-type valve originally used in aircraft. This action ultimately resulted in the very reliable, lightweight, compact, low-pressure-drop butterfly valve used extensively for propellant valves on booster engines. Major development effort centered around determining the combination of lip seal and disk design that permitted the disk to seal reliably and yet not damage the seal as the disk closed rapidly under flow conditions that deflected the seal into the path of the disk. Another major factor in the successful effort was the determination of the optimum combination of valve disk and seal dimensions and materials that would eliminate leakage due to thermal contractions in cryogenic service.

Redundant ball valves are used in man-rated applications requiring ultra-high reliability. In some of these applications, redundant upstream and downstream seals for each ball are used also to increase valve reliability. Features of current ball valve designs include the use of

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<sup>1</sup>Terms, symbols, materials, engines, and vehicles are defined or identified in the Glossary.

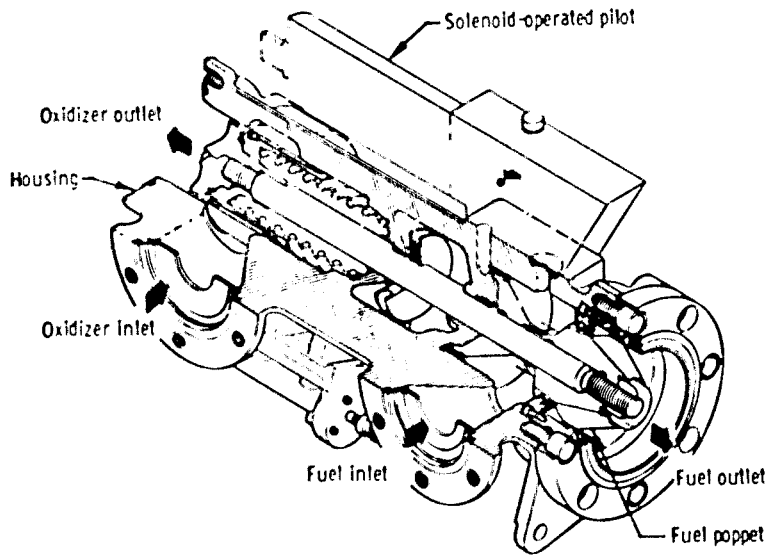


Figure 1. - Isometric cross section of poppet valve.

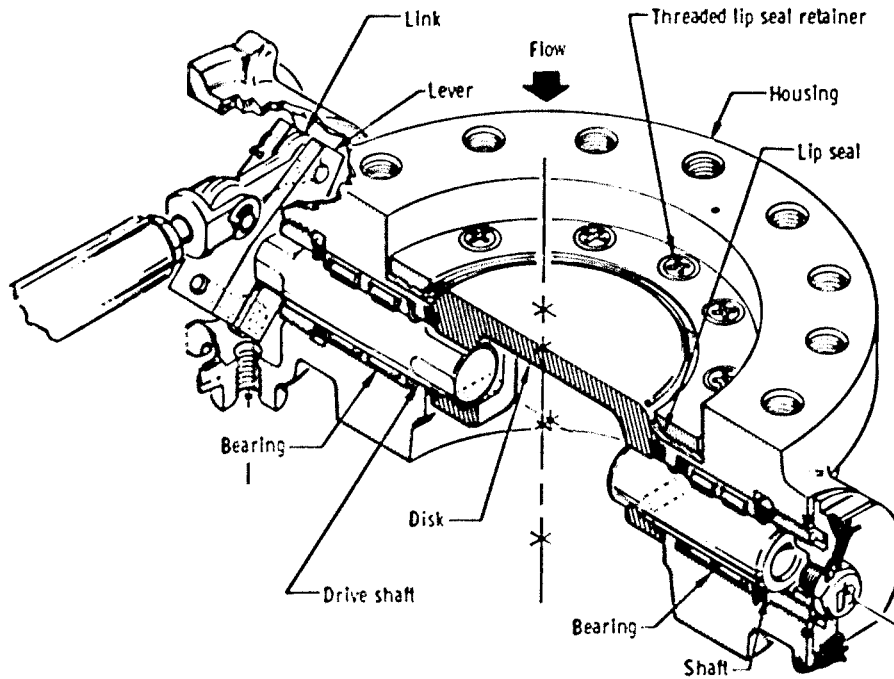


Figure 2. - Isometric cross section of butterfly valve.

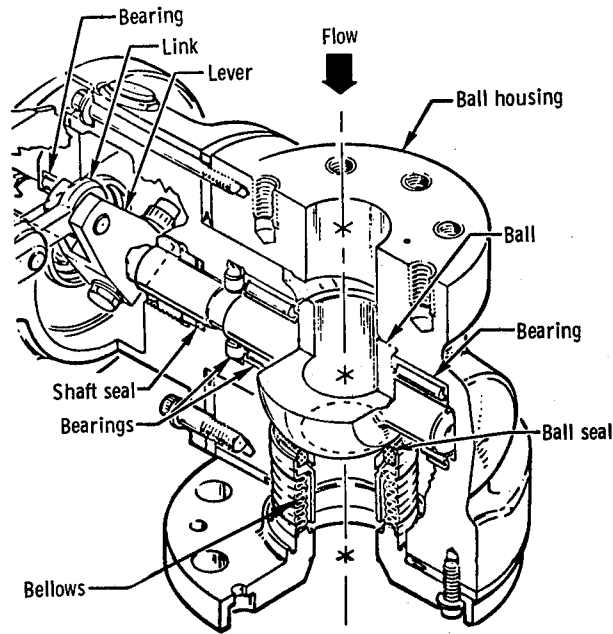


Figure 3. - Isometric cross section of ball valve.

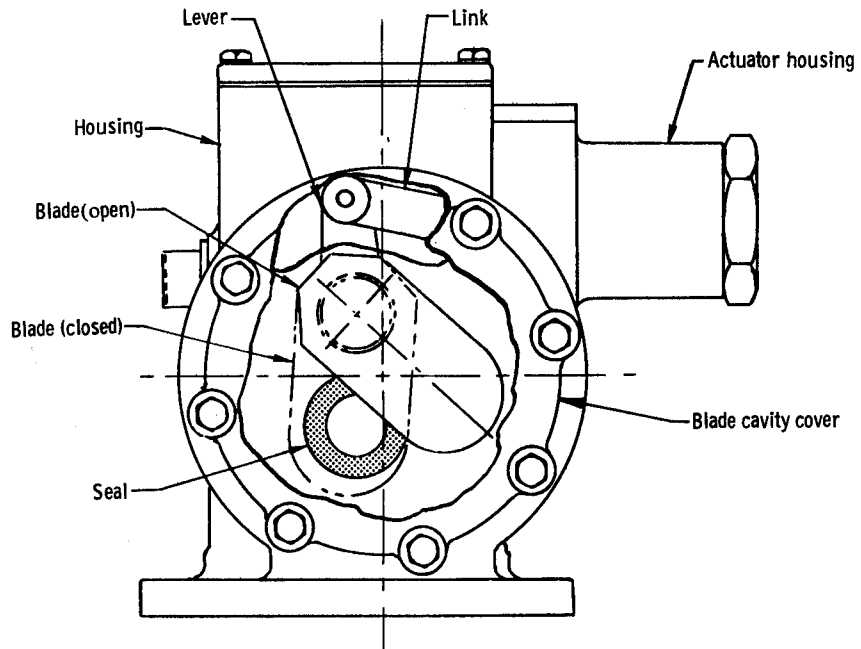


Figure 4. - Cross-section sketch of blade valve.



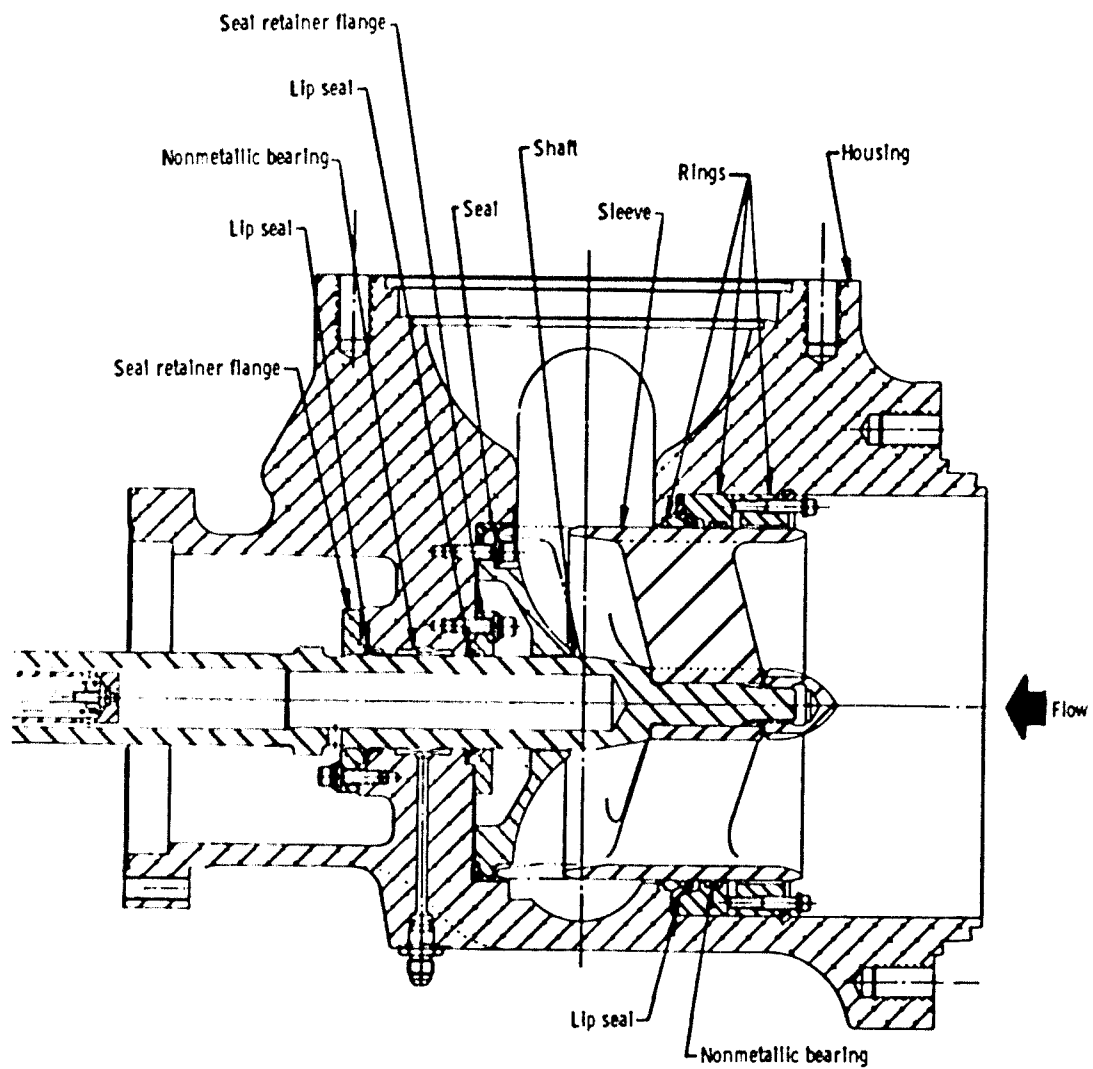


Figure 5. - Schematic of linear sleeve valve.

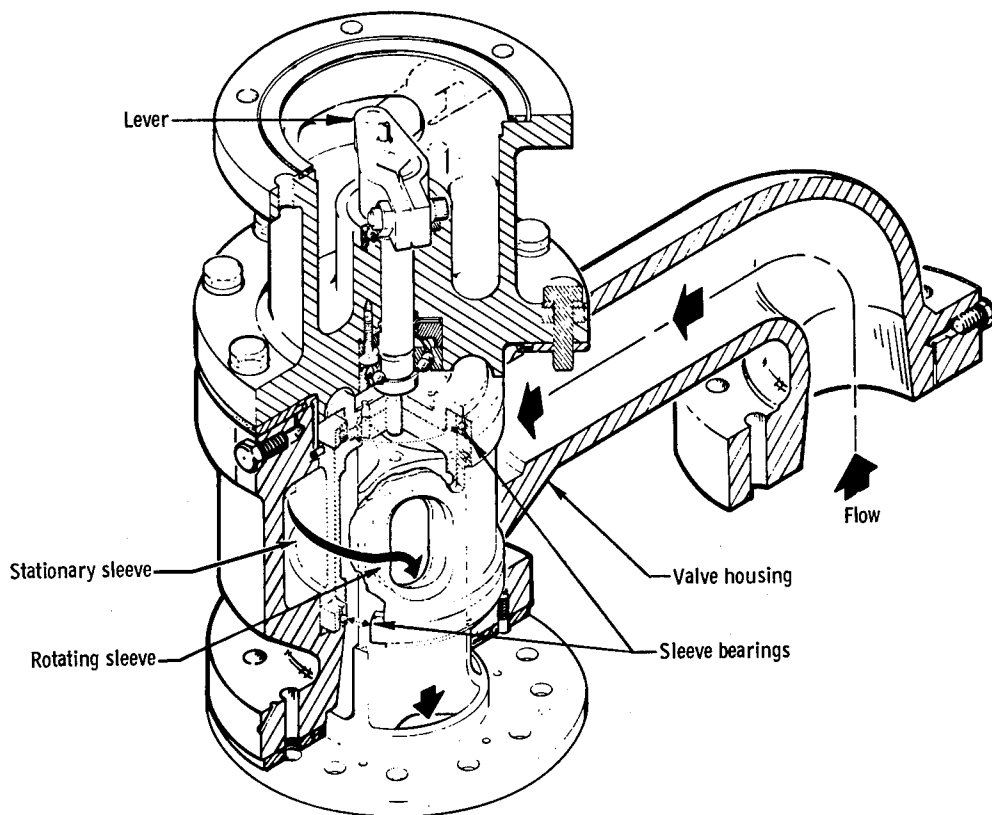


Figure 6. - Isometric cross section of rotary sleeve valve.

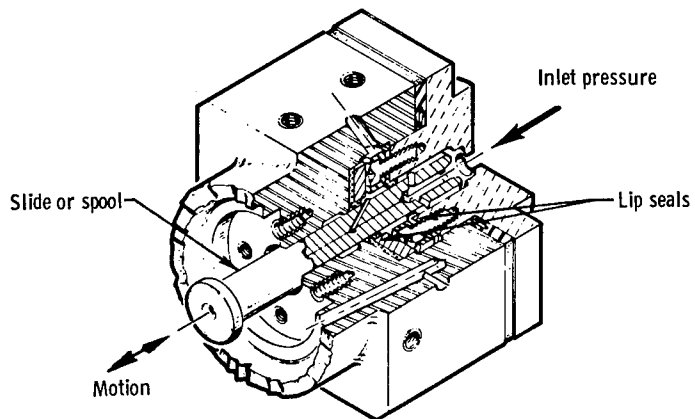


Figure 7. - Isometric cross section of cylindrical slide or spool valve.

Table 1. — Chief Features of Valve Components in Use on Operational Vehicles

Vehicle and engine	Valve use	Service fluid	Valving element	Valving element material	Seat or seal material	Valve housing material	Secondary seal type	Secondary seal material
Alfa VA 5	Boost MV	LO <sub>2</sub>	Butterfly	7055 T6	KelF	Tens 50	Lip seal	Mylar
	Boost MV	RP-1	Butterfly	7055 T6	KelF	Tens 50	O-ring	Buna N
	Sustainer GC	LO <sub>2</sub>	Blade	17-4 PH	KelF	156 T6	Face seal	KelF
	Inert gas	RP-1	Blade	17-4 PH	Buna N	156 T6	O-ring	Buna N
	Regulator	CS <sub>2</sub>	Poppet	303	Buna N	2024 T6	O-ring	Buna N
	Verter segment	He (gas)	Poppet	440C	nylon (Main seal)	2024 T6	O-ring	Buna N
		LO <sub>2</sub>	Poppet	2024 T4	KelF	156 T6	Lip seal	KelF
		RP-1	Poppet	2024 T4	KelF	7055 T6	O-ring	Buna N
		N <sub>2</sub> O <sub>2</sub>	Butterfly	17-4 PH	KelF	7055 T6	Spring loaded O-ring	Ethylene Propylene Rubber (EPR)
		A-50	Butterfly	17-4 PH	polypropylene	7055 T6	Urchandant Spring loaded O-ring	Rubber (EPR)
LR 91 AJ 5	GC MV	N <sub>2</sub> O <sub>2</sub>	Poppet	KelF	KelF	100 series	O-ring	Teflon
	GC MV	A-50	Poppet	KelF	KelF	156 T6	O-ring	Butyl rubber
	Four way control	A-50	Spool	440C	SA	156 T6	O-ring	Butyl rubber
	MOV	LO <sub>2</sub>	Butterfly	411	KelF	156 T6	Lip seal	Mylar
	MFV	RP-1	Butterfly	7055 T6	KelF	156 T6	O-ring	Buna N
	GC Inert gas	LO <sub>2</sub>	Poppet	A-286	FEP Teflon	44	O-ring	A-286
	Explosive-actuated two-way shutdown	RP-1	Poppet	A-286	FEP Teflon	44	Belows	Buna N
		RP-1	Diaphragm	17-4 PH	SA	2024 T4	O-ring	SA
		LO <sub>2</sub>	Poppet	6061 T6	Teflon	6151 T6	Vented redundant Special O-ring	Mylar
		RP-1	Poppet Ball	6061 T6	FEP Teflon	6151 T6	Vented redundant Special O-ring	Viton A
Saturn S-1C 1-1	Checkout valve	LO <sub>2</sub>	Ball	A-286	KelF	7055 T6 and Tens 50	Vented redundant Special O-ring	KelF
		RP-1	Ball	A-286	KelF	7055 T6 and Tens 50	Vented redundant Special O-ring	Buna N
		RP-1	Ball	Chrome-plated AM-100	FEP Teflon	Tens 50	O-ring	Buna N
	MOV	LO <sub>2</sub>	Butterfly	411	KelF	Tens 50	Lip seal	Mylar
	MFV	LO <sub>2</sub>	Butterfly	411	KelF	Tens 50	Lip seal	Mylar
	GC Inert gas	LO <sub>2</sub>	Poppet	A-286	FEP Teflon	6061 T6	Belows	A-286
	Oxidizer turbine bypass	LH <sub>2</sub>	Poppet	A-286	FEP Teflon	6061 T6	Belows	A-286
	Four way unidrom	Hot Gas (811 K)	Butterfly	Inconel X-750	FEP Teflon	Inconel X-750	Face seals on Belows	Chrome-plated Inconel X-750
	Regulator	He (gas)	Poppet	Fluorloy	440C	440C	N-wal	Nickel lead coated 17-4 PH
	Hy start tank vent and relief	He (gas)	Poppet	440C	440C	440C	Naflex <sup>®</sup> Packing	Teflon coated 4140 Steel
	H <sub>2</sub> (gas)	Poppet	440C	440C	440C	Naflex	Teflon coated 4140 Steel	

Apollo Service Module	Propellant utilization	LO <sub>2</sub>	Rotary sieve	440C	NA	Tens-50	Lip seal	Mylar
Apollo LEM LMDE	Prevalve	O <sub>2</sub> or H <sub>2</sub> (liquid)	Butterfly	A286	Kel-F	6061-T6	V-seal	Teflon-coated Inconel X-750
	Propellant tank vent and relief MFV sequence	O <sub>2</sub> or H <sub>2</sub> (gas)	Poppet	7075-T73	Kel-F	356-T6	Lip seal	Teflon
	MOV*	He (gas)	Poppet	A286	6061-T651	6061-T651	Lip seal	Kel-F
	MFV*	N <sub>2</sub> O <sub>4</sub>	Ball	17-4 PH	Filled Teflon	356-T6	Spring-loaded O-ring	Teflon
Apollo LEM LMDE	MOV*	A-50	Ball	17-4 PH	Filled Teflon	356-T6	O-ring	Butyl rubber
	MFV*	N <sub>2</sub> O <sub>4</sub>	Ball	17-4 PH	Teflon	6061-T6	Spring-loaded O-ring	Teflon
	Flow control	N <sub>2</sub> H <sub>4</sub> +UDMH	Ball	17-4 PH	Teflon	6061-T6	Spring-loaded O-ring	Teflon
	Prevalve	N <sub>2</sub> O <sub>4</sub>	Pinle	440C	NA	Ti-6Al-4V	Belows	AM-350
LMAE	MOV*	N <sub>2</sub> H <sub>4</sub> +UDMH	Poppet	Chrome plated 430	NA	Ti-6Al-4V	Belows	AM-350
	MFV*	N <sub>2</sub> O <sub>4</sub>	Ball	302	EPR	304L	O-ring	EPR
	Oxidizer inlet shutoff	A-50	Ball	302	Teflon	304L	Lip seal	Teflon
	Fuel inlet shutoff	LO <sub>2</sub>	Ball	Aluminum, highly polished, lightly anodized	Teflon, spring-loaded	Aluminum	O-ring	EPR
Centaur RL10	Main fuel shutoff	LH <sub>2</sub>	Ball	Aluminum, highly polished, lightly anodized	Teflon, spring-loaded	Aluminum	Belows	Stainless steel
	Oxidizer flow control	LH <sub>2</sub>	Poppet	Stainless steel	Stainless steel, lapped finish	Stainless steel	Belows	Stainless steel
	Fuel shutoff	LO <sub>2</sub>	Poppet	Aluminum	Aluminum, hardcoat finish	Aluminum	Belows	Stainless steel
	Oxidizer shutoff	N <sub>2</sub> O <sub>4</sub>	Poppet	Stellite No. 3	FEP Teflon	321	Weldment	NA
Geminis SE-6 & -7	Fuel shutoff	MMH	Poppet	Stellite No. 3	FEP Teflon	321	Weldment	NA
	Regulator	GN <sub>2</sub>	Poppet	Sapphire	6061-T6	6061-T6	Weldment	NA
	Fill	GN <sub>2</sub> or N <sub>2</sub> H <sub>4</sub>	Poppet	Ceramic	6061-T6	6061-T6	O-ring	Butyl rubber
	Explosive-actuated propellant	N <sub>2</sub> H <sub>4</sub>	Gate	303	6061-T6	6061-T6	Brazed joint	NA
Mariner Mars 69 M-50M	Bipropellant	N <sub>2</sub> O <sub>4</sub>	Poppet	17-4 PH	Teflon	17-4 PH	Weldment	NA
	Oxidizer boost termination	MMH	Poppet	17-4 PH	Teflon	17-4 PH	Weldment	NA
Mariner Mars 71 RS-21	Fuel boost termination	UDMH	Poppet	Tens-50-T4	1100	Tens-50	O-ring	Butyl rubber
	Fuel boost termination	UDMH	Poppet	Tens-50-T6	1100	Tens-50	O-ring	Butyl rubber

Notes: MOV = main oxidizer valve  
 MFV = main fuel valve  
 MOV\* = quad-redundant bipropellant valve assembly  
 MFV\* = not applicable  
 GG = gas generator  
 GGOV = gas generator oxidizer valve  
 GGFV = gas generator fuel valve  
 NA = not applicable  
 N<sub>2</sub>H<sub>4</sub>+UDMH = 50:50 mixture

Teflon and Kel-F as seal materials, improved seal retainers that limit the increase of seal load with pressure, and rolling-contact bearings to reduce friction in high-pressure applications. Wear of the seals under dry cycling conditions continues to be a problem.

Shaft seal designs have progressed from the O-ring to the plastic lip to the spring-loaded U-shaped plastic seal and culminate in bellows in an all-welded valve housing that reduces external leakage to zero. Flexure-tube-mounted poppets have eliminated the need for shaft seals in small valves and have minimized actuator galling and contamination generation.

General technological advances that played a part in successful valve design resulted from the generation of data on (1) material properties of metals and plastics at cryogenic temperatures, (2) compatibility of metals and plastics in various storable propellant applications, (3) properties of lubricants suitable for a given environment, and (4) cleanliness requirements to minimize contamination.

Valve designs for future liquid rocket engines probably will emphasize use of hard-seat poppet valves if engine requirements include more reactive propellants, greatly increased reusability, longer storage life, and capability to withstand prolonged exposure to radiation and vacuum environments. However, if compatibility is not a problem, soft seat valves may be used. Butterfly or ball valves used for propellant valves in booster applications may incorporate mechanisms that retract the seal so it is not damaged or worn during valving-element rotation. A seal-retracting mechanism has been used successfully in the 17-in. (43 cm)<sup>1</sup>- diameter ball (visor) valve used on Saturn IC (ref. 4, sec. 6.2.2.14).

## 2.1 POPPET VALVING UNIT

Concurrently with selection of a poppet valving unit, the designer must select the types of sealing surfaces, hard or soft, that are optimum for the poppet and the seat. In this monograph, hard and soft sealing surfaces are defined in terms of the kind of material composing the surface: a hard sealing surface is one fabricated of material (metal, ceramic, or cermet) that does not permanently yield or deform except with wear (flexible metal disks are a special type of hard sealing surface); a soft sealing surface is one fabricated of plastic or elastomer. Configuration designations for combinations of hard and soft sealing surfaces are as follows:

<u>Poppet Sealing Surface</u>	<u>Seat Sealing Surface</u>	<u>Configuration Designation</u>
Hard	Hard	Hard-on-hard
Hard	Soft	Hard-on-soft
Soft	Hard	Hard-on-soft

<sup>1</sup> Parenthetical units are in the International System of Units (SI units). See Mechtly, E. A.: *The International System of Units, Physical Constants and Conversion Factors, Second Revision*, NASA SP-7012, 1973.

The designer selects the sealing-surface materials (metal, ceramic, cermet, plastic, elastomer, or combinations thereof) on the basis of the design factors discussed below. In some cases, it may be necessary for the designer to consider all the factors successively and to weigh the advantages and disadvantages of each type of sealing surface; in other cases, a consideration of only one factor will lead to a selection; e.g., propellant compatibility considerations probably will lead to selection of hard sealing surfaces for a valving unit for fluorine service.

Allowable leakage rate. – The allowable leakage past the poppet valving unit is one of the most important considerations in selecting a hard seat or a soft seat. The allowable leakage may be specified in terms of leakage of propellant or leakage of a leak-test fluid.

Reference 5 indicates that it may take 3 to 4 hours to stabilize a liquid leakage rate, and that zero liquid leakage may be anticipated when gaseous leakage is approximately 0.03 scch of helium at a pressure differential of 14.7 psi (101.3 kN/m<sup>2</sup>). References 6 and 7 present methods for correlating leakage rates measured with different fluid media at different temperatures.

Acceptable leakage rates are achieved by blocking all microscopic leak paths by continuous mating between the two sealing surfaces. Such continuous mating may be achieved by high seat stresses and fine surface finishes or by making one surface soft and compliant so that it will readily conform to irregularities in the hard surface. If the allowable leakage rate is generous (e.g., the rate for a poppet valve in hydraulic service), then the designer usually selects a hard-on-hard configuration because it has fewer parts than a soft-sealing-surface configuration and potentially has less limitations due to useful temperature range and propellant compatibility. If the allowable leakage is small, selection is more difficult.

The technology of sealing a hard poppet on a hard seat is treated comprehensively in reference 8. The technology of sealing a hard poppet on a soft seat or soft poppet on a hard seat is not as well documented. Some of the principles presented in reference 8 are applicable, and other pertinent sealing technology is available in references 9 through 13 and section 6.2.3.11 of reference 4. Reference 14 discusses the use of elastomeric seals in poppet valving units. Small ( $\leq 0.25$  in. [6.35 mm]) hard-seat valves in pneumatic service typically have maximum allowable leakage rates of approximately 5000 scch gaseous helium at 2800 psi (19.3 MN/m<sup>2</sup>). An outstanding example of the low leakage possible with a hard-seat poppet valve, if extreme care is taken, is that of the Mariner-spacecraft pressure regulator, which has a maximum leakage specification of 10 scch of gaseous nitrogen at 300 psi (2.1 MN/m<sup>2</sup>) (ref. 15). Low leakage rates are attained by lapping poppet and seat to obtain better seat finishes. Reference 8 gives a method to predict the leakage of a hard-on-hard configuration for a given surface finish.

Normally open explosive-actuated valves with hard seats typically have maximum post-actuation allowable leakage rates of  $3.6 \times 10^{-3}$  scch helium at operating pressures ranging up to 3000 psi (21 MN/m<sup>2</sup>). This low value illustrates the leakage rate that can be

attained with a single actuation of a hard-seat poppet valve with a force sufficient to yield the sealing surfaces plastically.

Leakage of hard-on-soft poppet valves typically is less than that of hard-on-hard poppet valves of comparable size. A propellant valve with a Teflon seat used in the SE-8 reaction control system for the Apollo spacecraft has a maximum leakage specification of 0.2 sech of gaseous nitrogen at 300 psi (2.1 MN/m<sup>2</sup>). The propellant valve on the Intelsat III thruster had a leakage specification of 0.5 sech gaseous nitrogen and consistently demonstrated leakage rates in the range of  $3.6 \times 10^{-3}$  to  $3.6 \times 10^{-4}$  sech helium (ref. 14); this low rate required a mass spectrometer for measurement. This valve incorporated dual in-line poppets with approximately 1/8 in. (3.18 mm) elastomeric seals. The leakage rates were low because of the dual seals and because the elastomer is even softer and more compliant than the plastics commonly used for poppet valving unit seats or seals.

Thus, if other selection factors permit, the designer will choose a hard-on-soft configuration for low-leakage applications, and either a hard-on-soft or a hard-on-hard configuration when the maximum allowable leakage is significantly higher.

Contamination environment. – The contamination environment in which the valving unit must operate is critical because contamination can cause a leak either by lodging between the sealing surfaces or by damaging the sealing surfaces. The prime advantage of a soft sealing surface is the capability of enveloping contaminant particles after repeated cycling; the surface has a “self healing” capability. Elastomers in general are softer than plastics and thus have more ability to envelop contaminant particles. Hard-seat valving units are susceptible to leakage caused by entrapment of a particle between the sealing surfaces unless the actuator force is sufficient to crush or flatten the particle.

Small valves (<1 in. [2.54 cm]) such as pneumatic control valves or reaction control system valves usually incorporate filters in the valve inlet to remove contaminants. Although filters are disadvantageous because of the added size, weight, and pressure drop, larger valves may incorporate filters if there is a requirement for low leakage after multiple closures. The 2-in. (5.1 cm) propellant valves on the Lunar Module Descent Engine (LMDE) are examples of valves with filters to ensure a high degree of sealing reliability during prolonged coast periods. Generally, however, large propellant valves do not incorporate inlet filters because of the disadvantages noted; such valves then must exhibit a high degree of contamination tolerance.

Propellant compatibility. – The compatibility of sealing-surface materials with the propellant can be a major factor in the selection of hard or soft materials. With hard-on-hard configurations, many materials are compatible with state-of-the-art propellants, and a number of them can be used with the advanced fluorine-base oxidizers. In the selection of soft seats, however, compatibility with the propellant generally will restrict the number of suitable materials. Numerous plastic materials are compatible with current propellants, but

propellant compatibility may be the major factor in determining whether an elastomeric soft sealing surface can be utilized. Although several elastomeric compounds can be used with storable fuels, few are compatible with storable oxidizers. Soft sealing surfaces such as glass- and metal-filled Teflons have been subjected to limited testing in fluorine (ref. 16, para. 4.4.1); even with satisfactory results, however, it was concluded that the use of such soft seals in fluorine service would require extensive testing to demonstrate acceptable reliability.

Operating pressure. – Hard-on-hard configurations typically are used in pneumatic control valves that operate at high pressure, because the pressure differentials can result in high-velocity flow that can erode a soft seat. An unacceptable leak in a Kynar seat in a regulator operating at 5000 psi ( $34 \text{ MN/m}^2$ ) was attributed to initial seat damage caused by contaminant impact during a fast start of the engine; the initial damage was followed by a gas-particle/surface interaction. The latter caused massive failure of the seat (ref. 17). Soft sealing surfaces typically are used in propellant valves operating in the 100 to 300 psi ( $0.69$  to  $2.07 \text{ MN/m}^2$ ) pressure range.

Valve size. – Valve size is important because it affects alignment of the poppet and the seat. Typically, hard seats are used in small valves (i.e.,  $<1$  in. [ $2.5$  cm]). Hard seats in large valves require complex misalignment compensation that increases cost and extends fabrication time. In large valves, therefore, soft sealing surfaces are used, because the lower yield stresses of plastics or elastomers permit local deformation to compensate for misalignment between the poppet and seat.

Actuator size. – Generally, an actuator for a valve with a soft sealing surface is smaller than one for a valve with hard sealing surfaces, because a satisfactory seal can be achieved with less load. The lower seal-load requirement of a soft seal also is advantageous in keeping size small when a low pressure acts on a small unbalanced area to provide the force to hold the valve closed. The soft sealing surface also requires less sealing overload to achieve a satisfactory seal with contaminants present, whereas the hard-on-hard configuration must have sufficient sealing overload to crush or deform contaminants so that the sealing surfaces can make sealing contact.

Temperature limits. – Hard seats are, of course, satisfactory both at high temperatures and at cryogenic temperatures. The properties of the seat do not change greatly in the low-temperature environment. Soft seats, however, are much more limited in useful temperature range.

Valves with Teflon seats operated satisfactorily at  $200^\circ\text{F}$  ( $367 \text{ K}$ ) in development tests of SE-6 and SE-7 reaction control components (ref. 18). However, temperatures higher than  $250^\circ\text{F}$  ( $394 \text{ K}$ ) may cause detrimental creep or excessive yielding that could cause leakage and pressure-drop changes in valves with Teflon seats (refs. 19 and 20). Excessively high temperatures for soft-seal valves have resulted from high ambient temperature, long-term



energization of a solenoid, thrust chamber heat soakback (ref. 21), or sterilization temperatures. The change in properties of suitable soft sealing materials at cryogenic temperatures will usually require that relatively high sealing loads be used to achieve a satisfactory seal; also, seal deflection may be critical at low temperature.

Cycle life. – Soft sealing surfaces tend to yield after cycling, and the consequent reduced stress can cause leakage. In addition, seat yielding has changed the diameter at which the poppet seals and thus changed the force balance of the poppet and caused actuation problems. After very many cycles, a solenoid-operated propellant valve used on the SE-5 engine opened more slowly than specified during acceptance testing, because cracks in the Teflon seat changed the effective sealing diameter (ref. 22). However, similar Teflon-seat valves have operated satisfactorily after 50,000 actuation cycles (ref. 18). A soft-seat valve designed for a 0.5 lbf (2.2 N)-thrust development engine was in satisfactory condition after completion of 1,000,000 cycles.

Flat metal-to-metal sealing surfaces were satisfactory after 1,000,000 actuation cycles during the research and development program on poppets and seats reported in reference 8.

Valves having a cycle life of one actuation (e.g., an explosive-actuated valve) typically have hard seats, because high actuator forces are easily obtained and allowable seat deformation is very large.

While the development experience does not indicate a superior cycle life for either the hard or soft configuration, it is generally believed that the hard-on-hard configuration has an inherently superior cycle life because of its greater strength.

Operational and storage life. – Poppet valves with Teflon seats were used on lunar missions having durations of approximately one week. Studies of requirements for other missions predict durations of 2, 5, and 10 years. Component design guidelines established for these long-life spacecraft discourage use of plastic sealing surfaces if the surface is continuously loaded, because the plastic may yield over long storage periods. However, tests of hard-on-soft poppet valves after approximately 3 years under spring load indicated no degradation of valve performance; therefore, soft-sealing-surface valves may prove to be suitable for relatively-long-term use.

Soft-seal retention. – If a soft sealing surface is selected as desirable, then the designer must also decide whether the surface should be located in the housing or in the poppet. The location is determined largely by consideration of the space available and fabrication costs. Either location is satisfactory. However, the advantages of soft sealing surfaces must be balanced against the difficulty of retaining them in the poppet or in the housing. For example, press-fit retention of Teflon seats used in reaction control system valves may cause extremely high installation stresses that result in creep of the seat. Soft seats, especially O-rings, have been forced out of retainers by high-pressure and high-velocity flow during valve actuation (ref. 23). Additional detail parts are required to retain a soft-seal material.

Secondary leak path. — A disadvantage of soft seats or seals is that they introduce a secondary leak path, i.e., a path between the seat and its retainer. For example, a vent valve on a J-2 engine leaked past the flange of the soft seat when the retainer loosened under vibration. This kind of problem is minimized by careful control of seal and retainer loads, materials, fabrication, and installation.

Manufacturing requirements and cost. — Hard seats generally are integral with the poppet element or valve housing and do not require retention or installation. Fabrication difficulties with hard seats, however, result from the fine surface finish required and the protection necessary to prevent damage to the seat surface because the seat cannot be replaced. Generation of fine surface finishes is costly, especially if hand lapping processes are required.

Plastics, because they deform easily, are more difficult to machine and to inspect than metallic seats. Finish machine tolerances of Teflon seats may be required to be as low as  $+0.0005$  in.,  $-0.0000$  in. ( $+12.7$   $\mu\text{m}$ ,  $-0$   $\mu\text{m}$ ) in order to control the location of the sealing diameter and to control seat dimensions so that yielding is minimized. Welded-retainer designs and manufacturing techniques also must be carefully controlled to prevent welding heat from damaging a soft sealing surface (ref. 24). Teflon seat designs have had extremely high rejection rates in manufacture (ref. 25). Elastomeric seats and seals are generally low in cost; this is a major advantage of this type of seal. They are also less subject to damage or scratches in handling than are hard or plastic sealing surfaces.

Sealing-surface material properties. — One prerequisite for a successful analysis of sealing-surface design parameters is that the properties of the seal material be known. Material properties of the metals used are well known. The material properties of ceramics and cermets, particularly impact strength, are not well established. The properties of elastomers and plastics in actual valve operation are difficult to establish accurately, because the material properties are affected by rate of load application, temperature, age, number of cycles, reaction with propellant, fabrication processes, and seat encapsulation or backup configuration. Thus, the analytical basis for the design of soft sealing surfaces is less secure than that for hard seats and seals.

Commonality of design. — When both an oxidizer valve and fuel valve are required for a rocket engine system and the choice of hard seat or soft seat to seal one of the propellants has been made, usual practice is to use the same design to seal the other propellant if the seal material is compatible with both propellants. This practice minimizes design and development effort, since there is only one basic design. Thus, the suitability of a seat design for common use can be a factor in the choice of configuration.

## 2.1.1 Hard-on-Hard Unit

The hard-on-hard poppet valving unit incorporates a poppet and a seat fabricated of metal, ceramic, or cermet. The seat may be integral with the valve housing or it may be an insert in the housing. Satisfactory performance of a hard-seat unit depends on geometry, surface finish, static and transient stress, and alignment of the sealing surfaces.

### 2.1.1.1 GEOMETRY AND SURFACE FINISH

Selection of geometry for the poppet and seat is based on consideration of leakage, cycle life, fabrication, vibration, service environment, and other design factors. The most common combinations of poppet and seat geometries — conical on conical, spherical on spherical, flat on flat, and spherical on conical — are shown in figure 8. Reference 8 indicates that conical

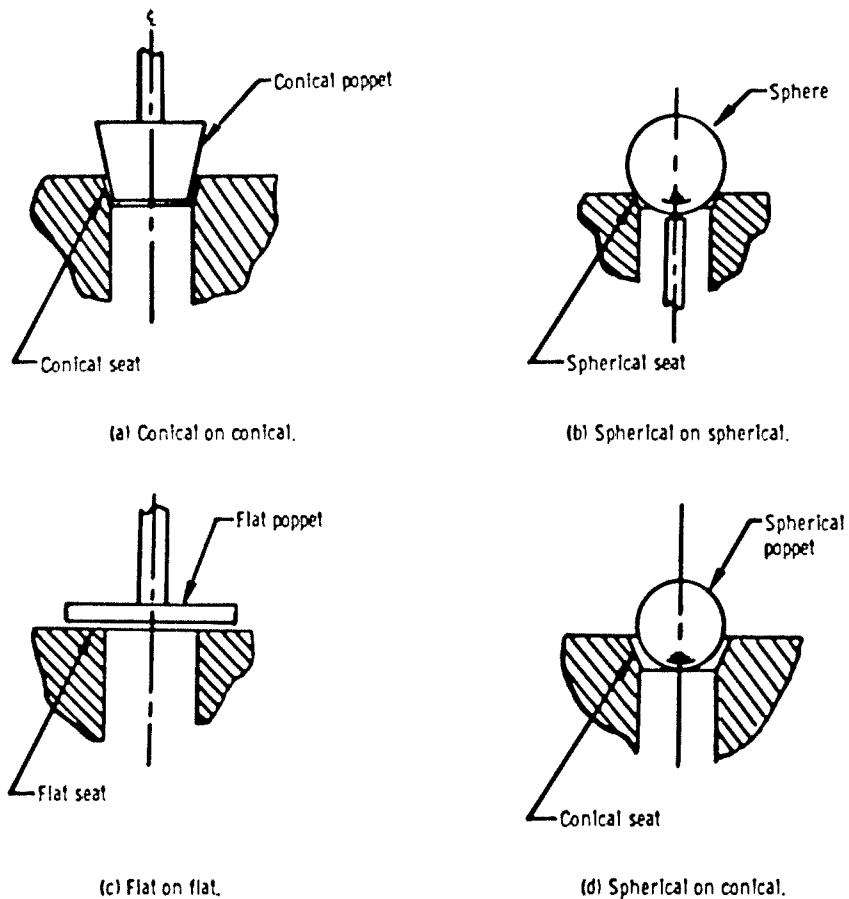


Figure 8. - Cross-section sketches of typical poppet-and-seat geometries.

and spherical geometries have the following advantages over flat surfaces in hard-seat designs:

- (1) The mechanical advantage inherent in these geometries makes it possible to achieve design seat stresses with a reduced load.
- (2) The intersection of the poppet and seat results in a narrow seat width.
- (3) When the poppet is seated, there exists a force component that resists lateral motion due to vibration.

An overriding disadvantage of conical and spherical geometries is that these shapes require a close match of physical dimensions: the included angles of the conical poppet and conical seat, and the radii of the spherical poppet and spherical seat. As a result, these configurations almost always have a taper gap ranging from a few microinches to several thousandths of an inch, depending upon size of the valve, fabrication process, and measurement precision. This gap results either in excessive wear during seating if seat loads are high or in excessive leakage if seat loads are low. In some valves, conical poppets and seats are machined to identical angles; in other valves, however, different angles are used, and the poppet and seat are lapped together to generate the same included angle in the contact area. Constrained conical poppets usually have axial tilt as a positional error.

An important advantage related to conical geometry is that for a given tilt angle between the axis of the poppet and axis of the seat (misalignment) the leakage gap of conical geometry is much less than the leakage gap of a flat configuration. Conical geometry typically is used in hydraulic system valves where the allowable leakage may be 1 in.<sup>3</sup> (16 cm<sup>3</sup>) of hydraulic oil per minute, and cost considerations prohibit fabrication of perfect alignment of a poppet and integral guided shaft configuration. This geometry is, of course, subject to wear problems if it is misaligned.

The great advantage of spherical geometry is that it is free from positional errors because it is self-aligning. Spherical geometry is more costly to fabricate than conical or flat geometry; however, cost may be reduced by using commercial spheres as poppet valving elements. The ratio of sphere diameter to spherical seal diameter generally is not critical. Relatively small ratios result in more wedging of the sphere, which increases the contact stress and decreases leakage. In high-pressure pneumatic valves, spheres and spherical seats are lapped to a surface finish of 1 to 2  $\mu$  in. (0.025 to 0.051  $\mu$ m) AA.

Flat sealing-surface geometry is easy to fabricate by lapping the seat to near-perfect conformance with the flat sealing surface on the poppet, and this mating is easy to inspect with an interference microscope. Low leakage and long cycle life result from use of flat hard poppets and seats. Flatness specifications range from 5.8 to 23.2  $\mu$  in. (0.15 to 0.59  $\mu$ m) AA. The fabrication process typically is grinding followed by master lapping by machine or by

hand. Lapping procedures and tooling are discussed in reference 9. Poppets and seats are not match lapped, because of interchangeability problems and because actuation of the poppet may change its orientation with the seat and cause leakage. An example of flat geometry is the 3-way helium-shutoff solenoid valve on the J-2 rocket engine. This valve has flat seats and poppet sealing surfaces that are lapped to better than  $2\mu$  in. ( $0.051\mu\text{m}$ ) AA. The valve easily meets leakage specifications of a maximum of 59 scch of helium over the range of 10 to 600 psi (69 to  $4140\text{ kN/m}^2$ ) at ambient temperature and a maximum of 4920 scch of helium over the same pressure range at  $-320^\circ\text{F}$  (78 K). The disadvantage of flat geometry is that a constrained flat poppet may not be parallel to the seat, so there may be a gap that permits leakage. However, by flexure mounting the poppet, as shown in figure 9, parallelism can be maintained and the basic benefits of flat geometry can be realized (ref. 26).

A variant of the flat-on-flat geometry is the flexible-metallic-disk design (fig. 10) that is used in a four-way control valve on the J-2 engine and on the S-II stage. This concept employs two pairs of simultaneously seating flexible disks that deflect sufficiently to compensate for seat-to-seat mismatch caused by dimensional variations due to tolerances and to temperature effects. Seat-to-seat mismatch is limited to 0.0002 in. ( $5.1\mu\text{m}$ ) by restricting the stackup of linear dimensional tolerances (refs. 27 and 28). The valve as used on the J-2 engine has maximum specified leakage rates of 5.0 scim (4916 scch) of helium at 400 psi ( $2.8\text{ MN/m}^2$ ) and room temperature and 50 scim (49,160 scch) of helium at 400 psi ( $2.8\text{ MN/m}^2$ ) and  $-320^\circ\text{F}$  (78 K) after 10,000 cycles. On the S-II stage, the valve controls 750 psi ( $5.17\text{ MN/m}^2$ ) helium gas.

Initial development cycling tests with sharp-corner seats in this valve resulted in gouged and abraded seats. Also, the leakage was excessive because, in addition to the abrasion, the leaking gas had almost no contact length to travel past the sharp seat corner. The seat was contoured to maximize seal contact length with the seat, and the sharp corner was radiused (fig. 11). A minimum disk-seal deflection substantially greater than the seat-to-seat mismatch was used to compensate for runout of the seat and ripples in the disk caused by retainer torque. Use of this valve has been limited to 750 psi ( $5.17\text{ MN/m}^2$ ) maximum operating pressure.

A spherical poppet and conical seat used as a combination offer both freedom from positional errors afforded by the spherical poppet and low cost for the conical seat. This configuration basically is equivalent to the spherical-poppet-on-spherical-seat configuration, since the conical seat tends to yield with cycling and approximate a spherical seat. An example of this geometry is a bipropellant valve designed for fluorine/hydrazine-blend propellants. This valve uses a flexible, spherical shell of phosphor bronze that seats against a conical seat in the valve body; the shell is designed to deflect and wipe against the seat. The valve has been subjected to limited testing in a development program. Cycle life is limited, apparently because the wiping action of the shell against the seat results in wear.

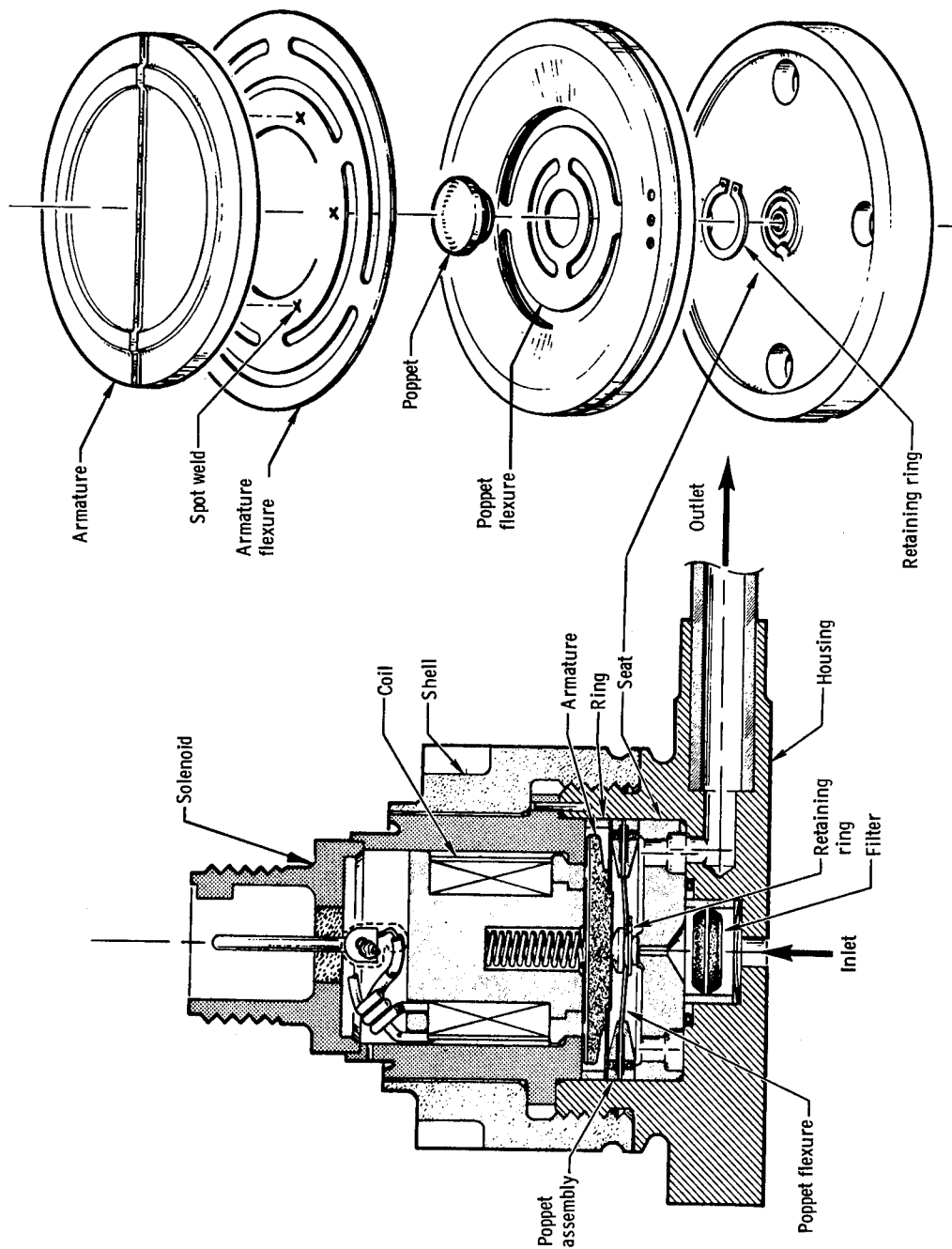


Figure 9. - Cross-section drawing of solenoid valve with flexure-mounted poppet.

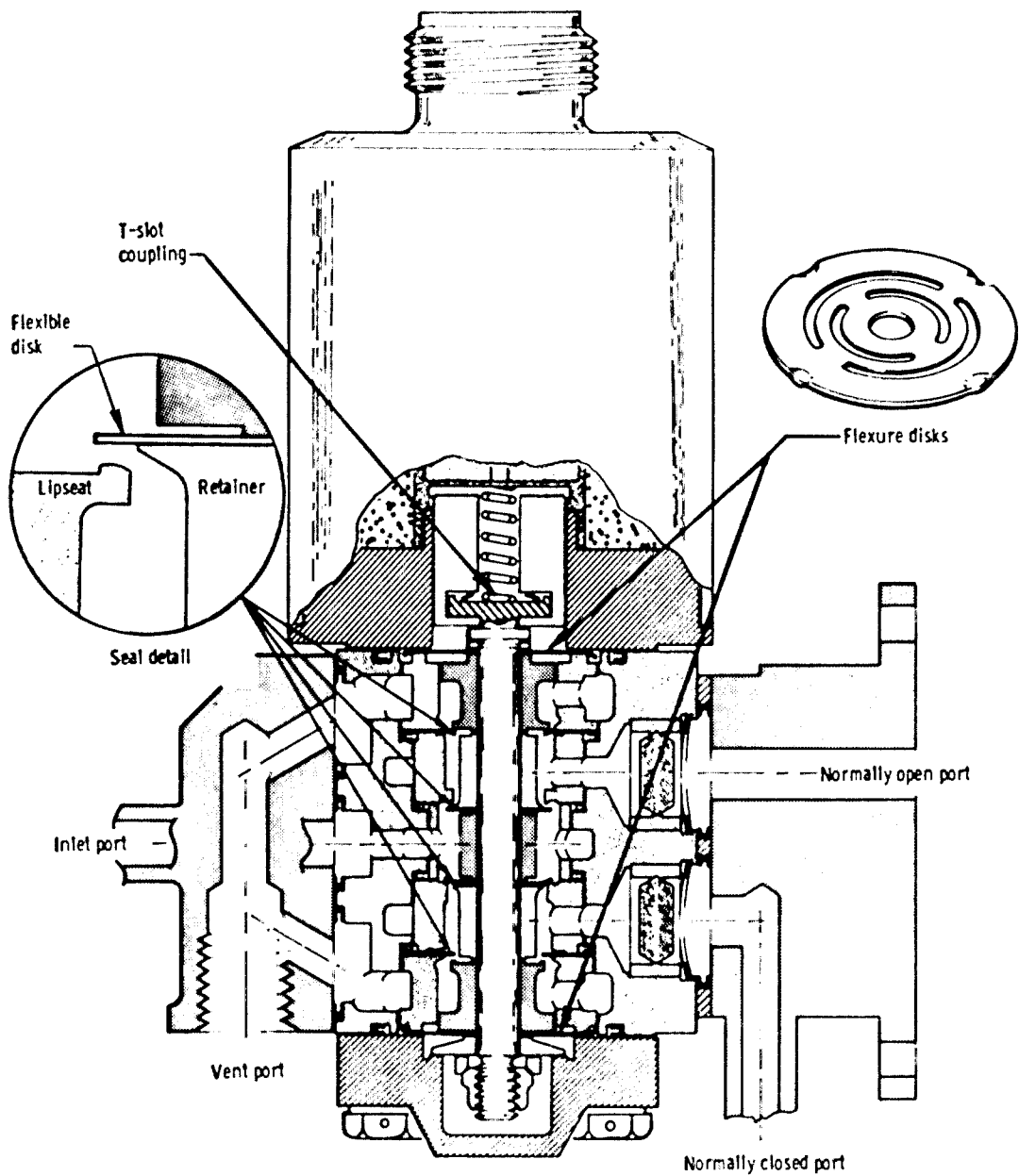


Figure 10. - Cross-section drawing of valve with flexible-disk seals (four-way control valve on J-2 engine).

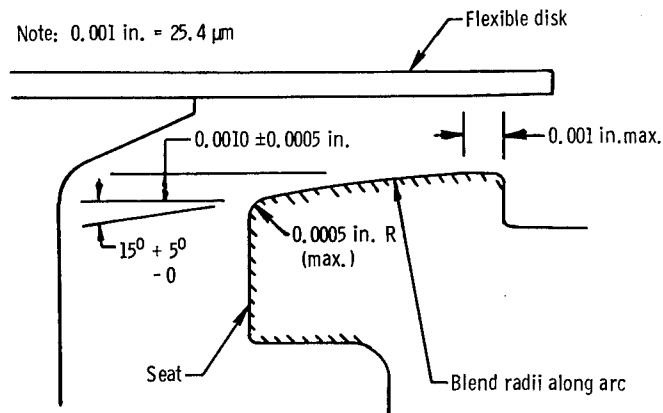


Figure 11. - Sketch of seat geometry for flexible-disk/hard-seat valve shown in figure 10.

### 2.1.1.2 STATIC SEAT STRESS

Factors that influence selection of static seat stress (seat load divided by seat land area in contact with the poppet) for hard-on-hard poppets and seats include the leakage requirement, fluid contamination level, seat shape, cycle life, available load, land width, surface finish, and material yield strength. Typical static seat stresses (assuming satisfactory alignment of the sealing surfaces) range from approximately 1000 to 10,000 psi (6.895 to 68.95 MN/m<sup>2</sup>). If the sealing surfaces have perfect geometry and alignment, satisfactory sealing can be attained with seating stresses as low as 100 psi (0.69 MN/m<sup>2</sup>); however, valves with a seat stress of only 100 psi (0.69 MN/m<sup>2</sup>) have leaked because the load was not sufficient to deform a contaminant particle or the sealing surfaces elastically and plastically so that sealing contact around the contaminant was achieved. Because perfect conformance of mating surfaces is not possible to achieve in practice, stress should be high enough to deform the sealing surfaces elastically to reduce gaps caused by positional errors. A static seat stress in excess of 10,000 psi (68.95 MN/m<sup>2</sup>), however, may result in local stresses greater than the material yield strength and cause permanent changes in hard geometry.

Stresses for the flexible-disk valving unit must necessarily be higher than those for a conventional poppet valving unit, because the disk must be deflected to the extent necessary to seal effectively. In this type of valve, the moving element stops on large-area metal-to-metal stops rather than on the seat, and therefore the impact stresses associated with a conventional poppet valving unit do not occur. The critical stresses are the stress of the flexible disk in the fully deflected position and the bearing stress of the seat. Bearing stresses are kept low to minimize wear. Because of the high stress values required for disk flexure and the relative complexity of the stress analysis, tests are performed to verify cycling capability.



### 2.1.1.3 TRANSIENT SEAT STRESS

Because of impact loads and stress concentrations, the actual stresses in a hard-seat poppet unit are many times greater than the static seating stress discussed above. However, impact stresses generally are only estimated. Excessive stresses result in yielding or wear of the seat and consequent leakage or, in a vacuum, may result in cold welding of the poppet to the seat (ref. 29). When the poppet closes against the seat, damage may occur because of high impact loads or because of temporary misalignment that concentrates the load in a local area. Also, a poppet without significant viscous damping (the condition in most poppet valves) tends to bounce and thereby experience several significant impacts per closing cycle. Some hard-on-hard designs employ an outer metal bumper land slightly below the sealing surface to receive the impact of a poppet that is not parallel to the seat at the time of impact (the usual case).

Limitation of high localized stresses due to stress concentration requires that the sharp corners of a hard seat that mates with a wider poppet land be relieved. Small-radius corners on the seat generally result in highly stressed areas at the edges. These areas predominate as the sealing surfaces, and comparatively short scratches on the corners (generally the most poorly finished areas) can cause leakage. Edge configurations that distribute the effective load over a wider central seat land area where the finish is likely to be more uniform have been developed (fig. 12). The crowned shape (fig. 12(a)) is good but difficult to fabricate as explained in reference 8. A modified chamfer (fig. 12(b)) on the sharp edges also is effective, and is not difficult to achieve.

### 2.1.1.4 ALIGNMENT

The sealing surfaces of a flat, hard-on-hard poppet valving unit must be parallel to each other if leakage is to be controlled to required values. Typically, in small valves, the poppet element is actuated by a free-floating push rod and therefore is self-aligning. Parallelism is more difficult to achieve in large valves, where the poppet disk is attached to the shaft. The lack of parallelism results from the tolerances required on normality of the poppet disk sealing surface to the poppet shaft and normality of the seat to the shaft bore. A tight clearance between the shaft and its bore may prevent the poppet from tilting sufficiently to close up this lack-of-parallelism gap. Diametral tolerances, normality tolerances, shaft guide length, and poppet sealing-surface diameter are dimensioned to permit adequate tilt of the poppet. Standard precision-machining techniques can produce a normality of the poppet sealing surface to its shaft and valve seat to shaft bore of 0.00025 in. (6.35  $\mu\text{m}$ ) for a 1.5-in. (3.81 cm) seat diameter, with correspondingly lower values for smaller seat diameters.

The propellant valves for the Lance propulsion system leaked during component testing because the sealing surfaces of the poppet and seat were not parallel at valve closure as a result of misalignment of the guided poppet configuration; the clearance between the

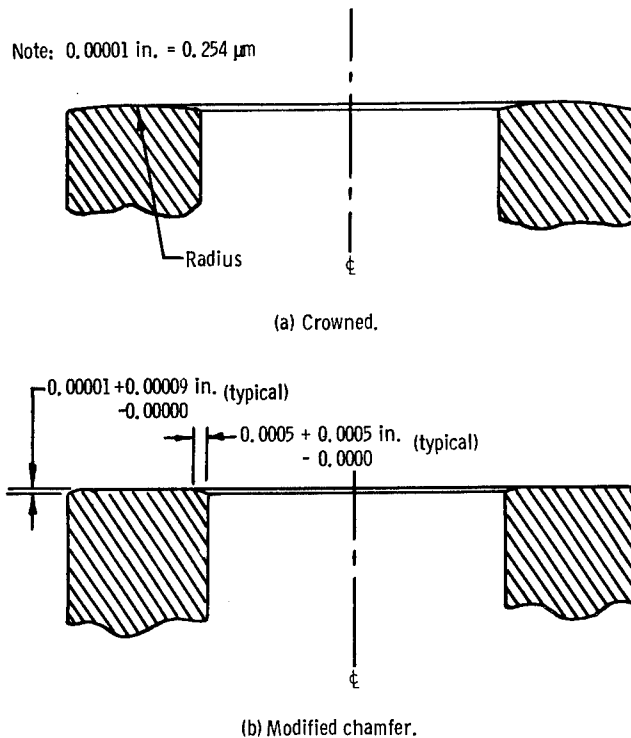


Figure 12. - Cross-section sketches of edge configurations for a flat-seat poppet.

poppet and guide was increased to allow the seat to conform even with small misalignments. In a check valve for the S-II stage of the Saturn V vehicle, compensation for misalignment resulting from normality tolerances is achieved with a ball joint mounted on the poppet.

A sphere and spherical seat may be misaligned prior to seating because their axes do not coincide. However, when fully seated, the sphere is self aligning. The sphere-in-a-conical-seat configuration also tends to be self aligning.

Axial tilt of conical seats and poppets presents a problem because alignment moments for a tilted cone are a complex function of the load application point and interfacial friction.

Reference 8 indicates that unless the seating load is applied below the seating line (towards the apex) and is perfectly axial, conical seats probably are not self-aligning. The conical configuration, therefore, must have a narrow land and high seating loads to reduce the gap caused by axial tilt.

## 2.1.2 Hard-on-Soft Unit

A poppet valving unit with a hard sealing surface meeting a soft sealing surface may incorporate the soft sealing surface as an insert of elastomer or plastic in the housing; this design is designated as a soft-seat poppet valving unit. Or it may incorporate the soft sealing surface either as an integral part of the poppet (e.g., a plastic poppet) or as an insert of elastomer or plastic in the poppet; this design is designated as a soft-poppet valving unit. (In either location, the soft sealing insert is designated as a seal).

Satisfactory operation of a hard-on-soft poppet valving unit requires consideration of the geometry and finish of the sealing surfaces, retention of the seal, stress at the seal interface, and alignment.

### 2.1.2.1 GEOMETRY AND FINISH

Current hard-on-soft poppet valving units incorporate various sealing-surface geometries including flat, spherical, and combination (e.g., a spherical poppet in a conical seat). Poppets that seal on plastic lip seals are used when a pressure-actuated seal is required or when over-travel is required, as in a dual-seat bipropellant valve (ref. 30).

Common practice in small valves is to use a commercial sphere as the poppet, or use a conical poppet. A spherical or conical poppet facilitates using a corner of a plastic seat as the point of sealing (fig. 13(a)); this practice provides a definite dimension on which to base the pressure-sealed area for calculating actuator force. A definite point of sealing of a flat seat is accomplished by using a narrow soft seat that may be encapsulated in a metal carrier to provide strength or by using a seat beveled at a small angle to provide a corner (fig. 13(b)).

Generally, local yielding is controlled by having a wide metal sealing surface contact a narrow soft sealing surface. The reverse practice can generate problems. For example, in the original design of a valve assembly for the gas generator on the J-2 engine, a metal poppet with a plastic sealing surface contacted a narrow land in the valve housing. This valve leaked after cycling. It was found that in operation the narrow land deformed the plastic seal

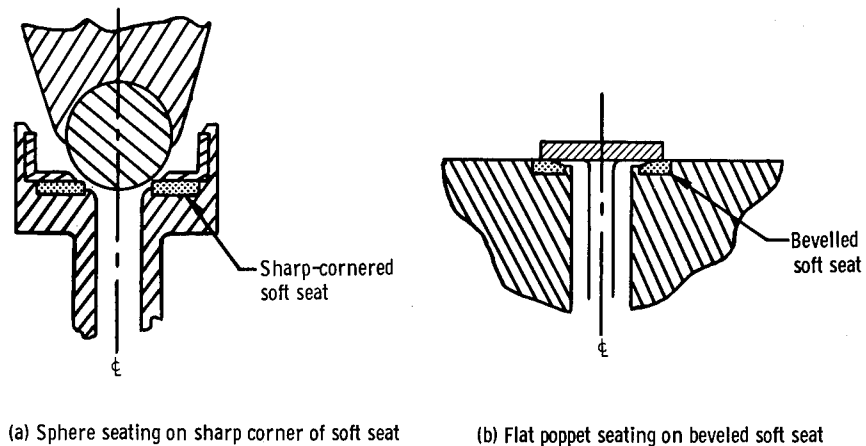


Figure 13. - Cross-section sketches of typical configuration for a soft-seat poppet.

locally. The deformation or groove where the seal had yielded provided a leak path on subsequent actuations when the poppet seated eccentrically. The poppet seal was redesigned to reduce the unsupported height of the poppet plastic seal surface above the metal base and thereby reduce the yielding. The problem did not recur.

Fabrication of spherical or conical geometry in a small soft seat usually is accomplished by pressurizing the poppet to form the seat. A typical forming process for Teflon seats comprises pressurizing the poppet to induce a seat stress of approximately 900 to 2300 psi (6.2 to 15.9 MN/m<sup>2</sup>) while maintaining a temperature of 200°F (367 K) for 1 hour to accelerate the forming process. The actual value for seat stress depends on the seal configuration and is established by experiment.

Surface finish of the sealing surfaces of soft seats or soft poppets is not as critical as that of hard seats, because soft seats and soft poppets yield easily; however, surface finishes in the range of 8 to 32  $\mu$  in. (0.20 to 0.81  $\mu$ m) AA are required.

### 2.1.2.2 SOFT SEAL RETENTION

The method of attaching soft seals to the valve body must prevent or limit distortion and creep and preclude dislodging. Typical methods of attachment include vulcanizing of elastomeric seals; high-temperature bonding of plastics to steel; epoxy bonding, spinning, or swaging of a metal lip over the seat or over a retainer that clamps the soft seal; and

mechanical attachments such as bolted, threaded, or welded retainers. Mechanical attachments are very reliable, but their use is restricted to relatively large valves.

The low fabrication cost of vulcanized or bonded seals is a major advantage. The process, however, involves manual operations that cannot be inspected readily, so the integrity of the bond may be questionable. Bonded Kel-F seals failed in a gas-generator valve that was exposed to unexpected hot-gas blowback during engine ignition sequence (ref. 31). Bonded rubber seals in a valve on a large booster engine failed at the bond (ref. 32) and also tore (ref. 33) during engine static firing. A poor bond was suspected as the cause of these failures. The failures could not be repeated in the laboratory. The valve was redesigned as a hard-on-hard unit to improve reliability.

Several different methods of improving bond reliability have been effective. The J-2 start-tank discharge valve has a bonded Teflon seat and uses a welded-on retainer to give high sealing reliability. Inspection of viewable detail parts and destructive tests of sample parts are performed to check bond strength. The bonded Teflon seal in the poppet of the relief valve of the recirculation valve for the S-II vehicle is subjected to 25 thermal-shock cycles to verify the integrity of the bond. The poppet is submerged in liquid nitrogen until temperature stabilized and then immersed in water. No failures of this bond have been reported. Another test method that has successfully detected voids in a Kel-F bond is to expose it to a high-pressure pneumatic gas, rapidly vent the gas, and visually inspect the bond.

Techniques used to retain Teflon seats in small propellant valves for reaction control systems include spinning a metal lip over the Teflon seal. This process is low in cost, but it is difficult to control the squeeze on the seal. Another type of design controls the squeeze on the Teflon by installing the seal in a closely dimensioned groove in a carrier, installing a retainer that bottoms out metal-to-metal, and then spinning an edge of the carrier over to retain the retainer. Press-fit retainers also are used.

The major problems associated with all of these types of retainers for Teflon seals in small propellant valves are as follows:

- Excessive manufacturing costs due to extremely tight tolerances.
- Voids between the seal and its retainer.
- Creep of the seat, which results in zero seal height.
- The necessity for almost 100-percent confinement of the seal to minimize creep.
- Dimensional changes of the seal during exposure to high temperature.

The (partial) resolution of these problems has been largely on a trial-and-error basis. Careful analyses of the tolerance extremes aid significantly in minimizing voids and restricting creep.

Teflon seals retained by press fit or shrink fit generally must be stress relieved to minimize detrimental creep resulting from installation stresses. In a small valve in a reaction control system, pressure drop increased because heat from an assembly weld that was performed while the valve was energized stress relieved the Teflon seat and caused the seat to elongate (ref. 24). Procedures were revised to limit heat input to the valve and to institute a stress-relief process.

In the Intelsat III propulsion system thruster, elastomeric seals in valves in hydrazine service are retained solely by a shrink-fitted mechanical retainer (fig. 14); bonding was considered,

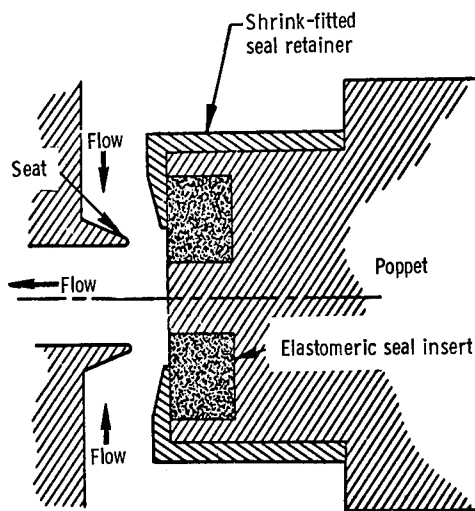


Figure 14. - Cross-section sketch of mechanically retained elastomeric seal.

but the elastomer, chosen for compatibility with hydrazine, did not bond well. Other elastomeric seals currently used in valves in service with 50 percent hydrazine – 50 percent unsymmetrical dimethylhydrazine are installed with a bonding agent and then vulcanized in place. The practice has been to use bonded seals when the propellant-compatible elastomer bonds well. Both the mechanically retained and bonded configurations have performed satisfactorily.

Pressure differentials in a fluid system produce flow forces that can cause poppet-seal fracture or ejection. This type of failure generally has occurred in small valves where space did not permit use of bolted or threaded retainers and where the soft seal was located in the poppet; this seal location is more likely to be subjected to large pressure differentials. Seal retainer failures occurred on a valve that utilized a soft aluminum seal ring shrink fitted onto

a heat-treated aluminum poppet. During valve closing, the seal ring separated from the poppet. Analysis indicated that separation occurred because the top of the seal ring was exposed to line pressure while the bottom of the seal ring was exposed to a very low pressure in the throttling region as the valve approached the closed position (fig. 15). This

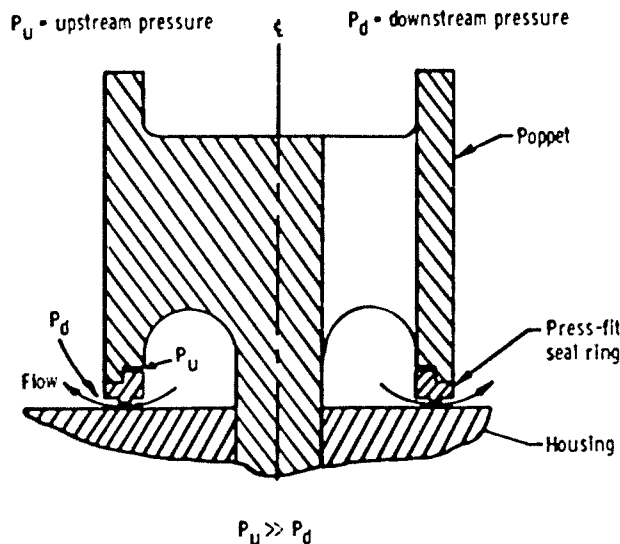


Figure 15. - Cross-section sketch illustrating flow pressure forces on seal ring.

pressure differential produced a large unbalanced force that ejected the seal ring from the poppet. The design was modified to provide a seal ring sealed on the high-pressure side so that no unbalanced expulsion forces on the seal ring were generated. The modification also increased the force retaining the seal in its retainer.

### 2.1.2.3 STATIC SEAL STRESS

Choice of static seal stress (steady-state-seating applied load divided by sealing-surface area) is critical to effect a seal and to prevent excessive deformation of a plastic seat or seal. High stresses over long-term storage periods may cause permanent deformation and result in leakage. The stress generally is well below the material yield strength; a seat stress of approximately 500 psi ( $3.4 \text{ MN/m}^2$ ) and 3000 psi ( $20.7 \text{ MN/m}^2$ ) is typical for Teflon and Kel-F, respectively. A seat stress of 1500 psi ( $10.3 \text{ MN/m}^2$ ) was used successfully on a storable-propellant valve utilizing a Teflon poppet, but it was necessary to limit the number of checkout cycles to maintain adequate operational life (ref. 15).

Plastics undergo a large increase in yield strength and in modulus of elasticity when cooled to cryogenic temperatures. Therefore, the seat loads for plastic seals in cryogenic service are set at levels that will produce high stresses relative to room-temperature mechanical

properties of the plastic, so that at cryogenic operating temperature the stress is sufficient to produce the required deflection in the stronger material and achieve sealing.

Stress in an elastomeric seat usually is not critical unless so much compressive yielding occurs that the pressure drop-vs-stroke characteristics of the valving unit change substantially. Shear stress between the compressive loaded section and the nonloaded section may be critical. The stress analysis is an approximation, because of the general lack of definitive material properties at the specific operating conditions. Cycling tests of the seal configuration are performed to verify its performance.

#### **2.1.2.4 TRANSIENT SEAL STRESS**

Impact forces increase the level of seat stress. Repeated cycling may cause seat yielding, change in poppet stroke, and hence change in poppet actuation time. Also, the seal may crack and leak. These effects are especially troublesome in valves used in reaction control systems, where repeatability of the engine impulse bit is critical, and any valve leakage can compromise the mission. Impact forces are limited by keeping the poppet weight to a minimum and by providing positive stops at the end of travel. Also, checkout operations are controlled to ensure that the valve is not rapid cycled in the dry mode, where lack of propellant damping can increase stresses.

#### **2.1.2.5 ALIGNMENT**

Alignment in a hard-on-soft valve is not quite as critical as in a hard-on-hard valve (sec. 2.1.1.4), because the soft seal will deflect locally to compensate for misalignment. The hard-on-soft valve typically, however, incorporates a guided poppet, which complicates alignment.

Normally, alignment is achieved by control of the valve tolerance; in a flat-seat geometry, for example, the normality of the poppet disk sealing surface to the shaft and the normality of the seat to the shaft bore are closely controlled so that the clearance between the shaft and its bore will permit the poppet to self align. Spherical-geometry poppets generally are self aligning. Axial tilt of conical poppets probably is not as critical for soft-seat valves as it is for hard-seat valves, because the lower interfacial friction of a soft seat is less and probably permits self-alignment if guide clearance is adequate.

## **2.2 BUTTERFLY DISK VALVING UNIT**

The butterfly valving unit consists of a relatively flat valving element (usually referred to as a disk) that may be rotated to control flow through the valve body. The disk can rotate on a



single-piece shaft or on a two-piece shaft that supports the disk on both sides; the shafts may be integral with the disk for greater rigidity. The center of rotation of the disk usually is offset to allow the disk to rotate off the primary seal. For low leakage, plastic lip seals in the housing usually are employed. Valve leakage due to loss of seal interference with the disk and seal breakage when the valve closes under flow conditions are basic problems with the butterfly valving unit. For special environments, metallic seals have been utilized; however, only limited test experience is available and no operational valves employ metallic disk seals.

## 2.2.1 Geometry and Surface Finish

Butterfly disks have been designed with spherical and conical shapes (fig. 16); both geometries have been evaluated for cost and sealing capability and cycle life. The spherical shape (fig. 16(a)) theoretically is best but is more expensive to fabricate. The conical shape (fig. 16(b)) is more susceptible to leakage, possibly because of greater wear.

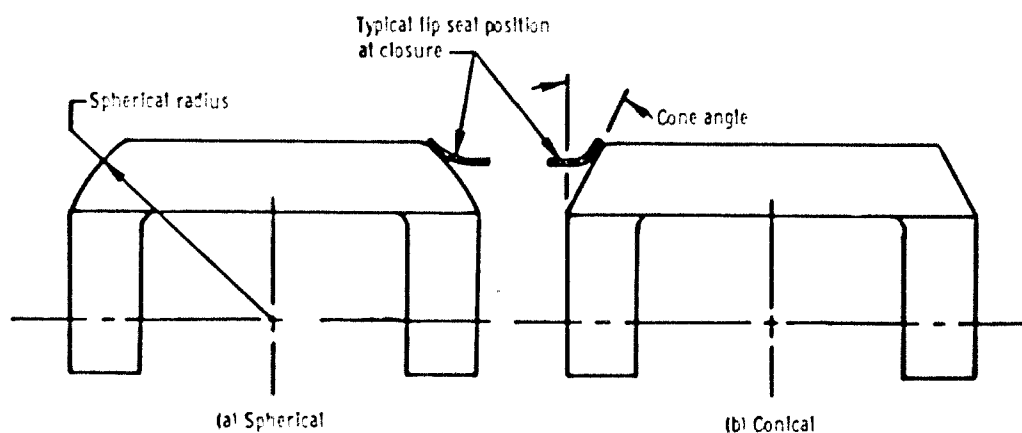


Figure 16. - Sketches of butterfly disk shapes and seal positions.

Breaking or folding under of the lip seal (fig. 17) when the disk closed rapidly under high-pressure flow occurred on the early butterfly valves used on the Atlas propulsion system; a similar condition occurred recently on advanced J-2 engines. The design was modified by contouring the entering edge of the disk to minimize hitting of the seal by the disk during closure.

Early designs also used a contoured lip in the housing, a concept intended to support the seal and prevent seal folding. In later designs, the contoured backup ring has been eliminated because it is difficult to machine to the tolerances that will provide support and yet not hold the seal off the disk under the tolerance extremes. Seal resistance to deflection caused

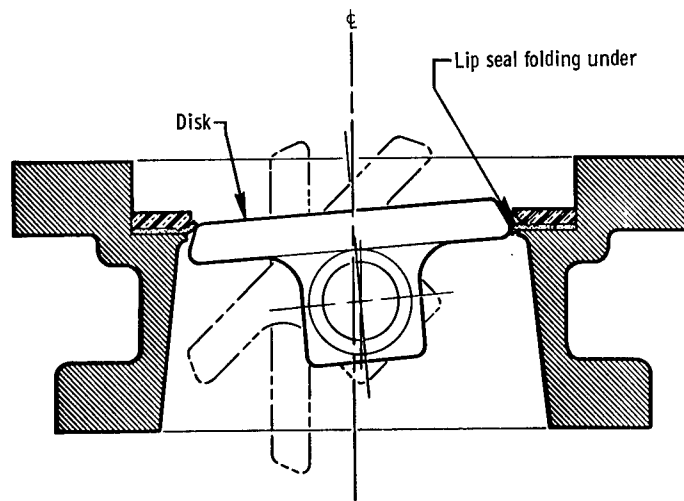


Figure 17. - Cross-section drawing illustrating lip seal folding under at closure.

by flow forces is critical, as it prevents breaking or folding under of the plastic lip seal. Excessive rigidity will cause increased friction and wear during actuation cycling. Excessive bending will result from too great a lip seal height (fig. 18) in relation to thickness; bending deflection is reduced by increasing the seal thickness. To prevent seal breaking or folding, materials used are evaluated critically.

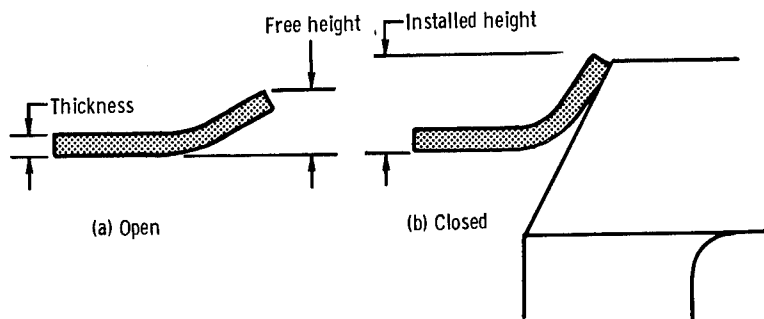


Figure 18. - Sketches showing butterfly lip seal height and thickness.

On the 8-in. (20 cm) butterfly prevalves used on the Saturn S-II stage, seal breaking was aggravated by engine conditioning procedures. The normally circular sealing surface of the lip was distorted to an elliptical shape while the valve was in its normally open position at room temperature. When the engine was preconditioned with cryogenic propellants, the initial tanking procedures placed the valve in the open position, which stabilized the seal in

the distorted position. When the valve subsequently was closed, the seal failed because of interference between the disk and lip seal at the minor axis of the ellipse (at right angles to the disk shaft). The tanking procedures were changed to allow the valve to be chilled in the closed position, thus ensuring that the seal was stabilized in a circular shape.

Many instances of leakage have occurred when the disk in the open position contacted the seal locally and indented it. This condition is aggravated when the open valve is subjected to high temperature (e.g., during an environmental test or drying process) or when an open valve is exposed suddenly to a cryogenic propellant. The problem is alleviated by offsetting the disk so that the disk in the open position does not contact the seat.

A radius on the sealing edge of the Kel-F seals has proved effective in reducing wear during cycling. Good surface finishes of the disk and sealing surfaces of the seal are required to achieve good sealing and to prevent wear.

## **2.2.2 Seal Retention**

### **2.2.2.1 RETAINER CONFIGURATION AND LOADS**

The seal retainer must retain the seal reliably and transmit sufficient load to seal the seat against the housing under varied pressure and temperature conditions without introducing creep of the seal flange under long-term storage and extreme temperature conditions. In addition, the seal retainer usually is designed for low cost and ease of assembly.

Typically, bolted-on retainers are used because of the reliability of threaded fasteners and the relatively high degree of control of applied load. The threaded fasteners must be locked so that they withstand the loosening forces of vibration and flow in the specific application; locking inserts in the housing have been used successfully. Serrations on the retainer are used to give a high unit load for sealing of the flange. A special tool generally is used to seat the serrations evenly, because use of the threaded fasteners may cock the retainer. Snap-ring retainers were used on early model valves, but were found to be difficult to assemble; also, the applied load was marginal and variable.

### **2.2.2.2 THERMAL CONTRACTION OF RETAINER BOLTS**

Leakage has occurred in valves in cryogenic service because the thermal contraction of the disk seal exceeded that of the corresponding length of seal retainer; the differential contraction decreased the bolt loads and permitted leakage. The designs were modified to change the thermal-contraction characteristics at the temperature-stabilized condition.

## 2.2.3 Interference of Disk and Seal

The lip seal geometry commonly used for butterfly valve seats is designed to be pressure assisted so that the valve will seal better at high pressures. When the inlet pressure to the valve is very low, interference between the disk and its seat to provide an initial sealing load is required. Because of the large thermal contractions in a valve operating at cryogenic temperature, the amount of interference is critical. For example, a large, low-pressure valve used with cryogenic propellants leaked excessively at cryogenic temperatures because the disk and shafts apparently deflected more than the soft Kel-F seal, thereby reducing the interference (ref. 34). The valve disk and shafts were increased in thickness to reduce the deflection.

Too much interference can result in seal breakage or in excessive wear when the valve is cycled at room temperature; the interference therefore is kept to the minimum allowable. Experience with similar valves provides the designer with a feel for the required interference on a new valve design. However, development testing is required to ensure adequate cycle life and sealing performance under both room-temperature and cryogenic conditions.

## 2.2.4 Alignment of Disk and Seal

To obtain good sealing and to prevent wear, the seat and the disk are designed to be concentric; tilt of the disk, however, is not critical, because of the spherical surface of the disk.

A typical method of alignment includes use of the retainer installation tool to align the seal with the disk during assembly and to maintain that alignment. Another method involves permitting the disk to float back and forth on its shaft so that it is self aligning in the fully assembled state. These two methods apply when there are no unbalanced pressure forces tending to load the disk against one side of the seat. Current designs balance out the pressure forces by making the shaft areas sealed by shaft seals equal in area and 180° apart so that the two forces cancel each other.

## 2.3 BALL VALVING UNIT

Characteristics of the ball valving unit are presented in reference 4, sections 5.2.3 and 6.2.3.2. Seal wear generally is the major problem for ball valving units. Flakes tend to wear off the seal and remain in the valve. When they can be tolerated in the system, lubricants are added to the ball/seal interface to reduce wear during dry cycling. Fast cycling tends to

generate heat, which aggravates the seal wear. Wear during dry cycling has been reduced by increasing the actuation time greatly (ref. 35) or by using a slow cycling rate with standard actuation times.

### 2.3.1 Geometry and Surface Finish

The ball valve element in noncritical small valves may be a sphere with a hole or flow bore through it. To reduce size and weight in large valves ( $>2$  in. [5.1 cm]), the element is designed as a truncated sphere (fig. 3), or in very large valves, where only about 1/4 of the total sphere area is used, as a visor (fig. 19). In the propellant shutoff valve for the LMDE (ref. 35), weight was reduced by modifying the ball as shown in figure 20.

One factor in selection of ball size for a given flow-bore size is the angle of rotation of the ball; typically, this angle is  $90^\circ$ . All dimensional tolerances affecting the relationship of the seal to the flow bore must be checked to ensure that the seal makes a full  $360^\circ$  contact with the ball in the open position; otherwise, uneven loadings may result in damage to the seal. For example, oxidizer leakage in the LMDE propellant valve was found to have been caused by creases in the soft seal (ref. 35). The valve had been left in a partially open position for several hours prior to closure; this condition brought the ball flow bore into line contact with the seal. The problem was corrected by a change in procedures that ensured that the ball was not left partially open.

In a large bipropellant valve (1.5 in. [3.81 cm] – oxidizer flow bore and 1.75 in. [4.45 cm] – fuel flow bore) used on the F-1 engine, cracking of the fuel-ball shaft occurred because of the large inertia of the fuel ball and high-speed actuation. The inertia was reduced by using a hollow ball fabricated by welding two sections together. The hollow ball, however, greatly increased the pressure drop of the valve; bellows inserts and streamlining of the passage were necessary to reduce the pressure loss.

The ball valve seal may be located either upstream or downstream of the ball. The upstream location is advantageous because it eliminates one external leak path when the valve is closed. Ball valves used in man-rated missions sometimes use both an upstream and a downstream seal to improve sealing reliability. This practice, however, requires provision for venting between the two seals. Venting usually is provided by designing the upstream seal to relieve if pressurized in the reverse direction. The intersection of the edge of the flow bore with the surface of the ball is critical, because seal wear will occur if the corner is too sharp or the surface finish is rough.

The dynamic sealing surface of the seal (the surface that contacts the ball) sometimes is machined on an angle or sharp corner, because it is difficult to machine a spherical surface on the seat. Actuation of the valve assembly wears down the line contact surface of the seal until it forms a spherical surface of sufficient area to reduce stresses to the level at which

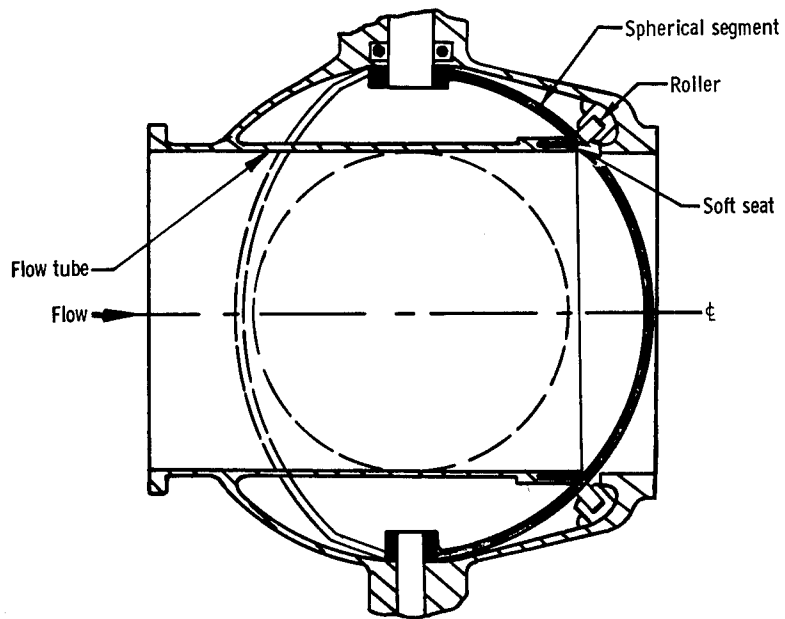


Figure 19. - Cross-section sketch of visor valve assembly.

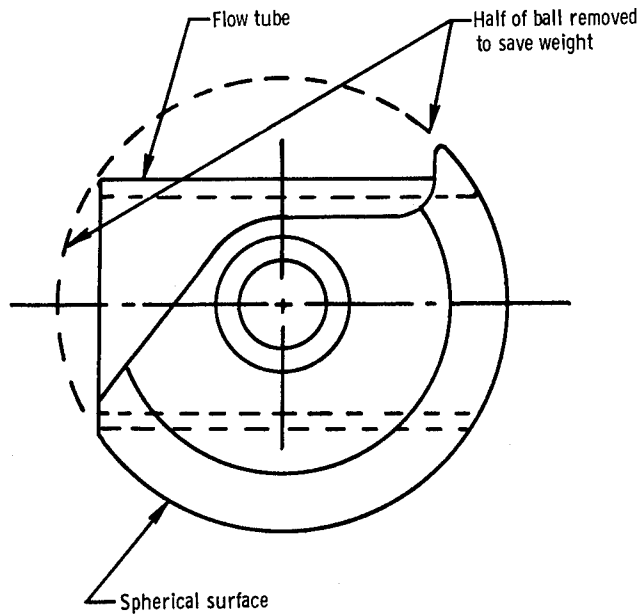


Figure 20. - Sketch of rotor design for LMDE ball valve.

there is no further significant wear. Alternately, the spherical surface of the seal is machined and then burnished by using a fixture with a ball (no flow bore) that rotates under a controlled load to wear in the seal (ref. 30). This process eliminates accumulation in the valve of particles worn off from the seal and produces a more uniform wear pattern. Actuation of the ball with the flow bore produces an unsymmetrical wear pattern, since contact area of the seal changes during valve actuation whereas the installation load is constant.

### 2.3.2 Seal Retention

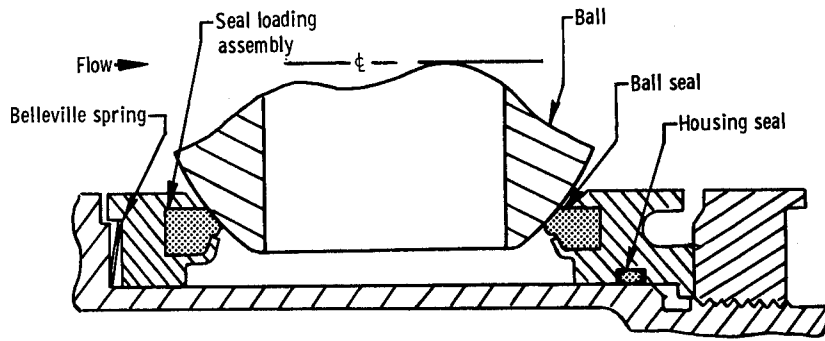
The seal usually is mounted so it will be pressure assisted; an initial preload is required to seal when the line pressure is low. Figure 21 illustrates three methods for mounting ball valve seals. A Belleville spring generally is used to provide the preload in low-pressure valves operating in the 100- to 300-psi (0.69 to 2.07 MN/m<sup>2</sup>) range. This configuration (fig. 21(a)) requires a seal between the retainer and the valve housing; in some designs, this seal is a dynamic seal and requires careful attention to design details. The primary seal in the LMDE ball valve incorporates a static seal to seal the housing interface (fig. 21(b)). A bellows-mounted seal (fig. 21(c)) generally is used in high-pressure valves, because the bellows gives an initial preload and in addition limits the pressure load because of the differential area characteristics of bellows. Bellows are comparatively expensive, but also have the advantage that they provide an external seal.

If the pressure load in a high-pressure valve is not limited, the excessive loads cause seal wear. Balancing of the loads, therefore, is critical in high-pressure valves. Determination of the effective diameter of the bellows and the diameters at which the seal seals against the ball and against its retainer is necessary if the seal load is to be established at a desired level. Deflection of the seal under pressure may change the point of sealing and upset the load balance. Vent holes are provided to vent any fluid leakage past the seal/retainer interface to prevent a pressure buildup that might separate the seal from its retainer and cause gross leakage.

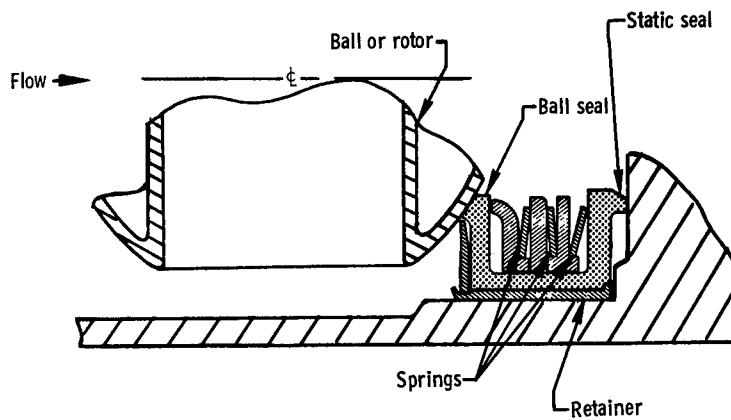
A bellows-mounted seal in the idle-mode ball valve on the J-2 engine leaked because inlet pressure tended to compress the bellows and permit a leak path between the bottom of the seal and its retainer. The seal was redesigned to incorporate a bevel on the bottom of the seal so that the point of sealing between the seal and retainer was moved outboard and the separating load was reduced.

### 2.3.3 Seat Stress

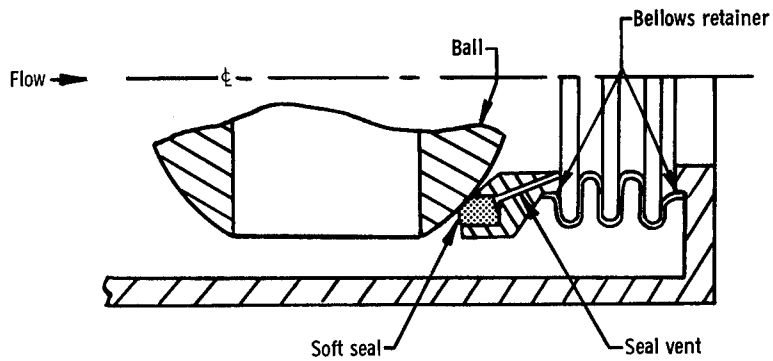
The amount of seat load of a ball valve is critical in preventing leakage and wear. Too low a seat load will result in leakage at the low pressures used for system checkout and test.



(a) Ball valve seal loaded by a Belleville spring



(b) Ball valve seal with internal springs and integral static seal (LMDE configuration)



(c) Bellows-mounted ball valve seal

Figure 21. - Cross-section drawings of three methods for mounting ball valve seals.



Determination of the maximum allowable load on the basis of the applied stress and the yield strength of the material is difficult when the initial contact area of the seat is a line. In this case, loads generally are determined empirically. When the initial contact area is machined to a finite area, a seat stress of 190 psi (1.31 MN/m<sup>2</sup>) has been effective with a glass-filled-Teflon seat. Seat stress with unfilled Teflon and Kel-F generally is kept below this level to avoid cold flow.

The ball valve on the gas generator on the F-1 engine exhibited wear during room-temperature cycling; there was little wear during cycling at cryogenic temperature, apparently because of the higher yield strength of the Kel-F seat at the low temperature. Analysis of the failures of the production valves showed that the seal loads of the valves with excessive leakage were all at the high end of the allowable load range. The maximum loads were reduced by decreasing the pressure-actuated differential area of the bellows used to load the seals; this action increased the cycle life of the ball seal at room temperature yet still provided a load at cryogenic temperature adequate for sealing.

### 2.3.4 Alignment

The spherical shape of the ball compensates for angular misalignment but not for an offset between the axes of the ball and the seat. Misalignment results in seal wear. It is theorized that a bellows may compensate for some misalignment by deflecting sideways, but this behavior has not been confirmed and probably does not occur at high pressures, where the bellows becomes very rigid. Any sideways deflection will, however, result in a detrimental load.

On ball valves, either the ball or the seal may be self aligning. On the valves on the Lunar Module Ascent Engine (LMAE), the ball floats between the redundant upstream and downstream seals. The square-end drive shafts engage the balls but provide no constraint. No special assembly techniques are required to ensure alignment. On the LMDE propellant valves, the ball seal cartridge is designed to move laterally in the body recess cavity to center itself on the spherical rotors during assembly.

The idle-mode ball valve on the J-2 engine has one shaft sealed and the other shaft exposed to line pressure; both seal wear and actuator-linkage binding were attributed to a lateral unbalanced load. The valve incorporated a thrust bearing to react against the unbalanced load, but the bearing was ineffective because the tolerance stackup left a gap between the bearing surface of the ball and the thrust bearing. The stackup problem was solved by adding shims between the ball and the thrust bearing until the ball was closely aligned with the seat.

## 2.4 BLADE VALVING UNIT

The blade valve (fig. 4) is considered favorably when inlet pressures are low (200 psi [1.4 MN/m<sup>2</sup>] or less) and a very short valve length is required. A cross section of a blade valve assembly is shown in figure 22.

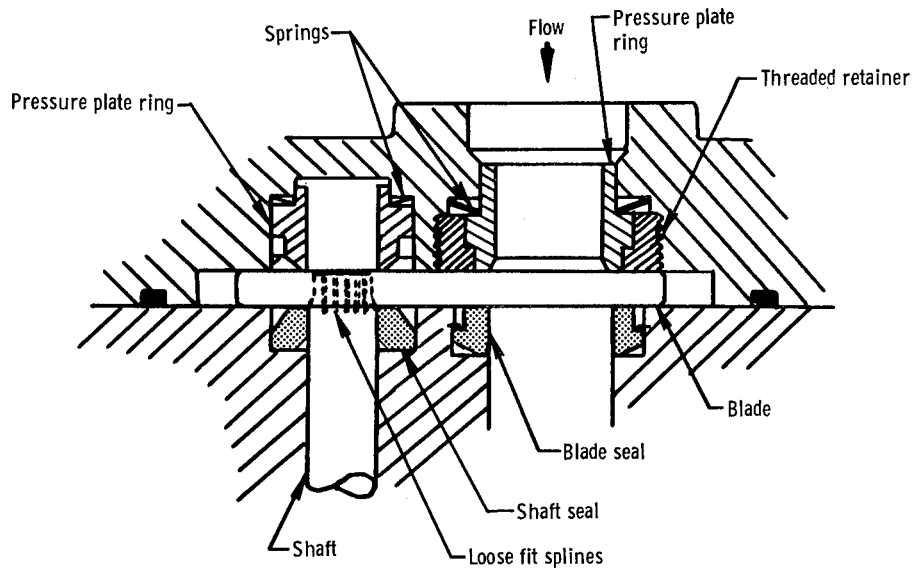


Figure 22. - Cross-section sketch of blade valve assembly.

### 2.4.1 Geometry and Surface Finish

Blade valving units have been designed with a solid blade that swings or pivots on and off the flow passage (fig. 4), and with a blade containing a hole that mates with the flow passage when the blade pivots (fig. 23). The former design suffered from scuffing and wear of the flow-passage seal as the blade moved off and on; the latter design maintains continuous contact with the seal and thereby achieves relatively long cycle life. For off-and-on blade contact designs, an 8 to 16  $\mu$  in. (0.20 to 0.41  $\mu$ m) AA surface finish on the blade flat and on the edge radius that first contacts the seal has been employed. Similar finishes have been used on the blade flat and on the hole edge radius of continuous-contact designs. Both configurations utilize a blade flatness dimension of 0.001 in. (25.4  $\mu$ m) TIR. A soft-seat surface finish of 32  $\mu$  in. (0.81  $\mu$ m) AA has been effective for both designs.

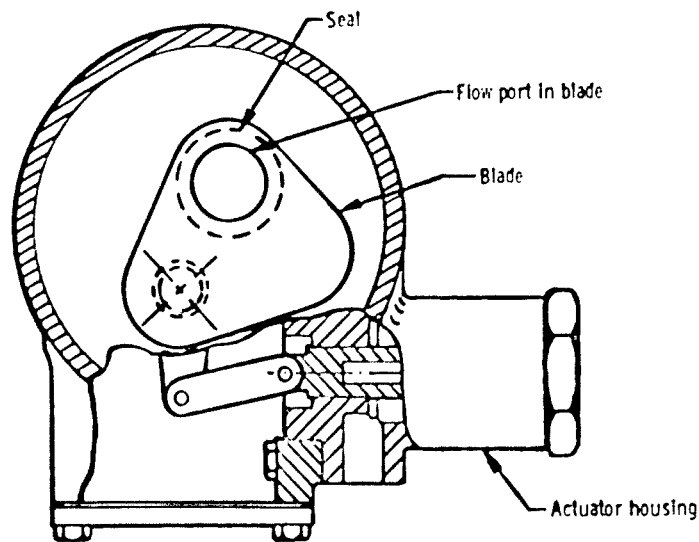


Figure 23. - Cutaway drawing of a continuous-contact rotary-slide blade valve.

## 2.4.2 Seal Retention

The seal typically is a machined piece of plastic that is pressed into the housing bore and retained by a snap ring or threaded retainer. In the Atlas gas-generator blade valve, the seal incorporated a silicone rubber O-ring on the outer diameter to seal against the housing. The O-rings sometimes leaked. The O-ring was eliminated and the design was modified to incorporate a narrow sealing surface on the bottom of the seal; the leakage stopped.

## 2.4.3 Seal Stress

The seal stress for a blade valve is similar to that for other soft seal valves (sec. 2.1.2.3). On the Atlas gas-generator blade valve noted above, a spring-loaded pressure plate is used to press the blade against the seal, as shown in the cross section in figure 22.

## 2.4.4 Alignment

Parallelism between the seal and blade is critical to good sealing and wear life characteristics. Typically, the blade is loosely engaged to a splined shaft so that it is self aligning.

## 2.5 SLEEVE VALVING UNIT

### 2.5.1 Rotary Sleeve Unit

The rotary sleeve valve (fig. 6) consists of a slotted spool that is rotated to align with similar parts in the body. Since the rotary sleeve is not readily adaptable to incorporation of seals, the valve is used for throttling applications rather than for shutoff. The major problem with this valve has been variation in operating torque due to friction.

#### 2.5.1.1 GEOMETRY AND SURFACE FINISH

In a typical application, the mixture-ratio control valve on the J-2 engine, the gate and body spacer assembly have a nominal diametral clearance of 0.0037 in. (94  $\mu\text{m}$ ). The cylindrical surfaces have a 16  $\mu$  in. (0.41  $\mu\text{m}$ ) AA finish.

#### 2.5.1.2 ALIGNMENT

Alignment considerations on the rotary sleeve valve include an evaluation of the adequacy of the diametral clearance to compensate for minor misalignments inherent in machining and assembly and for dimensional changes due to temperature extremes to which the assembly is exposed. On the J-2 engine, for example, a 2-in (5.1 cm) rotary sleeve is used to throttle liquid-oxygen flow. The diametral clearance between the sleeve outer diameter and the mating inner diameter of the spacer assembly (another sleeve) was sized to minimize leakage in the closed position. The clearance was further reduced by differential thermal contraction of the aluminum housing and the steel spacer assembly at liquid-oxygen temperature. The valve opened erratically when subjected to engine operating pressures, because the thin spacer assembly deflected under the high pressure differentials and rubbed against the sleeve. The diametral clearance of the sleeve to spacer assembly was increased. The absence of any erratic actuation characteristics was verified by testing the valve at high pressure differentials (in the closed position) at cryogenic temperature.

## 2.5.2 Linear Sleeve Unit

The linear sleeve valving unit is defined in this monograph as a sleeve that retracts from the primary seal to open and makes sliding contact with the primary seal to shut off (fig. 5). The major difficulty with the linear sleeve is the on-off contact of the sleeve with the primary seal; this kind of contact tends to abrade the seal and produce leakage. The sleeve valving unit usually is partially balanced by incorporating another seal on the sleeve outer diameter. The only recent application of a linear sleeve valve was the 10-in. (25.4 cm) nominal-line-size valve (fig. 5) designed for M-1 thrust chamber testing.

Several shutoff seal configurations were tested during development of the M-1 sleeve valve (refs. 36 and 37). Sealing surfaces of the sleeve were finished to  $4 \mu$  in. ( $0.10 \mu\text{m}$ ) AA, and the leading edge was beveled to facilitate entry of the sleeve into the main shutoff seal. Alignment of the sleeve and main shutoff seal was maintained by holding concentric the seal and sleeve installation diameters in the housing and holding concentric the sleeve beveled leading edge and outer diameter. Tests of the M-1 sleeve valve showed that leakage ranged from 30,000 to 600,000 sech at liquid-nitrogen temperature. Diametral interference of the main shutoff seal was varied from 0.080 to 0.150 in. (2.03 to 3.81 mm) on these tests. As interference was increased, seal leakage was reduced; however, friction increased, and seal breakage occurred more frequently.

## 2.6 CYLINDRICAL SLIDE VALVING UNIT

A cylindrical slide valve (fig. 24) typically incorporates a translating solid cylinder with an annular flow passage or a flow passage at right angles to the longitudinal axis of the cylinder or slide. In applications with relatively high allowable leakage rate (e.g., flow routing or pressure sequencing), there are no internal seals, and leakage is a function of the diametral clearance and the length of the land; this configuration is designated as a spool valving unit. A configuration that employs a lip seal for shutoff sealing has been used in one application.

### 2.6.1 Spool Unit

The spool valve configuration is similar to that of a servovalve. Reference 4, section 6.2.3.9, and reference 38 present information on spool valve design. Because there is no internal seal, there are no considerations involving seal stress, retention, and so forth. Alignment considerations include only an evaluation of the adequacy of the clearance to compensate for the minor misalignments inherent in machining operations.

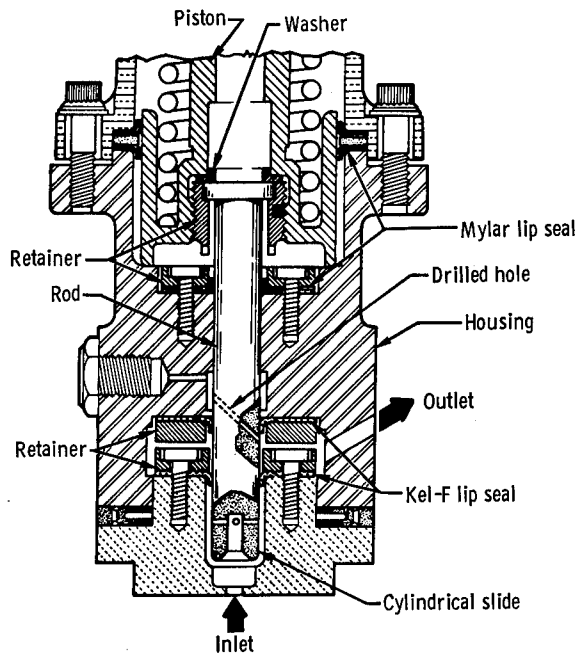


Figure 24. - Cross-section drawing of cylindrical slide valve with soft seals (sequence valve on J-2 engine).

### 2.6.1.1 GEOMETRY AND SURFACE FINISH

The diametral clearance between the spool outer diameter and its mating bore is critical because the clearance must be small enough to restrict leakage as required but must be large enough to ensure smooth actuation. The diametral clearance between the spool and housing is dependent on allowable spool tilt in the bore and allowable leakage flow. Since spool valves are not used for shutoff applications but rather for flow routing, clearances are based on allowable spool tilt rather than on allowable leakage. Excessive spool tilt can cause large lateral forces and friction. The L/D ratio is also an important parameter in smooth valve operation, since clearance and the L/D ratio are factors that directly affect spool tilt. Surface finish also is critical because it affects smoothness of operation.

An 0.5-in. (13 mm) four-way cylindrical slide valving unit used on the Titan II engine has a diametral clearance of 0.0003 to 0.0007 in. (7.6 to 17.8  $\mu\text{m}$ ) (ref. 24). Specified leakage rate is a maximum of 5 cu. in. (82  $\text{cm}^3$ ) of water per minute at a pressure differential of 1000 psi (6.895  $\text{MN}/\text{m}^2$ ). The slide and its mating sleeve are finished to 8  $\mu$  in. (0.20  $\mu\text{m}$ ) AA surfaces.

The 1-in. (2.54 cm) four-way valve that controls the flow of RP-1 fuel to the actuators of the main oxidizer, main fuel, and gas-generator valves on the F-1 engine incorporates a hollow spool (fig. 25). The hollow spool gives the valve some of the characteristics of a

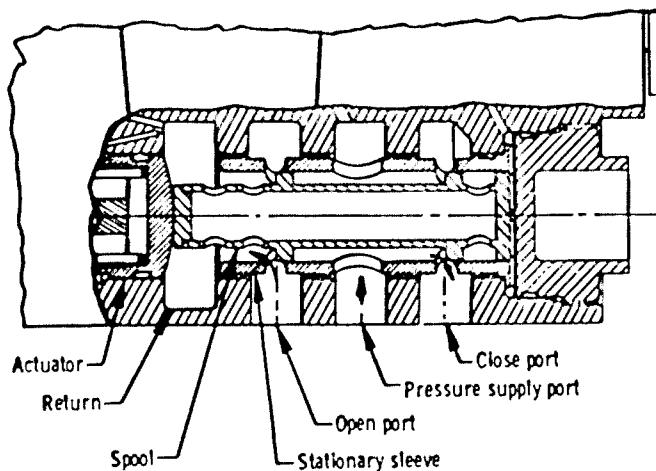


Figure 25. - Cutaway schematic of hollow-spool valving unit (four-way control valve on F-1 engine).

sleeve valve, but functionally the valve is identical to a spool valve. This valve has a poppet-to-stationary sleeve diametral clearance of 0.0001 to 0.0002 in. (2.54 to 5.08  $\mu\text{m}$ ); a poppet and a stationary sleeve are maintained as a matched set. Because of the runout of the stationary sleeve inner diameter and poppet outer diameter in this comparatively large valve, it is necessary to hold the clearance close to the maximum allowable in order to achieve smooth actuation. The surface finishes of the poppet outer diameter and sleeve inner diameter are 4  $\mu$  in. (0.10  $\mu\text{m}$ ) AA.

## 2.6.2 Cylindrical Unit with Internal Seal

Plastic lip seals are used in one cylindrical valve configuration. These seals are located in the valve housing and seal circular flow passages (holes) in the cylindrical slide (fig. 24). Actuation of the slide moves the drilled holes past the lip seal, an action that increases the possibility of damage to the lip seal.

Alignment considerations are not critical, because the plastic lip seals deflect to compensate for any eccentricity between the slide and the lip seals. A valve of this type used in the J-2

engine has a flow diameter of approximately 0.20 in. (5.08 mm); specified maximum leakage is 60,000 scch of helium at 400 psi (2.8 MN/m<sup>2</sup>) at -320°F (78 K).

### **2.6.2.1 GEOMETRY AND SURFACE FINISH**

The drilled hole through the cylindrical slide must have radii of intersections blended with the outer surface to avoid damage to the lip seal. The ratio of slide diameter to hole diameter also is critical to wear of the soft lip seal. Design information on this type of valve is meager because applications are few.

### **2.6.2.2 SEAL RETENTION**

The lip seals are retained with a bolted retainer that utilizes a single serration to give a high unit load for sealing and seal retention. Design of bolted retainers for lip seals is discussed in section 2.2.2.1.

## **2.7 GENERAL CONSIDERATIONS IN VALVING UNIT DESIGN**

The designer must consider the effects of vibration and contamination in the design of all types of valving units, especially the hard-seat poppet valving unit. The effects of vibration and particulate contamination on valve assemblies are discussed in reference 1.

### **2.7.1 Vibration**

Vibration can result in seat or seal wear by causing the valving element to rub on the seat or seal. A poppet may vibrate in the direction of its motion and cause leakage. A poppet valving element with large guide clearances may vibrate in a direction perpendicular to the poppet axis and cause scrubbing and wear of the seat. Vibration analyses are very difficult; usually, vibration tests are performed to determine if there are potential problems.

Fretting corrosion, a type of corrosion that occurs under small-amplitude oscillatory motion under load, can cause accelerated seat wear of a poppet valving unit. Fretting corrosion is decreased by increasing load to prevent slip, by mechanically restricting slip, by increasing hardness, or by providing lubrication. In an aerospace valve design, the loads are fixed by actuator size, the materials are already very hard, and lubrication retention over the required cycle life is not practical; the usual solution therefore is to restrict slip mechanically. Two examples illustrate the kinds of design solutions that have been employed.



A four-way solenoid valve used on the J-2 engine incorporated 0.001-in. (25.4  $\mu\text{m}$ )-thick Elgiloy disks that sealed by deflecting against 440C seats. The valve was found to have substantially increased leakage after a vibration test (refs. 26 and 27). Disassembly of the valve indicated fretting corrosion had degraded the sealing surfaces; apparently the guide bearing clearances permitted too much lateral motion. The valve was redesigned to incorporate flexure-disk mounting of the poppet shaft (fig. 10). The flexure disk substantially restricted lateral motion of the poppet shaft and prevented the fretting corrosion. Use of flexure disks is restricted to short-stroke applications. A method of stress analysis of flexures is presented in reference 29.

The original four-spoke sleeve of the M-1 sleeve valve had a low fundamental natural frequency that was excited by flow testing in water; the resonant vibration caused extensive damage to the lip seals and housing contact area between the four spokes. The sleeve was changed to an eight-spoke configuration, but this change did not result in the anticipated improvement. The nose seal was changed from a lip-seal type to a poppet type. This change significantly increased the vibration resistance of the sleeve valve and was recommended as the most acceptable configuration (ref. 36).

## 2.7.2 Contamination

### 2.7.2.1 CONTAMINATION TOLERANCE

Contamination is a primary cause of valve failure. Control of contamination becomes increasingly important with growing emphasis on decreased leakage rates for long-life missions and on use of lightweight, hard-seat poppet valves, because these valves are more susceptible to contamination-caused leakage than is a soft-seal valve. A valving unit with a rotating valving element such as a butterfly disk or a ball may tend to wipe a contaminant off the seal. The poppet valving element, however, will tend to trap a contaminant between the poppet and seat, the result being either damage to the sealing surfaces or creation of a gap that will permit leakage.

To provide the poppet valving element with a tolerance for contamination, seat load requirements are established high enough to produce sufficient load (overload) to plastically yield either a trapped particle or the sealing surfaces. For this overload analysis, the important factors are particle size, number of particles, and physical properties (particularly hardness) of the particles and of the mating poppet and seat. A thorough analysis of system contaminants is required to establish maximum particle size and material properties. Reference 4, section 6.2.2.14, provides data on sources of particles and size of wear particles in various environments. The poppet and seat material and hardness are also factors in setting overload, since particles trapped will be retained in material combinations such as 440C and copper, whereas with 440C poppet on 440C seat a particle trapped on any one

closure will be ejected when the valve is subsequently opened. A method for predicting the probability of trapping a particle during poppet valve closing is given in reference 39; also presented are criteria for design of contaminant-resistant poppet and seats. When trapped particles are ejected upon subsequent opening, the seat overload requirement is based on one trapped particle; when the material combination is such that particles are retained, the total number of particles accumulated after a given number of cycles is used to calculate the seat overload requirements. Methods of inelastic stress analysis are presented in reference 40. Although some load data on particle size and quantity for various poppet and seat material combinations are available in reference 39, an analysis of seat overload requirements must include static particle tests on model poppet and seat surfaces to provide load data for a specific closure design. Such an analysis gives only an approximation, but it provides valuable insight into the contamination problems in valve design, and thus makes it possible to avoid contamination-sensitive valves.

Valve stroke usually is very large compared with specified maximum particle size of contaminants. Small-capacity poppet-type valves may, however, have a stroke comparable to the particle size, and this condition can lead to problems. For example, a small pneumatic relief valve on the J-2 engine had a stroke of less than 0.001 in. (25.4  $\mu\text{m}$ ) (ref. 10). This valve was very sensitive to contamination. It was hypothesized that the valve did not open completely, but hinged about one side so that particles were trapped between the spherical poppet and its seat when the valve opened. The valve then leaked in the closed position.

### 2.7.2.2 CONTAMINATION CONTROL

References 4, 41, 42, and 43 present general information on the generation and control of contaminant particles.

Contamination often is introduced by operation of the valving unit itself, e.g., a poppet shaft rubbing on its guide. Flaking of platings and the mating of threads during the assembly process have been sources of contaminant particles in a valving unit. Poor quality platings have been at fault. In some cases, platings may be required to obtain hard surfaces that do not gall; however, use of plating in some reaction control systems has been banned because of the associated contamination problems. Rolled threads are preferred to machined threads because of better surface finish (less burrs) and higher strength. Generally, material selection avoids soft, like-on-like rubbing surfaces that tend to gall and generate large particles; a hard material produces smaller wear particles than does a soft material. Deburring of detail parts eliminates the generation of metallic contaminants that may result when a surface with a burr is operated or vibrated. For example, in a reaction control system, the number of failures of small propellant valves due to leakage decreased by 50 percent when "fine" burr removal procedures were initiated (ref. 24).

In small valves in critical applications, the possibility of failure from contaminants has been reduced by eliminating metal-to-metal sliding contacts within the fluid cavity of the valve. This contact may be avoided by mounting the poppet on a flexure disk (ref. 44), Belleville spring (ref. 45), or flexure tube. (Reference 1 includes a discussion of the flexure-tube mount for poppets.)

Effective contamination control also requires cleaning of all detail parts, as discussed in the section that follows. Detail parts are designed with a minimum of dead-end passages or capillary-size passages because such passages are extremely difficult to clean. Eliminating small entrapped volumes created by the overlapping of surfaces between mating components can also reduce contamination, since these entrapped areas become contamination pockets, difficult to clean with even stringent cleaning procedures. These pockets can release contamination into the system as a result of vibration or shock. For example, designs that permit the integral machining or welding of the sealing lands eliminate entrapped volumes near the valve seating surface. Mechanical attachments such as threaded interfaces, rolled lips, snap rings, overlapping retainers, and bolt fasteners can be sources of contamination, since small entrapped volumes are formed. These types of fasteners therefore are not used where they will be exposed to the flow stream.

Filters upstream of valve inlets are used successfully to remove contaminants from the propellants. Filters add size, weight, and pressure drop, however, and their use generally has been limited to valves  $\leq 1/2$  in. (12.7 mm); in critical systems such as those in LMDE, filters are used in 2-in. (5.08 cm) lines. Reference 46 includes information on the applications and capabilities of filters.

#### 2.7.2.2.1 Cleaning

Valve components must be cleaned to eliminate metallic particles that might cause seal leakage or galling and binding in tight clearances and to eliminate oils, greases, and other contaminants that might be incompatible with the service fluid. All metallic components except those that have been subjected to a soil-removal procedure (a requirement for passivating or anodizing) are precleaned by vapor degreasing, solvent hand wiping, or mild-alkaline cleaning. Complex assemblies such as adhesive-bonded, threaded, or brazed assemblies may require special cleaning. Dry-film-lubricated parts are cleaned only by vapor degreasing or solvent flushing. General information on cleaning methods, equipment, solvents, and facilities is presented in reference 47.

Precision cleaning of metallic components. – Precision cleaning normally is specified for components used in reaction control systems. All operations are performed in a controlled environment. Typical basic steps are as follows:

- (1) Clean ultrasonically in detergent solution at 125 to 140°F (325 to 333 K).

- (2) Soak 5 minutes in trichlorotrifluoroethane (e.g., Freon TF) that does not exceed specified levels of contamination, pH, and hydrocarbons.
- (3) Clean ultrasonically in trichlorotrifluoroethane for 1 to 5 minutes with a minimum power input of 2.5 W/in.<sup>2</sup> (0.39 W/cm<sup>2</sup>) of effective area; rotate the part to ensure cleaning of all surfaces.
- (4) Flush or spray rinse for 1 to 5 minutes with fresh, filtered trichlorotrifluoroethane.
- (5) Dry with 160°F (344 K) air or nitrogen filtered through a 0.45 μm membrane filter or vacuum dry at a vacuum of at least 22 in. Hg (74.3 kN/m<sup>2</sup>) while the part is at 160°F (344 K).
- (6) Select samples at random for quality-control monitoring of the effectiveness of the cleaning process.

General cleaning of metallic components. – Metallic components for booster engines, upperstage engines, hydraulic and pressurization systems, and vehicle systems usually are cleaned to a higher allowable level of contamination than that specified for reaction control systems. The usual basic steps for materials other than titanium are as follows:

- (1) Clean ultrasonically in trichloroethylene for 1 to 5 minutes at a minimum power input level of 5 W/in.<sup>2</sup> (0.78 W/cm<sup>2</sup>) of effective area. Do not allow part to dry.
- (2) Final rinse with either new or distilled trichloroethylene, trichloroethane, or trichlorotrifluoroethane for at least 2 minutes.
- (3) Inspect the effluent of a minimum of 5% of the detail parts for contamination. A typical quantitative contamination limit for surfaces that came in contact with the service fluid is specified below:

<u>Particle size range, μm</u>	<u>Maximum allowable particles</u>	
	per sq. ft.	per sq. m
175 – 700	5	54
700 – 2500	1	11
greater than 2500	0	0

- (4) Dry by oven bake, 260°F (400 K) maximum, or by purge with dry nitrogen gas.

Titanium parts are cleaned by the same procedure but trichlorotrifluoroethane is the only cleaning agent used. However, this cleaning agent must be completely removed by drying; otherwise, any residuals in contact with the propellant could result in deleterious effects to the metal. For instance, residuals from trichlorotrifluoroethane cleaning agent can cause embrittlement of 6Al-4V titanium parts when hydrazine is the service fluid. Thus, in the general cleaning processes described herein, there may be instances where the end application can affect the cleaning operation on a particular part. For cleaning titanium parts when complete drying could not be ensured, isopropyl alcohol has been utilized instead of trichlorotrifluoroethane.

Cleaning of nonmetallic items. – Nonmetallic items are cleaned by hand wiping with a lint-free cloth or brush moistened with solvent, and then are dried with a nitrogen gas purge. Trichlorotrifluoroethane is satisfactory for cleaning most nonmetallic parts; however, for elastomers such as EPR, alcohol is used. Items that are completely or partially fabricated of Kel-F are not exposed to halogenated solvents while externally stressed (e.g., an installed lip), because Kel-F is particularly prone to develop stress cracks.

#### **2.7.2.2.2 Contamination Barrier**

After cleaning of a detail part, it is placed in a contamination barrier or bag that will maintain the required cleanliness level of the detail part until assembly. The barrier material must minimize the sloughing off of particles by the detail part rubbing against the barrier, and it should not easily be punctured or torn. Formation of particles is particularly critical when small valves with close-fitting details are involved or when filters are in the system.

Polyethylene and nylon films are the two materials most widely used for barriers; the use of Aclar film is increasing. Nylon reduces sloughing off of particles and generally is used for packaging contamination-sensitive components; however, nylon provides a poor vapor barrier and therefore frequently is used with a polyethylene outer barrier. Aclar provides the best vapor barrier of the three. Components less sensitive to contaminants usually are packaged exclusively with polyethylene film. Although polyethylene and nylon are not compatible with many of the propellants to which the valve will ultimately be exposed, no incompatibility problem resulting from sloughed-off particles has occurred with these barrier materials. Aclar film provides a LOX-compatible barrier material and offers maximum protection for parts that will be exposed to LOX.

The barriers are heat sealed to maintain integrity (refs. 47 and 48). Cleanliness of the barrier itself is important. The barrier must be marked with the detail part number and cleaning process specification. Decals are used to identify the cleaning process, but may be tampered with; tamper-proof decals that delaminate upon removal are used. A recurring difficulty with heat-sealed barriers is that manufacturing personnel normally tend to cut open the barrier so that they can inspect the detail part; the barrier then is resealed. This practice can contaminate the detail part, yet leave no evidence of so doing. To prevent this kind of contamination, lettering is placed across the full length of the barrier, top and bottom, to give a visual indication of any alteration of the barrier's dimensions.

Internally cleaned parts fitted with protective closures can cause difficulties if the protective closure is not removed prior to operation. For example, a valve with an auxiliary bleed port in the housing was installed in an operational system; inadvertently, a plastic protective closure was left in the bleed port. The protective closure ruptured during operation and permitted a gross external leak. This kind of problem with protective closures is avoided when procedures to prevent inadvertent retention of the protective closure when the valve is installed in a system are established and enforced.

## **2.8 HOUSING**

### **2.8.1 Configuration**

The functions of the valve housing are to contain the service fluid, locate the valving unit and the secondary seal, and interface with the mating inlet and outlet ducts and the actuator; the housing weight, size, and pressure drop in combination with the valving element must be minimized. Design of the housing often is dictated by the particular considerations involved in installing the valving element in the housing. A ball, blade, or large poppet valving unit may require a split or two-piece housing so that the valving element may be installed; the two-piece housing usually is bolted together and requires a static seal and heavy flange sections on the two mating portions of the housing. The butterfly disk valving element, on the other hand, usually is installed in a one-piece housing that requires only a small, lightweight, bolted cover plate to retain a shaft seal; thus it may be significantly lighter in weight than a split housing.

To minimize misalignment with actuator linkage, the housing may incorporate an integral actuator instead of a valving unit housing and a separate actuator. In such integral designs, the more complex housing shape may require that special fabrication methods be employed to avoid distortion. Housing distortion and bearing failure occurred on early models of the Saturn S-II butterfly prevalue, which incorporated an integral actuator. As discussed in section 2.8.4, the machining process was changed to include cryogenic stabilization prior to finish machining.

### **2.8.2 Strength**

#### **2.8.2.1 INTERNAL LOADS**

The housing is designed to withstand specified and predicted pressure and pressure-transient loads without structural failure or detrimental deformation. Nonetheless, problems of various kinds have occurred, as the following examples illustrate. The housings of the main

propellant valves used on the F-1 engine failed on several occasions, apparently due in part to their large size and the high operating pressures. After application of proof pressure, a housing bore was distorted into an elliptical shape that was considered detrimental; preprocessing sequence was changed to eliminate the distortion. During F-1 engine development, rough combustion in the thrust chamber resulted in pressure surges in the feed system and high vibration loads that caused failure of the cast housing; the design was changed to call for a die forging instead of a sand casting. (Capability of the housing to withstand surges caused by rough combustion is, of course, not a requirement of an operational system but it is an unwritten practical requirement.)

The effect of too small a radius at transitions between sections of a housing may be critical, particularly where the material ductility is low (as in a casting), because a small radius results in stress concentration. A cast valve housing at a high internal pressure failed at a small radius. As a temporary "fix" to maintain testing schedules during development testing of the engine, the radius was increased to a limited extent; no other failures occurred.

Design of the housing also considers the effect of thermal shock loads or thermal transients; these phenomena may set up differential contractions that can generate extremely high stresses. For example, an aluminum valve housing in liquid-hydrogen service on the J-2 engine was overstressed and cracked by transient thermal stresses that developed as follows: the propellant feed system was purged with 150°F (339 K) gas, which heated the housing; when engine conditioning was started, liquid hydrogen that was dropped to the valve rapidly chilled a very thin section, adjacent to a bearing, to approximately -400°F (33 K) while the bulk of the exterior of the valve was still at 150°F (339 K). Interference between the housing and bearing also contributed to the failure. The section thickness was increased and the grain direction of the forging was aligned with the direction of maximum stress.

In valves that operate at a temperature other than room temperature, adjustments in allowable material properties and in the proof-test pressures are made to compensate for temperature/pressure differentials.

Welds in a housing require adjustment of the design yield strength to compensate for porosity and microcracking of the welds. The adjustment factor is designated as a weld efficiency and is based on the level of specified inspection requirements. Detailed information on welding is presented in reference 49.

### 2.8.2.2 EXTERNAL LOADS

External loads to a valve housing result from

- Use of the valve housing as a support or mount for other engine components.
- Misalignment of two sections of propellant line due to tolerance variations.

- Differential thermal contraction of propellant lines in cryogenic service.
- Vibration, shock, and acceleration loads on propellant lines (particularly when the lines are full).
- Actuator mounting.
- Flange loads required to achieve static seal compression.

When possible, a bellows may be used in the propellant lines to reduce external loads to the valve; however, the additional pressure drop associated with a section of flex line is a penalty that restricts its use. Also, use of the valve housing as a load-carrying member in the engine may not allow utilization of flexible lines. For example, an experimental 4-in. (10 cm) ball valve developed a cracked flange when the valve was installed between two misaligned sections of a propellant line and then chilled to  $-320^{\circ}\text{F}$  (78 K); the crack occurred at a sharp radius between the flange and remainder of the aluminum housing.

The gas-generator valve on the F-1 engine incorporates an outlet that is installed inside a mating duct but not rigidly attached to it. The valve outlet experienced fretting due to the motion between it and its mating duct. The fretting was reduced by incorporating a hard wear-resistant surface on the valve outlet, but fretting is a recurring problem.

### 2.8.3 Static Seals

Two-piece housings require a static seal between the two sections of the housing. Configuration and finish of the housing and static seal are critical in preventing leakage. Flange design is critical because pressure transients tend to cause deflections and hence leakage in the lightweight flange sections used in rocket engine valve housings. In an early LMDE, leakage in the oxidizer feed system occurred on shutdown because the load on the housing static seal was insufficient when the flange deflected during the transient pressure surge caused by the oxidizer valve closure (ref. 4).

Housing static seals for hydrocarbon or storable fuels typically are elastomeric seals such as O-rings; standard military specifications are followed for O-ring gland design. In early rocket engines, valve housing static seals in cryogenic oxidizer systems typically were asbestos-filled metal gaskets; these gaskets performed satisfactorily but are no longer used because of the high flange forces required to provide the deflection needed for reliable sealing.

Housing static seals for current use in cryogenic propellants or storable oxidizers generally are either a pressure-actuated, U-shaped metal seal coated with Teflon, or a pressure-actuated, C-shaped Teflon seal with a flat helical or segmented C-shaped spring that provides an initial sealing load.



Seals of butyl rubber have been used successfully for short-term exposure in the Lance storable-oxidizer feed system. An oversize nitroso terpolymer O-ring was successfully used in the LMDE nitrogen tetroxide feed system (ref. 14). An encapsulated butyl rubber seal was satisfactory in the nitrogen tetroxide feed system on the Titan II vehicle (ref. 50).

Detailed information on static seal design parameters is presented in references 4, 12, 13, and 51.

## 2.8.4 Fabrication

Housings are cast, forged, machined out of bar stock, or welded. The method chosen to fabricate the housing depends on the required size, configuration, tolerances, internal and external load requirements, schedule, cost, and housing material.

Large, complex housings may be sandcast, because this process is inexpensive yet flexible in that change in configuration may be easily made; permanent-mold castings may be used, despite higher cost, because they offer better material properties. Castings, however, are comparatively low in strength and ductility. These deficiencies make castings vulnerable to high pressure transients and to stress concentrations caused by too-small radii at changes in section. Castings also are porous, and poor quality castings may leak unless impregnated by a sealant. A static seal in the housing of the main propellant valve of the F-1 engine leaked because of porosity in the seal groove of the casting; the porosity was reduced by controlling the rate of cooling in the groove and in other critical areas of the casting. Detailed information on castings is presented in reference 52.

Forgings have been utilized for valve housings that are subject to substantial external loads. They have also been used to replace castings in uprated engines, where higher strength components are required and the envelope is already established. The main valves on the F-1 engine initially utilized cast housings; however, when flex couplings were eliminated and the valve housings were required to be load-carrying members of the engine, forgings were substituted. Although a forging may add weight and may require considerably more machining than a casting, the higher strength and ability to control grain direction usually outweigh these disadvantages. Detailed information on forgings is presented in reference 52.

Machining a housing out of bar stock is a method generally reserved for small valves.

Welded construction is used when the allowable external leakage is very low, e.g., a leakage rate of  $1 \times 10^{-3}$  scch, commonly specified for an extended-duration mission. Welded construction is relatively inefficient on a strength/weight ratio because of the comparatively low strength of a weldment, but overall weight advantages may be gained by elimination of the heavy flanges and bolts required for a bolted-flange design. A welded housing makes it difficult to disassemble the valve for servicing or for failure analysis.

Welding of the quality required for rocket engine valves requires close-tolerance machining, correct material callout, detailed welding procedures, quality control of the welding process, similar geometry of the to-be-welded sections, close control of cleanliness, and a leak test with a mass spectrometer. Electron-beam welding is widely used because of the close control of dimensions and quality possible with this type of weld. Failure to follow welding procedures can result in a poor weld. An explosive-actuated valve on the Gemini reaction control system leaked at a weld because the detail parts were not cleaned adequately prior to welding; the failures were random and were not detected by the lot acceptance test of the valve but were discovered subsequently during system testing.

Detailed information on welding is presented in reference 49.

## 2.9 SHAFT

The primary function of the shaft is to move and position the valving element, but requirements for sealing, valving element support and shaft guidance usually dictate the critical shaft design parameters. Because oversize areas of the shaft are multiplied in the other valve components (e.g., in the bearings, shaft seals, and housing), the necessity for minimizing weight usually results in the shaft being a highly stressed component. The rotating shaft is used with rotating valving elements such as butterfly disk, ball, blade, or rotary sleeve; the translating shaft, with translating valving elements such as a poppet or sleeve. The coupling between the shaft and valving element may be splines, drive flats, or similar mechanisms.

### 2.9.1 Dimensional Tolerances and Surface Finish

The required dimensional tolerances and finishes for a shaft depend on bearing and coupling seal contact area. Generally, a  $16\text{-}\mu$  in. ( $0.41\ \mu\text{m}$ ) AA shaft finish is utilized on shafts at the journal areas and  $16\text{-}\mu$  in. ( $0.41\ \mu\text{m}$ ) AA or better in the shaft clearances and seal requirements. Bearing clearances and shaft seal requirements are discussed in sections 2.10 and 2.11, respectively.

Dimensional control of coupling clearances between the shaft and valving element may be governed more by considerations of wear and fretting than by the required control of valve position. Drive flats, splines, cross pins, and similar components are subject to fretting. Because a loose fit on the coupling often is desirable to facilitate assembly, dry-film-type lubricants are used to limit fretting.

## 2.9.2 Strength

Points of maximum stress usually occur at a section discontinuity (e.g., relief for thread or spline). The best designs incorporate large fillets and gradual transitions between sections. Surface cracks generated by excessive heat during grinding create points of stress concentration; such cracks can be prevented by manufacturing process control. Shot peening of such highly stressed areas serves to reduce the surface tensile stress, and peening has been used to gain an additional safety factor. Specification of heat-treatment parameters for strength, ductility, and dimensional control is critical.

A rotating shaft is sized on the basis of an analysis of the combination of bending and torsional stresses. The shaft generally is designed to be capable of withstanding the maximum actuator torque output with the valve element locked at any partially open position. Maximum shaft bending stresses usually result from the fluid pressure acting on the closed valving element and imposing a bending moment on the shaft. Maximum torsional shaft stresses and even shaft failures may be caused by the inertial loads of a large, fast-operating valving element. A critical analysis of this condition sometimes indicates that consideration should be given to reducing the inertia of the valve element (e.g., use of a hollow ball). Dissipating impact energy over the length of the shaft also may be used to reduce stress. As an example, failure in a spline relief on the shaft of a fast-acting ball valve on the F-1 engine was eliminated by drilling lengthwise from the opposite end of the shaft. This axial hole reduced shaft torsional spring rate and allowed the energy of impact to be absorbed over the length of the shaft instead of being concentrated at the smallest section. The ball also was made hollow.

Design of a translating shaft is similar to that of a rotating shaft except that the designer is concerned with axial stresses rather than torsional stresses. The shaft is sized on the basis of a critical analysis of tensile (or compressive) loads and possibly bending loads. The bending may result from flow forces acting on the valving element and its shaft. These flow forces generally are at right angles to the direction of motion of the valving elements. Since maximum stresses usually are associated with dynamic loads (impact or vibration), a good understanding of actual operating requirements is required in order to produce a good design. For example, a "T"-slot connection between the armature and shaft of the J-2 engine four-way solenoid valve (fig. 10) failed when exposed to vibration at low-temperatures. A design change to a thicker armature section and increased fillet radii in the slot corrected the problem.

The proper handling of bending loads in a translating shaft is equally as important as the consideration of axial loads. Many of the problems occurring in valve assemblies with a linear-motion valving element result from shaft bending, the effects of which appear as binding, slow operation, or galling in a partially open position. Although these problems may be induced by shaft bending, size and weight limitations on the shaft design may prevent resolution by stiffening the shaft; instead, the resolution is more likely to be associated with the bearing design as discussed in the following section on bearings.

## 2.10 BEARINGS

Poorly selected length-to-diameter ratios in spring-operated valves, poor alignment, poor materials, and shaft deflection resulting from structural loads are common examples of factors that contribute to valve failure to open or close properly. Bearings in a valve assembly guide and support the valving element or shaft and thereby minimize this kind of valve failure. Bearings may be classified as rolling contact or sliding surface. Rolling-contact bearings include ball, roller, or needle bearings used in rotary-motion valves (butterfly, ball, and blade) for either radial or thrust loads. Sliding-surface bearings include guide bearings used for axial linear motion in poppet and sleeve valves and journal and thrust bearings used for rotary-motion valves. Sliding-surface bearings are simple and inexpensive, but usually have a higher coefficient of friction than rolling-contact bearings. Friction is an important consideration in rocket engine valve bearings, because oil-film lubrication generally is not available. For cryogenic applications, where moisture from atmospheric condensation in bearings may freeze, the rolling-contact bearing is less susceptible than the journal bearing to "hangups", because of the greater void volume of the roller bearing. However, rolling-contact bearings may corrode in a wet environment. For this reason, many main cryogenic-propellant valves that are located adjacent to the thrust chamber, and are therefore close to atmospheric moisture, operate "dry", i.e., they are sealed on both the inboard and outboard ends. Gas-generator valves, which are further upstream of the thrust chamber, use bearings that operate "wet", i.e., they are sealed only on the outboard end of the bearing and are therefore exposed to the operational fluid.

For valve bearings in which conventional bearing materials and lubricants are acceptable, standard handbooks such as reference 53 provide adequate design information on bearing load capacity. An aerospace valve, however, operates in reactive service fluids, over wide temperature ranges, and under severe environmental conditions. Lubrication and material-compatibility requirements impose severe limitations on valve bearing design. Bearing loads must be kept low in order to limit friction and minimize wear. Bearing clearances must be sized to allow for large thermal expansions and contractions.

### 2.10.1 Guide Bearings

Valve guides are sliding-surface bearings that prevent or restrict lateral motion of a translating valving element or shaft. Control of lateral motion is necessary during opening or closing of the valve so that scrubbing of the poppet on the seat is limited and the resulting wear reduced. The precision required for this control depends on valving element-to-seat dimensional relationships and geometry and on the operating loads.

### 2.10.1.1 GEOMETRY

A guide bearing may be machined as an integral part of the valve housing, or it may be a separate bushing when a preferred bearing/guide material combination is desired. A guide bearing may be one continuous bore; two separate bores may be used where greater length is desired without the higher friction loads that accompany increased surface area contact. The length-to-diameter (L/D) ratio of a guide bearing is an important design consideration, because too small a ratio may result in cocking and galling of the shaft in the guide. In general, the L/D ratio is made not less than 1.0, and a ratio greater than 1.0 is preferred.

Control of the lateral motion between the valving element and seat in a poppet valve, important in preventing seat wear, requires minimum clearance between guide and shaft. Equally basic to the control of lateral motion is the geometrical relationship between the overall guide length and the distance between the guide and seat. A common design deficiency in poppet valves is excessive overhang of the poppet and guide as compared to the overall guide length. Binding induced by thermal gradients is a common occurrence usually corrected by increasing the diametral clearance or by changing the materials. Load-induced deflections must be considered.

### 2.10.1.2 LOAD CAPACITY

The guide is designed to have sufficient bearing capacity to withstand all imposed loads including those caused by flow, pressure, vibration, and acceleration; generally, guides are used when loads are low. Neglect of proper consideration for load capacity can lead to difficulties. For example, the oxidizer boost-termination valve in an early Lance engine failed to close completely. Investigation disclosed that the valving element, a plug, was subjected to a large pressure drop at the almost-closed position; this condition produced an excessive lateral force on the bearing. Inadequate bearing length then resulted in cocking and galling of the valving element.

### 2.10.1.3 SURFACE FINISH

Guide bearing surfaces are highly finished to minimize friction and prevent wear and the generation of contaminants. Typical surface finishes are 32  $\mu$  in. (0.81  $\mu$ m) AA for plastics or soft bearing materials. Edges are bell-beveled or radiused to minimize edge loading and damage during assembly.

## 2.10.2 Journal Bearings

Journal bearings are sliding-surface bearings used in radially loaded rotary-motion valves. In most rotary valve applications, journal bearings are subjected to oscillatory motion, usually with an angular rotation of less than 90°.

### 2.10.2.1 GEOMETRY

Journal-bearing geometry is treated in references 4 and 53. A typical specification is roundness within 0.0005 in. (13  $\mu\text{m}$ ). Specification for the bearing length must consider possible misalignment. If the deflection angle is too great, either the length of the bearing is reduced or self-aligning spherical mounts are employed.

### 2.10.2.2 LOAD CAPACITY

Journal bearings usually are limited to valves operating in low-pressure systems. Plastics, dry-film-type lubricants, and soft-metal platings have been used to minimize friction in a wide range of applications. Reference 4 contains design information on many of these materials and the allowable bearing pressure (bearing load divided by the projected area of the bearing). The 8-in. (20 cm) liquid-hydrogen and liquid-oxygen butterfly prevalues used on the Saturn S-II stage utilize an all-metal journal bearing with a soft-metal plating. This bearing has a high load capacity with reasonably low friction and a minimum clearance change due to temperature variations. Bearing pressure is limited to approximately 6000 psi (41  $\text{MN}/\text{m}^2$ ). Most plastic bearings will not sustain as high a bearing pressure but may have considerably lower friction.

### 2.10.2.3 SURFACE FINISH

As with other bearings, the journal bearing finish is important in keeping friction and wear low and in minimizing generation of contamination. Typical journal bearing finishes are 32  $\mu\text{in.}$  (0.81  $\mu\text{m}$ ) AA for the bearing and 16  $\mu\text{in.}$  (0.41  $\mu\text{m}$ ) AA for the shaft.

## 2.10.3 Rolling-Contact Bearings

Rolling-contact bearings include the many varied configurations of ball and roller bearings. Their usual function is to support a shaft against radial loads, although rolling-contact bearings frequently are used for thrust bearings. The basic advantage of a rolling-contact bearing is low friction, and its application as a valve shaft bearing usually involves oscillatory motion with relatively high intermittent loading. The principal factors involved in successful design include satisfactory load capacity, adequate lubrication, resistance to corrosion, and good alignment in the valve. Standards for surface finish of rolling-contact bearings have been established by the Anti-friction Bearing Manufacturers Association (AFBMA), and surface finish usually requires no specific attention from the valve designer. Rolling-contact bearings normally are considered as special components and are procured from specialized manufacturers to meet the individual valve design requirements.

Rolling-contact bearings often are used as combination radial and thrust bearings. Ball bearings of this type are commonly called angular contact; roller bearings of this type usually incorporate a tapered or barrel-shaped roller. Ball bearings are comparatively in short length, can accept both radial and thrust loads, and are low in cost; roller bearings have a higher load capacity and are diametrically smaller.

When the service fluid permits, e.g., in an RP-1 propellant system, rolling-contact bearings in valves are lubricated with a grease-type lubricant. In cryogenic, highly corrosive, or high-temperature fluids, however, bearings usually operate with no lubrication. Reduced friction between sliding surfaces often is provided by a separator made from a plastic or 300-series steel.

Roller bearings are used in the butterfly valves for the RP-1 feed systems on the Atlas booster and sustainer engines. A full complement of rollers yields a diametrically thin bearing that minimizes valve size and weight. These bearings are packed with a grease-type lubricant that effectively minimizes rubbing friction between rollers.

Full-complement roller bearings have a considerably higher load capacity than bearings with caged rollers, but have a tendency to skew in operation. Skewing can cause bearing failure because of severe wear at the ends of the rollers as they contact the end retainer. This condition occurred in the butterfly valve in the LOX propellant feed system on H-1 engines; it was eliminated by placing a Teflon washer between the rollers and the end retainer.

### 2.10.3.1 GEOMETRY

Rolling-contact bearings typically are designed to operate with relatively small internal diametral clearances (approximately 0.0005 in. (13  $\mu\text{m}$ ) minimum), so that a maximum of restraint on the valve element is provided. Additional internal or external clearance may be required to accommodate transient and steady-state thermal contractions. In cryogenic valves, this clearance usually is provided between the outer race and housing, although this practice generally is undesirable from the standpoint of valve position control. The requirement for additional clearance was compensated for in a liquid-hydrogen pre valve for an upper stage by mounting the bearing race in an Invar ring. At cryogenic temperatures, the aluminum housing contracted around the Invar ring instead of the bearing race. The cross section of the ring was proportioned in relation to the aluminum housing so that the ring, although compressed at cryogenic temperature, maintained a constant diametral clearance with the bearing race. In hot-gas service, valves have failed to operate satisfactorily when there was not enough clearance between the outer race and the housing to prevent binding resulting from thermal expansion of the bearing while the housing was at room temperature; appropriate clearances had to be provided.

### 2.10.3.2 LOAD CAPACITY

The most significant stress parameter involved in evaluating the capacity of a rolling-contact bearing is Hertz stress: the maximum compressive stress that occurs at the center of the surfaces of contact. This stress is a function of radius of curvature of the bearing as related to curvature of the race, Young's modulus and Poisson's ratio for the materials, and the load carried by the bearing element. Hertz stress and the load-life factor for a given bearing depend to a very large extent on the detail design of the bearing and on the manufacturing processes. Hertz stresses at the design load in high-performance bearings usually exceed the elastic limit of the material. The stresses that actually cause bearing failure, however, are shear stresses that occur slightly beneath the surface. The effect of these stresses is commonly observed as a spalling of the bearing surface. Subsurface shear stress and Hertz stress are directly related, and common practice is to use Hertz stress as the significant parameter in evaluating capacity. Reference 54 analyzes Hertz stress as related to ball and roller bearings. Operating life also must be considered as a critical factor when the bearing application is evaluated. Reference 55 discusses the factors commonly considered in relating load and life for a given application.

### 2.10.4 Thrust Bearings

Thrust bearings support axial loading of a rotating shaft or valve element; they may incorporate either sliding-surface or rolling-contact elements. The principal factors in design are capacity, friction, and cost. The thrust bearing generally is used to absorb loads generated by fluid pressure acting on the valve stem at the stem seal; these loads can be very high. In some instances, the thrust bearing may absorb only the load generated by valve element-to-seat reactions or flow forces on the valve element; these forces can be quite low. Often the thrust bearing requirements can be met by an angular-contact or tapered-roller shaft bearing. References 4 and 53 present general information on thrust bearings. See also sections 2.10.2 and 2.10.3.

A typical thrust bearing use is in the ball valve for controlling liquid oxygen for the gas generator on the F-1 engine, where a 1250-pound (5.56 kN) thrust load is absorbed at operating pressure. The bearing uses straight rollers and thrust washers. In the J-2 engine throttling valve, an angular-contact ball bearing absorbs hydraulic flow forces that act both radially and axially on the rotary sleeve. Loads are relatively low, and the bearing provides low friction and accurate positioning of the valve element. In the liquid-oxygen feed system on the S-II stage of the Saturn vehicle, the shutoff valve uses a plain sliding-surface thrust washer to guide the butterfly. The material is asbestos-filled Teflon, which is utilized for good compressive strength and thermal expansion characteristics to maintain a fixed clearance between the steel butterfly and aluminum housing.



#### 2.10.4.1 LOAD CAPACITY

The designer performs a stress analysis to ensure that the thrust bearing possesses load capacity adequate to withstand all anticipated loads. As required, the bearing design is modified to provide the necessary capacity.

#### 2.10.4.2 SURFACE FINISH

As with other components, surface finish is a factor in wear and generation of contamination. Typical values of surface finishes for a journal-type thrust bearing are  $32\mu$  in. ( $0.81\mu\text{m}$ ) AA for the bearing and  $16\mu$  in. ( $0.41\mu\text{m}$ ) AA for the shaft.

### 2.11 SECONDARY SEALS

Secondary seals include the dynamic seals that seal the valve shaft or valving element and prevent external leakage and those that seal the pressure-balancing piston of a poppet valve or the sleeve of a sleeve valve. Secondary seals for either a rotating or translating element are subject to continuous-contact, oscillatory, sliding motion. Choice of a seal type depends on the motion of the shaft or surface being sealed and on environmental conditions. The oscillatory motion in the valve may require special design solutions or control of the checkout procedure to avoid seal failure. Design information on secondary seals is presented in references 4 and 51. Reference 3 also presents information on actuator seals that is applicable to secondary seals.

#### 2.11.1 O-Rings

O-rings, because of their simplicity of design, extensive standardization, ease of installation, and excellent reliability record in the aerospace industry are, within their temperature and compatibility limitations, preferred as shaft seals for both translating shafts and rotary shafts with  $90^\circ$  oscillatory motion. O-rings are, however, subject to special kinds of problems.

When rapid oscillatory motion is involved in valve operation, it is especially important that the proper gland design be utilized in the shaft seal. An O-ring failure on a series-redundant ball valve on an upper stage of the Saturn V was traced to improper O-ring gland design. The design utilized a small-cross-section seal; the O-ring twisted and failed when the valve was operated dry. Although small-cross-section O-rings are used successfully in high-speed continuous rotary motion, oscillatory stop-and-start motion requires an O-ring of relatively

large cross section. Since a larger gland design was required to incorporate the recommended O-ring, no design change was made, but rather the valve operation was slowed for dry checkout operations.

Extension of an O-ring into the clearance between the shaft and housing can result in O-ring cutting. Reference 56 gives standard O-ring groove dimensions. Limits of extrusion are presented in reference 57 for various O-ring materials and hardness. O-ring damage during installation can result from sharp corners in the bore. For example, sharp corners on intersecting flow passages caused O-rings to be cut during installation in the boost shutoff valve on the Lance engine (ref. 58); more generous radii were utilized to correct the problem. O-ring lubrication prior to assembly reduces the chance of installation damage. In addition, lubrication reduces friction and thereby increases seal life under dry checkout conditions. At low pressures, O-rings may leak after extended service. On the F-1 engine four-way control valve, this kind of leakage was caused by O-ring flattening that resulted in a loss of squeeze (ref. 59). Investigations revealed that the flattening apparently was caused by the temperatures and pressures of the service conditions combined with leaching of plasticizer from the Buna-N rubber by the fuel (RP-1).

### 2.11.2 Lip Seals

A lip seal, as applied to a valve shaft seal, is a seal that has a relatively thin lip lying on the shaft and extending into the fluid cavity so that fluid pressure acts to increase the seal load. This type of seal probably has the most wide and varied application of any dynamic seal. Lip seals have been fabricated from nearly every material that can be flexed, from leather to high-strength metals.

A lip seal offers a wide range of design parameters (material, thickness, length of lip, flange retention) to meet particular requirements. An advantage of a lip seal is that it will accommodate more radial shaft motion than an O-ring. This type of seal has performed well with either rotating or translating valve shafts in such fluids as hydraulic oil, RP-1 fuel, and storable propellants.

Problems with lip seals usually are related to contamination (sec. 2.7.2), insufficient pressure between the seal lip and the shaft, roughness of the lip and shaft dynamic sealing surfaces, or a leak at the lip seal flange. For example, numerous failures of the shaft lip seals on the main liquid-hydrogen valve of the J-2 engine occurred. It was found that moisture was trapped on the shaft and then froze when propellant was dropped (ref. 60). The ice abraded the Mylar lip seal and caused Mylar flakes to be trapped on the lip seal; the seal then leaked.

Successful design of a lip seal involves compromise. For example, high bearing pressure or interference between the lip and shaft promotes good sealing but also induces rapid wear of the sealing surface. A long lip generally aids sealing but tends to result in more wear. Properties and thickness of the seal material and the operating environment affect the design. Multi-ply seals are often used to withstand high operating pressures while maintaining lip flexibility. Wedge-shaped backup rings between the seal and housing and shaft are used to support the seal against high pressure.

Seal wear is also related to surface roughness. Reference 61 discusses the various types of surface textures. It is helpful to observe various surfaces with an electron microscope or interference microscope and perhaps correlate wear test results with the observations. Two surfaces of the same roughness value may not have the same wear characteristics; the wear characteristics seem to be related to the frequency of the projections as well as the surface roughness. For example, an as-processed hard-anodized surface will be much more abrasive than a hard-anodized surface that has been ground back to attain the same roughness value; the as-processed surface has sharper projections than the ground surface. A surface with high-frequency roughness (fig. 26(a)) is usually more abrasive than a surface with

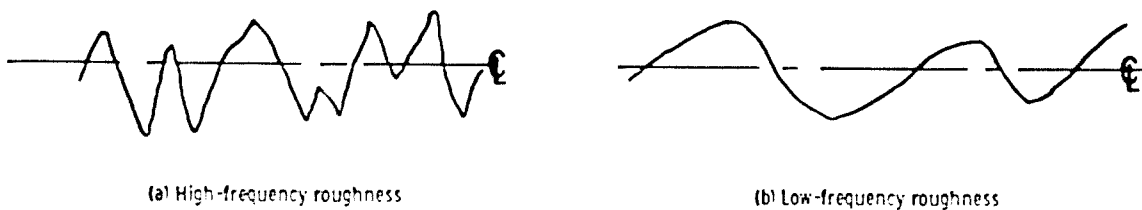


Figure 26. - Illustrative sketches of types of surface roughness.

low-frequency roughness (fig. 26(b)) of the same roughness value because the high-frequency surface generally will have sharp projections.

The characteristics of a surface that affect seal wear are dependent in part on the process used in finishing. Lapping and turning tend to give good wear characteristics, whereas grinding to the same roughness value tends to give relatively poor wear characteristics.

Design of the lip seal flange to effect a seal to the housing is a critical feature of a plastic seal that must operate at cryogenic temperatures. Leakage past the flange of the lip seal has occurred because the sealing load on the flange decreased when the valve was chilled. Leakage past the flange has also occurred as a result of creep of the static seal when the seal was subjected to additional loading from proof-pressure testing; retorquing was necessary to reduce leakage to an acceptable level. However, the practice of retorquing indicates a comparative narrow margin of performance when the component goes to the field. Seal retainers have been designed with Belleville springs to load the seal and thus avoid retorquing after creep had occurred. An aid in precluding both the loss of seal load due to relaxation at cryogenic temperatures and material creep under pressure at ambient

temperatures is a seal retainer with serrations (fig. 27) or raised lands. Also, temperature compensation can be accomplished through the use of material combinations that maintain (or preferably increase) the flange load at low temperatures.

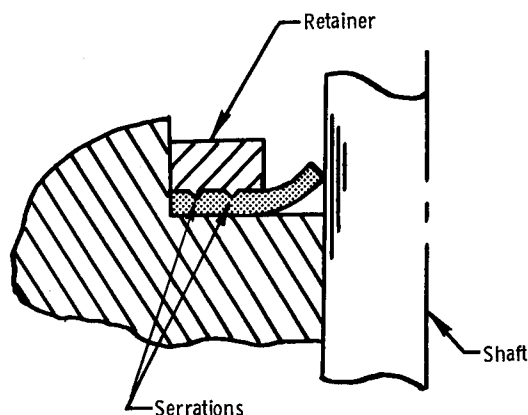


Figure 27. - Cross-section sketch of retainer with serrations for preventing creep of lip seal material.

### 2.11.3 Face Seals

Face seals are most commonly used to seal a rotating shaft when an all-metal valve design is required. Figure 28 shows an all-metal bellows-loaded face seal, the seal being integral with the bellows. The advantage of a face seal thus lies in the choice of materials available to meet high temperature, propellant compatibility, or other operating requirements that rule out plastics or elastomers. Liquid-fluorine and hot-gas systems, for example, require valves that must operate in environments where plastics and elastomers are unsuitable. The seal interface usually is obtained by simple flat lapping, a process that yields a high quality sealing surface at a relatively low cost. Lapped surfaces  $4 \mu$  in. ( $0.10 \mu\text{m}$ ) AA or better have been successfully utilized.

A metal bellows generally is used to support the face seal and provide a means of compensating for variations in radial location between the shaft and seal. High cost of the bellows is the principal disadvantage of the face seal. Also, the face seal is relatively large compared with other seal types. Alignment requirements dictated by the sealing-surface bearing pressure necessary to satisfy the leakage and cycle-life requirements also result in

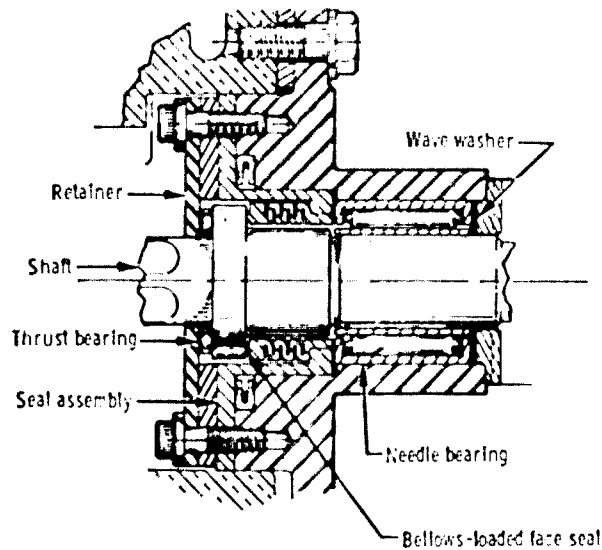


Figure 26. - Cross-section drawing of installed all-metal bellows-loaded face seal.

relatively expensive fabrication and assembly techniques. A successful design is achieved when specified cycle life is demonstrated without producing enough wear on the sealing interfaces to cause excessive leakage. Conditions that cause wear and leakage are discussed in reference 62.

The operating load utilized for a face seal is based on the allowable operating PV factor of the material combination (PV factor is defined as bearing pressure  $P$  times average rubbing velocity  $V$ ). A bellows is installed in compression so that it preloads the face seal. The effective diameter of the bellows is sized to pressure load the seal interface to a controlled level.

## 2.11.4 Bellows

Bellows are used for the shaft seal on a reciprocating shaft to achieve a frictionless "zero-leakage" seal, and for the secondary seal on a rotating shaft. Used as reciprocating-shaft seals, bellows eliminate the need for sliding seal surfaces and the attendant leakage; used in conjunction with a face seal on rotating shafts, they accommodate axial displacement between the shaft and valve housing. Critical parameters that must be considered in bellows design include material compatibility, stress level, cycle life and cycling speed, length and direction of stroke (compression or tension), required spring rate, service environment, and fabrication methods (ref. 4).

#### 2.11.4.1 OPERATING CHARACTERISTICS

Reference 63 presents rigorous methods for stress analysis of bellows; reference 64 presents a useful simplified analysis that permits a bulging-stress index number 1.5 times the allowable material properties. Since accuracy of stress analysis is highly dependent on actual bellows form and wall thickness, frequently a sample bellows is sectioned to determine the actual dimensions. Bellows perform better in compression; they are less nearly linear in tension. For this reason, external pressurization is preferred to internal pressurization.

Bellows can be made from a single ply of material or from multiple plies. A single-ply bellows is relatively low in cost and easy to inspect. Multi-ply bellows offer several clear advantages:

- (1) A through crack in one ply will not leak and cause system failure.
- (2) The interply friction damps vibration and thereby improves bellows fatigue life.
- (3) The multi-ply bellows gives a lower spring rate than a comparable single ply.

A disadvantage of the multi-ply bellows is that a small leak in one ply will cause a failure if a strong oxidizer (e.g., fluorine) reacts with contamination trapped between the plies. When a design requires a bellows to provide a spring load as well as a secondary seal, any yielding of the bellows will relax the spring load and may cause an internal leak. A better design utilizes a bellows only as a secondary seal and employs an auxiliary spring to provide a seal load if required. A bellows in a valve on the gas generator on the H-1 engine had a marginal capability of withstanding the stresses incurred in cycling, and consequently the bellows yielded during cycling of some valves. Acceptable bellows were obtained by cycling the bellows 25 times and then measuring the bellows lengths to find those that had not yielded.

#### 2.11.4.2 FABRICATION

Bellows generally are fabricated with specialized equipment, and consequently most bellows are procured from suppliers specializing in their design and fabrication. Bellows are made by welding, machining from solid stock, metal deposition on a form, or forming from a seamless tube (hydroforming). Welded bellows have suffered in the past because of poor weld quality that resulted in poor cycle-life characteristics. Machined bellows offer the advantage of fast fabrication, but it is difficult to machine the thin wall required for low spring rates. Hydroforming is the most common method of fabrication and may be used in multi-ply construction with distinct advantages such as consistent quality and low cost in production quantities.

## 2.11.5 Pressure-Balancing-Piston Seal

Dynamic seals are required for the pressure-balancing piston of a large poppet valve or for the sleeve in a sleeve valve. The seal may be an O-ring, a lip seal, or a spring-loaded C-shaped Teflon seal. Because of the relatively large size of the seal, problems of strength, friction, and thermal contraction are aggravated. Seal configuration and material are selected so that the seal is flexible enough to be pressure actuated, is stiff enough to withstand the applied pressures, and has sliding friction characteristics that permit a reasonable actuator size.

The sleeve seal on the cryogenic-propellant poppet valve on the M-1 engine utilized an 11-in. (27.9 cm)-diameter Kel-F lip seal (fig. 29). A major effort centered on development of a

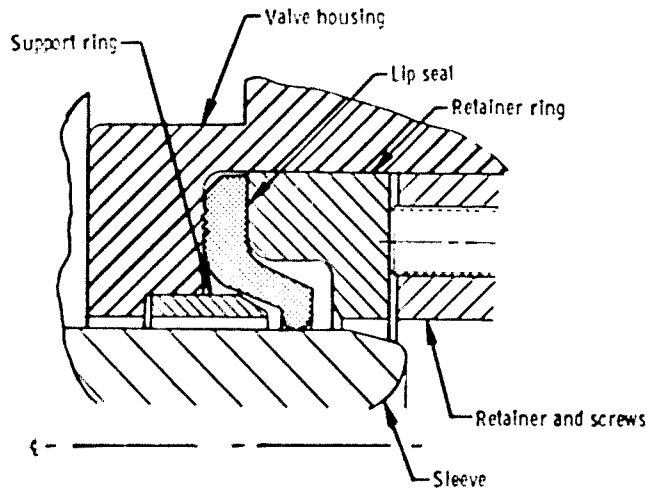


Figure 29. - Cross-section sketch of lip seal installation in sleeve valve on M-1 engine.

support ring to withstand the forces generated by burst-test pressures of 2400 psi (16.5 MN/m<sup>2</sup>) that failed an unsupported seal (ref. 36). The support ring made the friction forces less sensitive to applied pressure. Friction forces were high because the diametral interference required to seal was relatively high (0.10 to 0.14 in. [2.54 to 3.56 mm]).

The sleeve seal on the 8-in. (20 cm) cryogenic-propellant poppet valve on the F-1 engine incorporates a U-shaped Teflon seal ring that is loaded by an expander ring and a compressor ring (fig. 30); the loading rings are made of metals selected for coefficients of contraction that tend to keep the seal in intimate contact with the housing and sleeve during the period when the seal has chilled and contracted but the housing has not yet chilled and contracted. This seal configuration was preferred to a conventional plastic lip seal primarily because the Invar expander ring minimizes transient chilldown leakage in this large valve

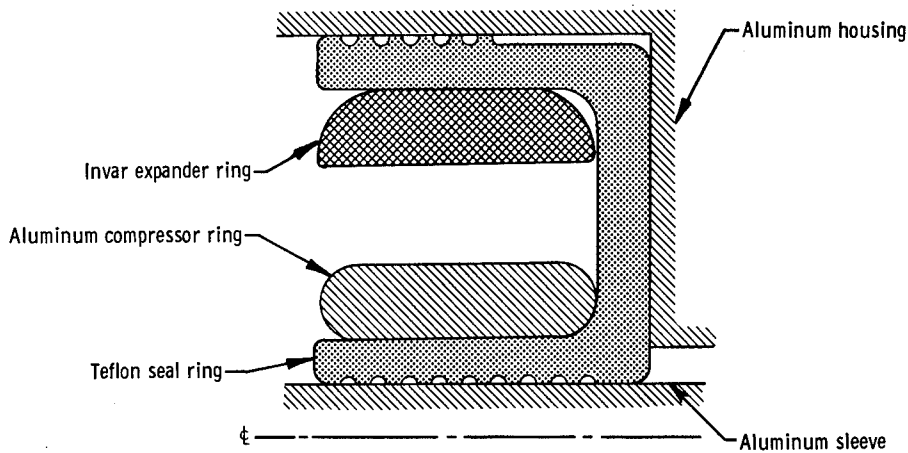


Figure 30. - Cross-section drawing of installation of sleeve seal on poppet valve on F-1 engine.

where the housing chills down slowly because of its low ratio of surface area to mass. The expander and compressor rings apply constant loads to the Teflon seal ring that minimize the effect of load changes on the seal resulting from changes in the applied pressure; these rings also simplify valve assembly by eliminating the need for a retainer with multiple retainer bolts. Thermally induced tensile cracks at the corners of the Teflon "U" have been observed in valves that were subjected to extended hot firing of the engine. Development tests have shown that the occurrence of these cracks can be minimized by (1) limiting the offset between the legs of the U, and (2) establishing the tolerance on the seal to ensure that the seal is squeezed between the piston and the valve body, thus placing the Teflon in compression at room temperature.

## 2.12 MATERIALS

Materials for valve components are selected for strength and ductility over the operating temperature range, compatibility with the service fluid, stress- and galvanic-corrosion resistance, and hardness. Consideration also is given to weight, cost of the material, fabrication costs, the coefficient of thermal expansion and contraction, dimensional stability, magnetic properties, and availability. Data on material selection and properties are presented in references 65 and 66. Compatibility data are presented in references 41 and 42.

Some materials have special limitations that restrict their use. For example, detrimental embrittlement of Inconel 718 may occur when it is exposed to pure hydrogen gas at a pressure of 5000 psi (34 MN/m<sup>2</sup>). The 17-4 PH and 17-7 PH steels are subject to stress



corrosion, but are satisfactory in applications where the part is not under stress (other than residual stresses) during storage and is protected against corrosion.

Corrosion-resistant steel alloys usually are passivated and aluminum alloys anodized for corrosion protection. Passivation is the forming of a passive oxide coating on CRES parts by an acid bath. Exposure time and strength of the acid must be closely controlled to limit the etching effect. A series of failures of a valve on the SE-8 reaction control system occurred after new passivation and cleaning procedures were instituted. The processor had instituted a passivation procedure that included a 30-minute exposure to a solution of nitric and hydrofluoric acids. This process etched the CRES valve element; its coefficient of friction with the housing increased to the extent that the valve did not actuate fully. The process was corrected, and a specification to control the process was written and enforced.

## 2.12.1 Poppet Valving Unit

### 2.12.1.1 HARD-ON-HARD UNIT

An essential property for a hard-on-hard poppet valving unit is hardness of the sealing surfaces of the poppet and seat. Hardness is essential for wear resistance and for finishing the sealing surfaces to the surface fineness required.

Flat poppets and seats of 440C steel are used for room-temperature gas and fuel service and for gas service at temperatures down to  $-250^{\circ}\text{F}$  (117 K). The material has low impact strength at cryogenic temperatures, but it is adequate in small pneumatic valves. Other steels such as 347 and 303 have been used successfully in flat poppets and seats in the same environment as described for the 440C steel, but with greater allowable leakage rates. It is difficult to lap 300-series alloys to the fine finishes required for low leakage rates.

Spheres fabricated of a cermet such as tungsten carbide or a ceramic such as synthetic sapphire or ruby also are used with 440C steel seats in a cryogenic environment. Generally, a spherical poppet or ball is fabricated of a material harder than the seat so that deformation occurs in the seat rather than in the ball. A comparatively soft ball may nick if the ball impacts a sharp corner of the seat. Subsequent actuation of the ball will rotate it and cause the nick to damage the seat in several places and cause leakage. A local compressive yielding of the seat is less apt to cause leakage. A poppet and seat combination of a synthetic sapphire with a 6061-T6 aluminum seat has been used and shows satisfactory cycle life (ref. 15).

Propellant compatibility may also be critical because of the possibility of attack on the sealing surfaces. Initial-design propellant valves for the Gemini SE-6 reaction control system used a tungsten carbide ball as the poppet in  $\text{N}_2\text{O}_4$ /MMH service. Operation of the valve in  $\text{N}_2\text{O}_4$  resulted in pitting of the ball and subsequent leakage. The pitting occurred because

the  $N_2O_4$  had 0.3 percent water content (the specified water content was 0.1 percent) and reacted to form nitric acid, which attacked the cobalt binder in the tungsten carbide ball. Pitting of the 440C steel seat also occurred because of attack by the nitric acid. The 440C is, however, reported to be compatible with  $N_2O_4$  having a water content less than 0.2 percent (ref. 41). Another design incorporated a sapphire ball that was compatible with the "wet"  $N_2O_4$ , but cracked after 10,000 cycles, apparently due to impact. The final design utilized a Stellite ball and a Teflon soft seat, both of which are compatible with  $N_2O_4$  having a 0.3-percent water content.

Seal material in a flexible-disk-seal valving unit is critical because of the wear resistance and sealing-surface finish required. Initial seal material for the J-2 four-way solenoid valve (ref. 26) was 440C, but its use was discontinued after consideration of the low impact strength of 440C at cryogenic temperature. Havar and Elgiloy (cobalt-based spring alloys) then were evaluated. Sealing performance with the Havar was not satisfactory because its surface finish (5 to 6  $\mu$  in. (0.13 to 0.15  $\mu$ m) AA) was relatively poor. Elgiloy, in the heat-treated, as-rolled condition, proved satisfactory. The seats were heat-treated 440C steel.

#### 2.12.1.2 HARD-ON-SOFT UNIT

Materials for poppets for a hard-on-soft poppet valving unit often must satisfy special requirements as well as the general requirements noted earlier. A soft seal, for example, almost always is bonded to a steel poppet, since the bonding process requires a high temperature that would affect the heat treatment of an aluminum poppet. Typical materials for the poppet of a hard-on-soft poppet valving unit for pneumatic and hydrocarbon fuel service include aluminum alloys 6061-T6 and 7075-T73 and steel alloys such as 17-4 PH, 17-7 PH, 304L, 347, A286, and Inconel 718. Poppet materials for storable propellants (IRFNA and  $N_2O_4$  oxidizers and hydrazine-type fuels) are 6061-T6 aluminum and 17-4 PH and 17-7 PH steels. Typical solenoid-operated valves for reaction control systems use armatures fabricated of 430F steel with a mechanically retained Stellite No. 3 ball and of 446 steel with a hardened sealing surface.

Poppet material for the valves in the Lance engine, which uses storable propellants, is Tens-50 permanent-mold casting. The oxidizer poppet, subject to high impact stresses, is heat treated to the T4 condition to improve elongation and thereby eliminate impact failures.

Typical poppet materials for cryogenic service (including liquid oxygen, liquid hydrogen, and low-temperature gases) are A286, 6061-T6, and Inconel 718. These metals all have adequate elongation even at  $-420^\circ\text{F}$  (22 K). Inconel 718 is preferred to A286 because of its higher strength and less tendency to gall.

## 2.12.2 Butterfly Disk Valving Element

The disk material is a critical factor in preventing leakage during chilldown of a valve in cryogenic service. The main oxidizer (and main fuel) butterfly valves on the Atlas vehicle used Kel-F lip seals with 7075-T73 aluminum disks and aluminum housings to reduce weight. The oxidizer valve leaked during propellant conditioning when liquid oxygen was first dropped from the tank to the valve; the leakage would later stop. A change in disk material from aluminum to 431 steel eliminated the transient leakage. Apparently the leakage occurred because the disk chilled rapidly and pulled away from the seal; the seal did not contract initially because it was restrained by the housing, which because of its greater mass did not chill as rapidly as the disk. The 431-steel disks were machined from forgings. The as-forged surfaces (e.g., the bottom of the gate), which were not machined because it was unnecessary, tended to corrode. Later designs used A286 and Inconel 718, materials with better corrosion resistance and low-temperature ductility. It may be noted that the same disk material also is used in hydrocarbon-fuel service on the Atlas, because use of the same material simplifies procurement and logistics when oxidizer valve and fuel valve are the same size, the usual case in large booster engines.

Typical disk materials for hot-gas throttling valves are Inconel 718 forgings, Rene 41, and Inconel X-750, all of which have good strength properties up to approximately 1300°F (978 K).

## 2.12.3 Ball Valving Element

Special requirements for the material for a ball valving element include a low coefficient of friction in combination with the seal, because friction is a major factor in sizing the valve actuator and in achieving good wear resistance. If the shaft is integral with the ball, strength or ductility requirements for the shaft may dictate ball material.

In ball valves in liquid-oxygen and hydrocarbon-fuel service, the balls are fabricated of high-nickel alloys such as Inconel 718 and A286. Dense chrome plating, 0.0001 to 0.0003 in. (2.54 to 7.62  $\mu\text{m}$ ) thick, is applied to the balls to minimize ball wear. Ball materials in current use in storable-propellant service include 302 and 17-4 PH steels.

## 2.12.4 Blade Valving Element

The material requirements for a blade are similar to those for a ball valving element. Typical materials in use are 347 and 17-7 PH steels.

### 2.12.5 Sleeve Valving Element

A specific requirement for sleeve material is hardness adequate to withstand scratching by the sleeve seal during valve operation. The sleeve of the M-1 sleeve valve is made of Inconel X-750 flame-sprayed with a nickel-base alloy. This process results in a sleeve hardness of Rockwell C58 to 61. Such hardness is required because the mating seal is Kel-F with a comparatively high degree of interference that tends to scratch the sleeve. The poppets of the main propellant valves on the F-1 engine have a pressure-balancing-piston section that functionally is the same as the corresponding section of a sleeve valving element. The poppets are fabricated of 6061-T6 chromic-acid anodized. The pressure-balancing-piston seal is fabricated of Teflon.

### 2.12.6 Cylindrical Slide Valving Unit

The spool-valve configuration typically incorporates both a spool and a stationary sleeve (or housing) of 440C steel. The 440C usually is fully hardened because the hard surfaces will shear contaminant particles and in addition can be ground to sharp corners and fine finishes for smooth operation. Fabricating the spool and the stationary sleeve of the same material eliminates problems due to differential thermal contraction. Like-on-like sliding surfaces can, of course, cause galling, but the 440C combination is used routinely in hydraulic-fluid service without galling.

In the four-way slide valve on the Titan II engine, however, galling of the 440C slide with the 440C stationary sleeve occurred in hydrazine service (ref. 30). Hydrazine is a poor lubricant and apparently did not provide sufficient lubrication in this application. A dry-film lubricant eliminated the galling. The spool and stationary sleeve of the four-way valve of the F-1 engine are also fabricated of 440C; the service fluid is RP-1, which has good lubricating properties. Minor corrosion occurred; however, a material change was not feasible, so the corrosion resistance of the 440C was improved by finishing the entire surface to 16  $\mu$  in. (0.41  $\mu$ m) AA or better.

### 2.12.7 Elastomeric or Plastic Seal for Valving Unit

In storable-propellant service, the most important property of an elastomeric or plastic seal is compatibility; in cryogenic-propellant service, the strength and ductility at cryogenic temperature are most important. Elastomers are not currently used in cryogenic service, although some compounds have been used as static seals in the past.

Elastomers that can be used in hydrazine service include butyl rubber, Buna-N, neoprene, natural rubber, ethylene propylene rubber, styrene-butadiene rubber, and silicone rubber (ref. 14). Resin-cured butyl rubber O-rings are used in the IRFNA feed system of the Lance engine, where the period of exposure is limited to a maximum of 5 minutes. Nitroso rubber is used as static-seal material in the LMDF nitrogen tetroxide feed system; however, because nitroso has poor elastic memory, a satisfactory seal was obtained only by using an O-ring larger in cross section than specified by military standards (ref. 14). The Intelsat thruster valve had a small stroke ( $<0.010$  in. (0.254 mm)); a mere 0.001 in. (0.025 mm) swell or 0.001 in. (0.025 mm) compression set of the seal changed the engine thrust by more than 10 percent. The best compound for this service (hydrazine, water, and isopropyl alcohol) was found to be an ethylene propylene terpolymer with a (Shore A) hardness value of 90 (ref. 14).

Teflon generally is used as the material for a dynamic soft seat or seal in storable-propellant service, although Kel-F poppets are used in at least one storable-propellant application. Although Kel-F is reported to be compatible with  $N_2O_4$  (ref. 41), its strength in  $N_2O_4$  service decreases rapidly; after a 4-hour exposure to  $N_2O_4$ , Kel-F is no stronger than Teflon. When exposed to amine fuels, Kel-F develops cracks and checks that can damage the sealing surfaces.

Both FEP Teflon (fluorinated ethylene propylene) and Teflon (tetrafluoroethylene) are used in storable propellants. The advantages of FEP Teflon are lower thermal expansion and contraction, more uniform properties, and more resistance to erosion. In strength, FEP Teflon and Teflon are very similar. Both materials can be filled with substances such as glass or ceramics that substantially increase the mechanical strength properties. Filled Teflons, however, are less homogeneous than unfilled and therefore are more difficult to finish to the fine surface finishes required for sealing.

A valve on the Surveyor spacecraft incorporated a conical steel poppet with a sprayed-on Teflon coating. During cycling, the Teflon wore and peeled off (ref. 15), and excessive leakage resulted. Because of poor adhesion properties, sprayed-on Teflon coatings are seldom used.

Poppet valving units. – Various plastics including Kel-F, Teflon, Mylar, Vespel, and Kynar have been used as seat or seal materials in poppet valves for service in cryogenic or hydrocarbon fuels. Compared with Teflon, Kel-F has higher strength, a smaller coefficient of thermal expansion and contraction, and is less subject to creep. However, a useful characteristic of FEP Teflon is its capability of being bonded to steel; bonding is a good method for attaching soft material to seat retainers. Mylar, Vespel, and Kynar have good strength properties. Mylar film, however, is available in a maximum thickness of 0.014 in. (0.356 mm), and its use is therefore limited. A 0.010-in. (0.254 mm)-thick Mylar seal, used as a primary seal on a vent valve, separated into two 0.005-in. (0.13 mm)-thick sections after cycling and then leaked (ref. 67). Mylar film 0.014-in. (0.356 mm) thick is reported to be

prone to such delamination; the 0.0075-in. (0.19 mm)-thick film has been used successfully with little or no delamination difficulties. Mylar generally is not used in liquid-oxygen valves, because it has produced positive reactions in LOX impact-sensitivity tests at high impact-energy levels.

Butterfly valving units. — Mylar, Kel-F, Teflon, Vespel, and Kynar have also found application for shaft lip seals and for primary valving element seals on butterfly valves. Mylar sheet offers a combination of high strength and low-temperature elasticity that makes it ideally suited for many lip-seal applications. However, it is limited as described above by a maximum thickness and by questionable compatibility with oxygen. Mylar has been used successfully as a shaft seal for liquid-hydrogen butterfly valves operating in the 1900 psi (13.1 MN/m<sup>2</sup>) range. Kel-F and Teflon also are used for lip seals in a wide range of temperatures and fluid environments; however, cold-flow creep presents a problem that requires careful analysis in the seal design. Kel-F is used as the butterfly disk seal material for cryogenic- and hydrocarbon-fuel service because of its relatively high strength and flexibility at low temperatures. Both hot-formed Kel-F and machined Kel-F disk seals are in use in cryogenic service. Hot-formed Kel-F seals have better ductility and are perhaps less likely to break under gate impact at closure, but machined Kel-F seals have better dimensional stability; they are less likely to exhibit detrimental dimensional changes during temperature excursions and can be fabricated to closer tolerances.

Kel-F has also been utilized as the primary butterfly disk seal in storable systems. Conversion of the Titan I to a storable system (with a ground rule of making a minimum of changes) resulted in a Kel-F seal in N<sub>2</sub>O<sub>4</sub> service and a polypropylene seal for service in amine fuel. However, the Kel-F seal weakened after exposure to N<sub>2</sub>O<sub>4</sub>, and the operational life of the valve was essentially limited to one cycle. Another program utilized Kel-F seals for service in the amine fuel and Teflon for service in N<sub>2</sub>O<sub>4</sub>. The Kel-F seals turned black and became brittle, while the Teflon seals were too weak and folded under the gate at closure. Both butterfly disk seals then were fabricated of Kynar. Satisfactory performance was obtained for a period of 6 months, although the Kynar seal turned black and lost strength in both propellants.

Both Kynar and nylon have been evaluated as disk seal materials for hydrocarbon-fuel service, and nylon was used in the fuel system of an early ballistic missile; however, dry cycling tests show that both materials exhibit more wear than is shown by Kel-F. Nylon also is hygroscopic and undergoes undesirable dimensional changes and loss in strength as its moisture content increases. Kynar is a relatively new material in this application and therefore has less history than Kel-F or Teflon. Vespel, a polyimide, is a soft seal material used primarily in low-temperature (−250°F [117 K]) pneumatic valves.

Ball valving unit. — The most critical property of a ball valving unit seal is wear resistance. Both Teflon and Kel-F are used in hydrocarbon-fuel and cryogenic-propellant service. Kel-F is preferred because its greater strength minimizes wear and because it has a smaller coefficient of thermal expansion and contraction. Teflon is used, however, when a low

coefficient of friction is required to minimize actuator size. Seats in valves for storable-propellant service are made of Teflon, because it is compatible with both storable oxidizers and storable fuels. In a bipropellant redundant ball valve on the Apollo Service Module engine, the FEP Teflon primary seals showed wear during dry cycling tests. The seals were changed to Teflon because it was considered that the lower coefficient of friction of Teflon as compared with FEP Teflon would diminish the seal wear. Seal wear problems persisted; therefore, glass and metal oxide fillers were added to the Teflon to increase the compressive strength and wear resistance. Satisfactory performance was achieved, and the valve met operational life requirements.

## 2.12.8 Housing

Important material properties for a housing include corrosion resistance, weight, ease of machining, coefficient of thermal contraction and expansion, weldability, and adaptability to be cast or forged or used as a bearing material. Yield strength and modulus of elasticity of housing materials used in hot-gas service are critical in preventing housing failure and excessive deformation. Aluminum housings are widely used because of their low weight; often, in an integral bellows design, an aluminum housing is brazed to a steel bellows to minimize weight. Housings of aluminum alloys are chromic-acid or sulfuric-acid anodized to provide corrosion resistance; steel housings are fabricated from CRES alloys.

The housing itself must be leak tight. Housings that must be welded in place to obtain an hermetically sealed joint with the mating ducts are fabricated of steel. Although complex shapes are obtained easily with castings, the material properties generally are less attractive than those of bar stock or a forging; poor-quality castings may leak. The casting sometimes is impregnated with a substance such as waterglass (sodium silicate) to minimize leakage. Two examples indicate the kinds of housing-leak problem that have occurred. A propellant-valve housing used with a helium pressurization system lost pressure over a 14-day period because of helium permeating through the casting; the housing material was changed to bar stock. A J-2 engine check valve fabricated of 321 steel bar stock leaked through titanium carbide "stringers" (inclusions) in the housing wall; the material was changed to consumable-electrode-melted 321 steel (commonly designated as vacuum melt) to minimize the presence of inclusions.

Certain high-strength aluminum alloys and steels are subject to stress-corrosion cracking. Stress-corrosion cracking of a material results from the interaction of sustained tensile stress and a corrosive environment. The degree of cracking within a given period of time depends on the material, the environment, and the magnitude of the stress. As an example, a valve housing flange fabricated of 7075-T6 aluminum failed after a relatively short installation period (ref. 68). The flange was sealed with a compression-type seal that required considerable force to compress; a steady-state stress thus imposed on the flange led to the stress-corrosion cracking. Residual stresses in the housing also probably contributed to the failure.

Stress-corrosion cracking of the housing is minimized or avoided by choosing materials that are not subject to corrosion in the intended service environment, by designing to avoid stress concentrations or the imposition of steady-state stress like that in the example, and by processing materials and fabricated parts to relieve residual stresses.

## 2.12.9 Shaft

Important properties for shaft material are corrosion resistance, strength (including endurance strength if high cycle life is required), and hardness for a good bearing surface. Inconel 718 is used in the shafts of the butterfly valves in the LOX and LH<sub>2</sub> feed systems on the Saturn S-II stage. Bearing-race hardness and resistance to atmospheric corrosion are achieved by the use of 440C steel in the main fuel valve (RP-1 service) for the sustainer engine on the Atlas vehicle. The corrosion-resistant properties of 17-7 PH and 300-series alloys make them suitable for use in many highly corrosive applications such as storable oxidizers and hypergolic propellants. However, 17-7 PH is susceptible to stress corrosion and is used only where the shaft is not subjected to sustained tensile or shear stress in storage. Hot-gas valves operating at temperatures to 1200°F (922 K) have performed well with Rene 41 shafts; this material also has good resistance to corrosion by hot gases.

When certain materials are cooled to cryogenic temperature, grain growth may occur; 440C steel, for instance, has been shown to grow on the order of 0.5 percent when cooled to liquid-nitrogen temperature. This growth has been known to cause binding in close-fitting bearings. The same phenomena may also produce distortion of highly finished surfaces and thereby degrade sealing surfaces. Both dimensional growth and surface-finish degradation are virtually eliminated by chilling the detail part to operating temperature before finishing.

## 2.12.10 Bearings

The material properties of prime importance for valve bearings are coefficient of friction for journal bearings and hardness for rolling-contact bearings.

Low friction in metal-to-metal journal bearings is achieved by the use of soft, antigalling metals running against hard, highly finished mating bearing surfaces. In the 8-in. (20 cm) liquid-hydrogen and liquid-oxygen prevalves on the S-II stage of the Saturn vehicle, an Inconel 718 shaft with a thin, dense, highly polished chrome plating runs in a beryllium-copper journal bearing with a thin flashed silver plating (ref. 69); the silver plate, however, has a tendency to ball up and wear off. Unacceptable wear and sticking partially open has occurred in valves with aluminum components; these conditions were corrected by hard anodizing the bearing surfaces. Installation of a glass-filled Teflon sleeve eliminated



binding and galling of a metal-to-metal guide in cryogenic service where lubricants could not be used and bearing stresses were within those allowable for Teflon. Dry-film lubricants have been used to achieve low friction, but these materials can cause problems if worn off; particles from the dry film can contaminate a shaft seal and cause leakage.

Hardness in a rolling-contact bearing is required to sustain the high Hertz stresses in high-performance bearings. The roller bearings on the butterfly valve in the LOX feed system on the upper stage of the Saturn vehicle operate at a Hertz stress of 700,000 psi (4826 MN/m<sup>2</sup>) as a result of surge pressures at engine cutoff. The bearings for this application are 440C hardened to a minimum of Rockwell C 53.

Environmental compatibility is a prime consideration in the choice of bearing materials. For journal bearings or guides in storable or cryogenic service, the fluorocarbon plastics are preferred because they do not corrode. In a severely corrosive fluid, (e.g., liquid fluorine) beryllium-copper guides have been satisfactory in experimental propulsion systems. The 440C alloy commonly is used for rolling-contact bearings in hydrocarbon-fuel and cryogenic-propellant service, and has been used successfully in FLOX in experimental programs. High-strength nickel-cobalt alloys and titanium carbide have been proposed as materials for improved corrosion resistance.

At high temperatures (e.g., bearings operating in hot gas at 1200°F [922 K]), where plastics and the soft metals are not serviceable, flame plating with tungsten carbide or high-temperature alloys such as Haynes 25 have been used.

### 3. DESIGN CRITERIA and Recommended Practices

#### 3.1 POPPET VALVING UNIT

*The poppet/seat sealing-surface combination shall be appropriate for the specified valve performance and conditions of use.*

It is recommended that sealing-surface configurations for a poppet valving unit be selected after weighing the merits of each seat type for specific design conditions, as presented below:

<u>Design Condition</u>	<u>Preferences and Comments</u>
	(S = soft sealing surface preferred) (H = hard seat preferred)
Low leakage	S
Highly contaminated environment	S
Highly reactive propellant	H
High operating pressure	H
Large size	S
Small actuator size	S
High operating temperature	H
Extreme cycle life	H or S
Extended operational life	H
Long storage life	H
Soft-sealing-surface retention	Disadvantage of soft seal
Seal of secondary leak path	Disadvantage of soft seal
Manufacturing requirements and cost	S
Sealing-surface material properties	H
Commonality of design	H

For additional guidance, consult references 70 and 71.

### 3.1.1 Hard-on-Hard Unit

#### 3.1.1.1 GEOMETRY AND SURFACE FINISH

*The geometry and surface finish of poppet and seat shall be achievable by conventional techniques and skills and shall ensure an acceptable leakage rate and cycle life*

A flat seat and poppet is recommended when vibration is limited and a long cycle life is required. Flat lap to obtain a flatness of  $11.6 \mu$  in. ( $0.295 \mu\text{m}$ ) (1 light band of helium) and a surface finish of 2 to  $4 \mu$  in. ( $0.051$  to  $0.102 \mu\text{m}$ ) AA for low-leakage applications. Inspect the flatness with an interference microscope. Use a spherical seat and poppet when vibration is severe and the applied load is light. For poppets, use high-precision commercial spheres having diametral tolerances of  $\pm 0.0001$  in. ( $\pm 2.54 \mu\text{m}$ ) and sphericity of  $0.000020$  in. ( $0.508 \mu\text{m}$ ) or better, the amount depending on size. Use sphere surface finishes of 1 to  $2 \mu$  in. ( $0.025$  to  $0.051 \mu\text{m}$ ) AA and lap the seat to 1 to  $2 \mu$  in. ( $0.025$  to  $0.051 \mu\text{m}$ ) AA in low-leakage applications. Conical poppets and seats should be used only when leakage is not critical, as in a closed-loop hydraulic system. Fabricate conical geometry seats and poppets by machining and precision grinding.

Fabricate flat and spherical poppets and seats by machining and lapping. Establish effective machining operations and written process specifications for lapping to minimize lapping time. Master lap only. Use machine lapping in preference to hand lapping. Follow the information presented in reference 9. Use random motion of the lapping tool and the workpiece; use figure-8 strokes in flat hand lapping and simultaneous rotational and oscillatory motion in machining lapping. Use a grooved lapping tool of Meehanite cast iron. Use aluminum oxide abrasive on comparatively soft materials like 300-series steel and diamond dust on comparatively hard materials like 400-series steel. Spherical hand lap by cementing a 440C steel or tungsten carbide ball to a rod, rotating the tool in a random pattern on the seat, reversing the direction of rotation frequently, and repositioning the tool surface of the ball. Inspect the workpiece frequently during lapping with an interference microscope at approximately 40X magnification.

Use a flexible disk seal when a poppet must simultaneously seal dual seats (as in a four-way valve), all-metal construction is required, and the pressure rating is 750 psi ( $5.17 \text{ MN/m}^2$ ) or less. The flexible disk compensates for seat-to-seat linear mismatch due to tolerance stackups. Linear dimensional tolerances of the detail parts affecting seat-to-seat mismatch should be approximately  $\pm 0.00005$  in. ( $\pm 1.3 \mu\text{m}$ ) so that the seat-to-seat mismatch is in the range of 0.0002 to 0.0004 in. ( $5.1$  to  $10.2 \mu\text{m}$ ). Design the disk for a minimum deflection of approximately 0.001 in. ( $25.4 \mu\text{m}$ ) and a maximum deflection of 0.003 in. ( $76.2 \mu\text{m}$ ). Use the minimum possible disk thickness.

Determine the theoretical seat-to-disk contact angle by calculating the deflection angle of the disk. Use the largest possible radius on the seat at an angle approximately equal to the maximum theoretical seat-to-disk contact angle. Ensure that the disk seats on the radius of the seat, not on the edge. Radius the edge of the seat. Consider a seat geometry and finish similar to that shown in figure 11 for operating pressures up to 750 psi (5.17 MN/m<sup>2</sup>). Use a 4  $\mu$  in. (0.10  $\mu$ m) AA finish on the disk. Do not lap the disk, as lapped surfaces wear faster than rolled surfaces.

### 3.1.1.2 STATIC SEAT STRESS

*Static seat stress shall be low enough to result in the required cycle life and yet high enough to limit the leakage to the specified rate under all storage and operating conditions.*

A static seat stress between 2,000 and 10,000 psi (14 and 70 MN/m<sup>2</sup>) is recommended as a starting point in a new valve seat design. However, the actual seat stress to be utilized usually is not amenable to an analytical verification, but can only be verified by test. There are no successful analytical techniques for establishing a value for stress that would allow the valve poppet and seat to be opened and closed the specified number of times without degrading the surfaces, causing deformations, or possibly changing material properties through work hardening. Similarly, there is no analysis available to the designer that would allow seat stress to be established so that there would be sufficient actuator force available to crush contaminants, or to deform particles or the land or both, to achieve sealing. Therefore, in a new valve design, seat stress values must be verified experimentally. It is recommended that cycle testing be performed; seat leakage measurements made during the test should be used to provide data on wear, deformation, and material properties. In addition, particle-placement tests on the valve seat must be made together with leakage measurements to provide data on contamination tolerance. From these experimental data, the upper limit (stress value determined by cycle life requirements) and the lower limit (stress value determined by contamination tolerance) can be established and verified. The seat design utilized must then be checked to ensure that tolerances on land width from machining operations, tolerances on actuator forces from variations in spring force and actuation pressure, and changes in effective seating area do not result in seat stress values outside of the range shown experimentally to be workable.

Recommended seat bearing stresses for the flexible-disk poppet valving unit range from 5,000 to 50,000 psi (34.5 to 345 MN/m<sup>2</sup>). Use a disk flexure stress that does not exceed the tensile strength of the disk material. Follow the stress analysis procedures outlined in reference 27. Perform a cycling test to verify cycle life.

### 3.1.1.3 TRANSIENT SEAT STRESS

*Poppets and seats shall withstand the operating stresses over the required cycle life.*

Determine analytically the stresses at impact. Use appropriate safety factors on material strength (e.g., 1.32 on yield and 1.5 on ultimate) to obtain allowable values for working stresses. Increase the safety factors if a large number of valve cycles is a requirement. Flat-seat geometry is recommended for high impact load and high cycle life.

Use a "bumper land" slightly below the sealing surface (fig. 31(a)) to limit loads caused by an out-of-parallel flat poppet impacting the seat. Use a modified chamfered edge (fig. 31(b)) for seat configurations subject to extremely high impact stresses.

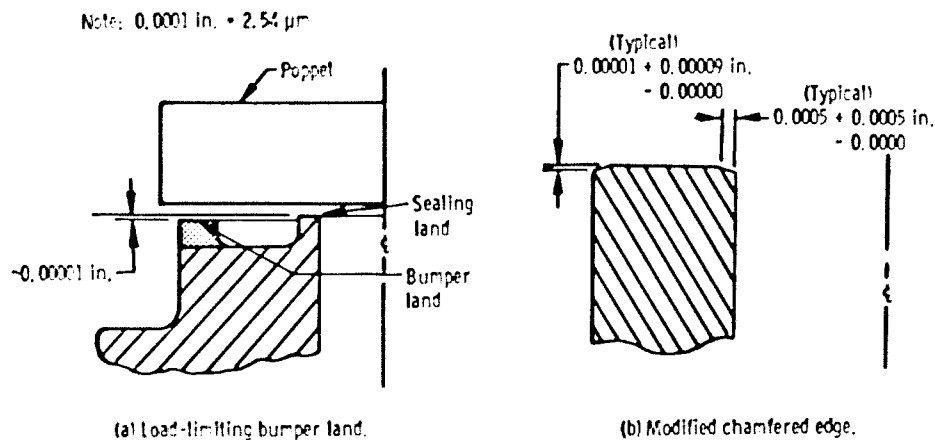


Figure 31. - Cross-section sketches of recommended seat configurations for high-impact poppets.

### 3.1.1.4 ALIGNMENT

*The alignment between the poppet and its seat shall preclude (1) gaps that allow unacceptable leakage and (2) scrubbing during closure that will reduce cycle life.*

Use a self-aligning flat disk actuated by a push rod (fig. 32(a)), or section a standard sphere to provide a flat sealing surface and a spherical surface in a rod end to permit slight rotation for alignment (fig. 32(b)).

For designs that utilize a poppet disk integral with the poppet shaft, misalignments should be minimized. The normality requirements between the poppet sealing surface and the shaft and between the seat and the shaft guides in the housing should be established after critical features have been optimized as follows:

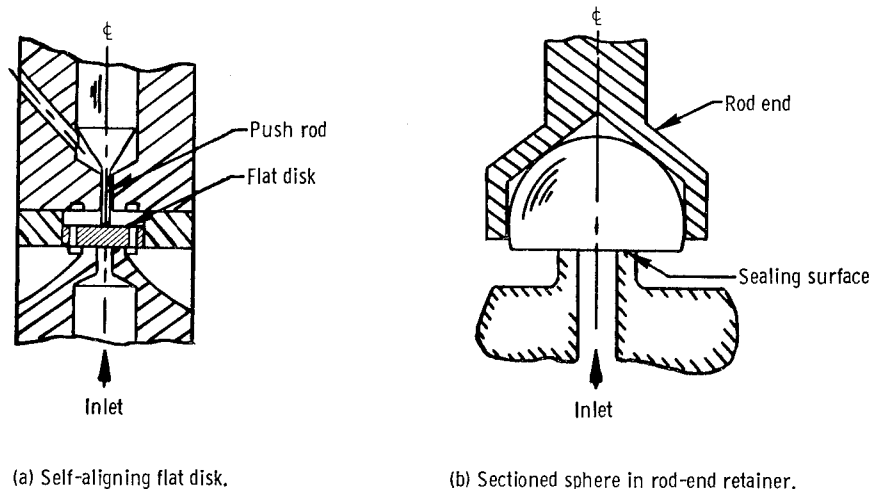


Figure 32. - Cross-section sketches of recommended configurations for poppet alignment.

- (1) Effective shaft length. — Critically review the design to ensure the length between the shaft guide and the sealing surface is the minimum possible, since the longer the unsupported length, the greater the degree of scrubbing at the poppet and seat interface due to misalignment at closure.
- (2) Clearance between poppet shaft and guide. — Make a 10X layout and analyze the geometry, using guide lengths, clearances, and effective shaft length to show what normality differences can be compensated for at the seat diameter utilized. Calculate the maximum permissible normality deviation based on this analysis and establish normality tolerances based on both the design requirement and machining costs.

Provide sufficient clearance between a ball poppet and its retainer so that the ball can self align in its spherical seat. Recommended types of retainers are shown in figure 33.

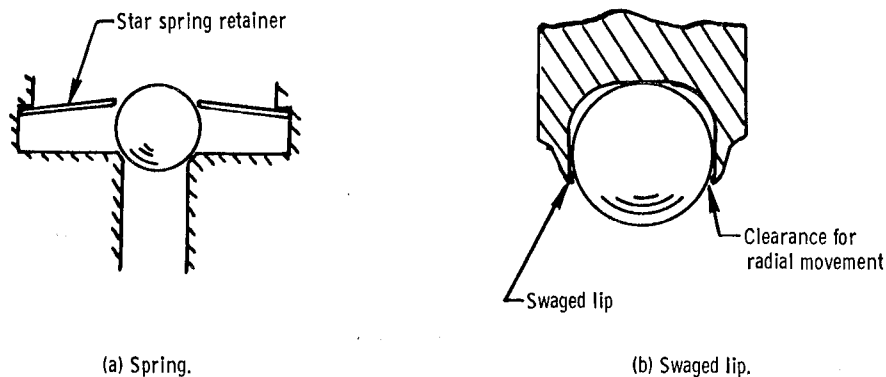


Figure 33. - Cross-section sketches of recommended types of ball-poppet retainers.

Use narrow seat widths with conical poppets and seats to minimize the effect of tilt; a width of approximately 0.005 in. (127  $\mu\text{m}$ ) is recommended.

### 3.1.2 Hard-on-Soft Unit

#### 3.1.2.1 GEOMETRY AND FINISH

*The geometry and surface finish of poppet and seat shall ensure an acceptable leakage rate and cycle life.*

Perform a tradeoff study to determine the advantages of flat, spherical, conical, or combined geometry for a specific application of a poppet valving unit. In small valves, where changes in actuation-force requirements are critical, use a spherical or conical poppet that seats on a corner of a plastic seat or a flat poppet that seats against a narrow seat or a beveled seating surface. In large valves, use conical poppets and seats. Minimize wear by making the hard seat in the housing wider than the plastic seal of the poppet (fig. 34(a)), or by making the metal sealing surface of the poppet wider than the plastic seat (fig. 34(b)).

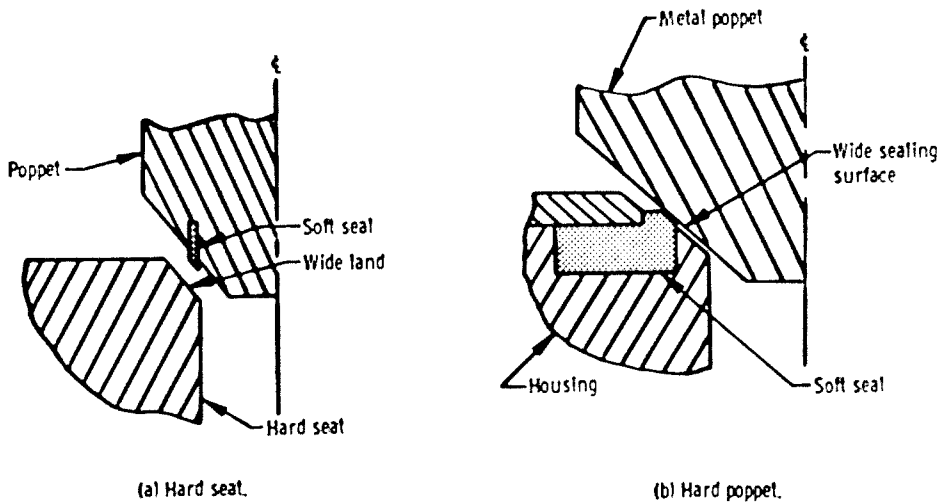


Figure 34. - Cross-section sketches of recommended hard-on-soft sealing configurations.

Use circular-lay surface finishes of 16  $\mu\text{in.}$  (0.41  $\mu\text{m}$ ) AA and 8  $\mu\text{in.}$  (0.20  $\mu\text{m}$ ) AA for a soft seal and hard sealing surface, respectively.

Form small spherical and conical plastic seals by imposing a seat stress of 900 to 2300 psi (6.2 to 15.9  $\text{MN}/\text{m}^2$ ) and heating at 200°F (367 K) for approximately 1 hour (ref. 24).

### 3.1.2.2 SOFT SEAL RETENTION

*The seal retention method shall prevent seal distortion, creep, dislodgement, and other phenomena that result in secondary leak paths or changes in valve pressure drop.*

Analyze failure modes and possible off-design-point testing of the rocket engine system. Determine if any adverse environments such as high temperature could affect the seal retainer or seal bond.

Perform an analysis to determine the magnitude of unbalanced forces during poppet opening and closing and provide adequate retention capability to withstand those forces. Allow for the possibility of seal flange leakage or equivalent forces that could expel the seal. If a poppet nose O-ring seal is to be incorporated, use the design techniques shown in the section on dynamic seals in reference 4. Minimize the area of a seal that is exposed to flow or to large pressure differentials.

Analyze the effect on valve pressure drop of dimensional changes in the soft seal due to allowable squeeze tolerance, elevated temperature conditions, and propellant exposure conditions. When possible, utilize additional stroke so that the valve is not sensitive to small dimensional changes of the soft seat.

Use a threaded or bolted retainer when valve size permits (sec. 3.2.2.2); with this method of retention, seal squeeze can be accurately controlled.

Use bonded seals only where experience has indicated a specific bonding process is satisfactory. When bonded seals are used, conduct 100-percent nondestructive inspection of the bond and destructive testing of a sample from each process lot. Immerse Kel-F or Teflon bonded parts in liquid nitrogen for 2 minutes and then immerse them in water for a total of 25 cycles; inspect the bond. Another method recommended for checking for voids in the bond is to apply a high pressure (e.g., 5000 psi (34.5 MN/m<sup>2</sup>)), quickly vent the pressure, and inspect for voids.

Use swaged, spun-over, or pressfit types of retainers when analyses of dimensional tolerance extremes and proposed manufacturing processes indicate that these types are feasible. Perform a thermal (expansion and contraction) analysis of the seat assembly and establish time-temperature relationships for defining behavior of the soft seat material under mechanical and thermal stress.

Teflon seals retained by the recommended retainers should be stress relieved after installation of the seal and prior to final machining. Subject the part to a temperature approximately 50°F (28 K) higher than the anticipated maximum temperature to which the seal will be exposed during manufacturing, operation, and storage. A recommended



generalized stress-relief process is exposure to 200°F (367 K) for 8 to 10 hours. When welding operations are performed near the seal, limit heat input from the welding by specifying duration of the heat input and duration of the intervening cooling period.

### 3.1.2.3 STATIC SEAL STRESS

*Static seal stress shall be low enough to result in the required cycle life and yet high enough to limit the leakage to the specified rate under all storage and operating conditions.*

Use a design seal stress of 40 to 50 percent of the room-temperature yield strength of the material for a seal in a valve operating at room temperature, e.g., approximately 500 psi (3.4 MN/m<sup>2</sup>) for Teflon seals and 3000 psi (20.7 MN/m<sup>2</sup>) for Kel-F seals. If the valve is also required to operate at cryogenic temperature, use a design seal stress of 70 to 80 percent of the room-temperature yield strength.

Perform a cycling test of plastic and elastomeric seals to verify that the design will meet cycle-life requirements. Verify also that static load on the seal over long storage periods has not induced cold flow of the seal material and thereby changed the seal geometry sufficiently to develop leak paths.

### 3.1.2.4 TRANSIENT SEAL STRESS

*The seal shall withstand the operating stresses over the required cycle life.*

Perform an analysis to determine the magnitude of impact stresses and their effect on the seal. Conduct cycle tests to verify performance.

With a valving unit in the 1/4-in. (6.35 mm) size range, use a metal land or stop 0.0002 to 0.0005 in. (5.1 to 12.7 μm) below a plastic sealing surface to prevent excessive distortion. Perform a cycling test at high temperature to verify the cycle-life capability of plastic and elastomeric seals.

### 3.1.2.5 ALIGNMENT

*Alignment between the poppet and its seat shall preclude (1) gaps that can cause leakage and (2) localized high stress areas that may cause yielding of the soft sealing material.*

Alignment between the poppet and seat required for a soft-seal valve is similar to that for the hard-on-hard unit. However, the soft-seal poppet is more tolerant and can provide conformance with some misalignment present. The recommended practice presented in section 3.1.1.4 for a flat-seat poppet disk integral with poppet rod should be followed for soft-seal valve design with similar geometry.

For soft-seal poppets with spherical or conical sealing surfaces, provide sufficient clearance between the poppet and its guide so that the poppet can self align in the seat at closure.

## 3.2 BUTTERFLY DISK VALVING UNIT

### 3.2.1 Geometry and Surface Finish

*Geometry and surface finish of the disk and seal shall ensure effective sealing under all storage and operating conditions and provide the required cycle life.*

Use a spherical disk with diametral tolerances of  $\pm 0.002$  in. ( $51 \mu\text{m}$ ) and sphericity within  $0.0005$  in. ( $12.7 \mu\text{m}$ ). Use the following procedure to determine the disk radius R1 (fig. 35) at which to locate a vertical surface that is slightly smaller than the anticipated minimum seal radius and provides a “shoehorn” to enter the disk into the seal:

- (1) Determine the minimum specified diameter of the seal. If the seal is to operate at cryogenic temperature, subtract the contraction of the seal diameter from the minimum allowable seal diameter. If the seal flange is rigidly clamped to the housing, calculate the thermal contraction of the seal as if the thermal coefficient of contraction of the seal were the same as that of the housing.
- (2) Determine the required disk radius R1 by subtracting  $0.005$  in. ( $127 \mu\text{m}$ ) (an estimate of the decrease in seal radius due to deflection caused by flow forces) from the minimum seal radius at its minimum operating temperature. Provide a vertical surface on the side of the disk that contacts the top of the seal during valve closure.

Then provide a radius R2 to blend the vertical surface into the spherical diameter of the disk. The radius R2 should blend in the vertical surface over an included angle of approximately  $100^\circ$ . Radius or blend all changes in contour of disk surfaces that contact the seal.

Make a 10X layout of the disk as it begins to contact the top of the seal during valve closure; then check all dimensions to ensure that the contours and the vertical surface of the disk are oriented properly with the lip such that there is no interference that could cause bending and breaking of the lip.

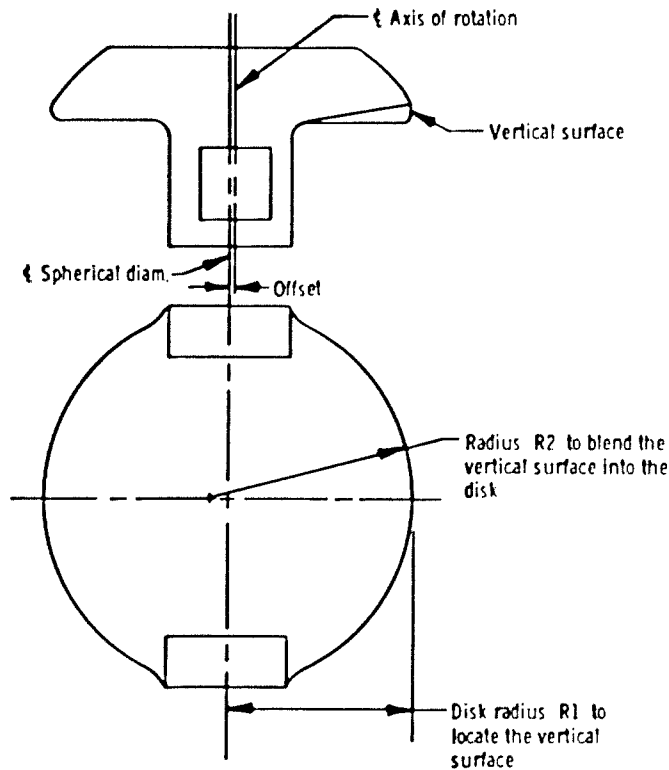


Figure 35. - Schematic for determination of vertical surface on butterfly disk.

Make a 10X layout of the disk and seal contact areas in the open position. Offset the centerline of rotation from the geometric center of the disk to the distance required to preclude contact between the disk in the open position and the seal. Use an offset (fig. 35) ranging from 0.062 in. (1.57 mm) for a 4-in. (10 cm) valve to 0.080 in. (2.03 mm) for an 8-in. (20 cm) valve.

With Kel-F disk seals, use thicknesses and free heights (fig. 18) as indicated in Table II for valves closing in approximately 100 msec.

If seal breakage or folding is a potential problem, perform component blowdown tests simulating actual operating conditions to determine final configuration and material processing. Monitor tests of the butterfly valve on the rocket engine or vehicle to ensure satisfactory seal rigidity.

Analyze the probable deflection of the seal, disk, and shaft of large butterfly valves to ensure that the seal will follow the disk under all conditions, particularly cryogenic

Table II. — Recommended Dimensions for Kel-F Disk Seals

Valve size		Operating pressure		Seal thickness		Seal free height	
in.	cm	psi	MN/m <sup>2</sup>	in.	mm	in.	mm
8	20	100	0.69	0.060	1.52	0.140	3.56
3 to 4	7.6 to 10	500	3.45	0.062	1.57	0.125	3.18
3 to 4	7.6 to 10	900	6.2	0.075	1.91	0.130	3.30
3 to 4	7.6 to 10	1200	8.3	0.090	2.29	0.130	3.30
3 to 4	7.6 to 10	1500	10.3	0.125	3.18	0.165	4.19

temperature. The deflection of the seal must be greater than the sum of the deflections of the disk and shaft plus the allowable bearing clearances.

Use a 0.010- to 0.030-in. (0.25 to 0.76 mm) radius on the sealing edge of the seal. Use a surface finish of 8  $\mu$  in. (0.20  $\mu$ m) AA for the disk and 32  $\mu$  in. (0.81  $\mu$ m) AA for the sealing surface of the seal that contacts the disk.

## 3.2.2 Seal Retention

### 3.2.2.1 RETAINER CONFIGURATION AND LOADS

*The retainer shall retain the seal and provide sufficient load to prevent secondary leak paths under all storage and operating conditions.*

Use a bolted-on retainer with a serration that is approximately 0.010- to 0.015-in. (0.25 to 0.38 mm) high and has a 0.005-in. (0.13 mm) maximum radius at the edge (fig. 36). Design the retainer and bolt loads such that the serration penetrates 100 percent and the area in bearing increases to provide approximately 1500 psi (10.3 MN/m<sup>2</sup>) bearing stress with a Kel-F seal and 500 psi (3.4 MN/m<sup>2</sup>) with a Teflon seal. This comparatively low bearing stress minimizes the possibility that the portion of the seal that is clamped between housing flange and retainer will creep under storage conditions.

Use comparatively high bolt loads to minimize the possibility of thermal or pressure transients unloading the seal flange.

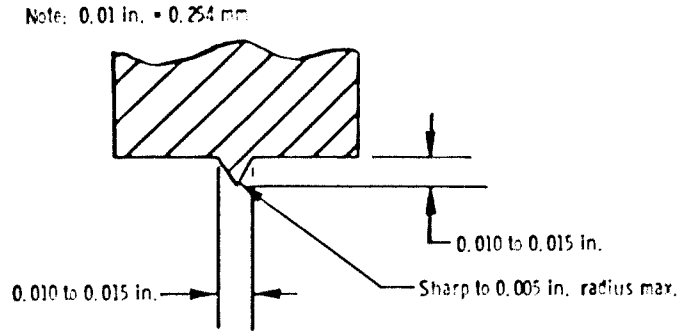


Figure 36. - Sketch showing recommended dimensions for seal-retainer serration.

### 3.2.2.2 THERMAL CONTRACTION OF RETAINER BOLTS

*The retainer bolt loads shall remain constant over the range of operating temperatures.*

Add an Invar spacer to the seal, retainer, and retainer-bolt installation (fig. 37). Select a length of Invar spacer that will make the thermal contraction of the spacer, retainer, and seal approximately equal to that of the corresponding length of the retainer bolt.

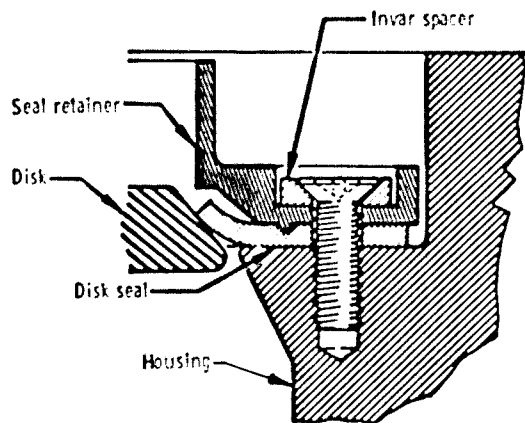


Figure 37. - Cross-section drawing of Invar spacer installation for thermal compensation of retainer bolt.

### 3.2.3 Interference of Disk and Seal

*The interference between the disk and seal shall provide effective sealing without excessive wear or breakage.*

Use a diametral interference of approximately 0.005 to 0.040 in. (0.13 to 1.02 mm) between a Kel-F seal and the disk; i.e., make the diameter of the seal 0.005 to 0.040 in. (0.13 to 1.02 mm) smaller than the distance across the disk at the point where the disk and seal first contact during closure. If manufacturing tolerances permit, make the maximum interference even smaller than 0.040 in. (1.02 mm), so that seal breakage during valve closure is held to a minimum.

### 3.2.4 Alignment of Disk and Seal

*The alignment of the disk and seal shall preclude any disk-to-seal eccentricity that will result in unacceptable leakage.*

Eliminate unbalanced lateral pressure forces on the shaft by making shaft areas equal. Develop assembly techniques that establish and maintain centerline-to-centerline alignment of the disk and seal. Fix the disk position; permit the seal to align itself with the disk and then fix the seal in position. Establish and enforce written assembly procedures.

## 3.3 BALL VALVING UNIT

### 3.3.1 Geometry and Surface Finish

*Geometry and surface finish of the ball and seal shall ensure effective sealing under all storage and operating conditions and provide the required cycle life.*

Make 10X layouts and study the tolerances of the ball and seat to ensure that the seat makes 360° contact with the ball in the open and closed position. Use ball sphericities within 0.0005-in. (13  $\mu\text{m}$ ) total indicator runout (TIR). Specify spherical diameter tolerances ranging from  $\pm 0.0008$ -in. ( $\pm 20.3$   $\mu\text{m}$ ) in the 1.37-in. (3.48 cm) ball size range to  $\pm 0.0015$ -in. ( $\pm 38.1$   $\mu\text{m}$ ) in the 4-in. (10 cm) ball size range for leakages in the 15 scim (14,748 scch) and 50 scim (49,161 scch) range at room and cryogenic temperature, respectively. A ball surface finish of 4  $\mu$  in. (0.10  $\mu\text{m}$ ) AA and a seal surface finish of 16  $\mu$  in. (0.41  $\mu\text{m}$ ) AA with circular lay are recommended for low leakage rate. At the intersection of the flow bore

with the ball, use an edge radius 2.5 to 4.5 percent of the ball diameter (larger values are preferred) and a radius surface finish of  $4 \mu$  in. ( $0.10 \mu\text{m}$ ) AA. Consider use of both upstream and downstream seals when high reliability is required; when both seals are used, make provision for venting between the two seals. Avoid leaving the ball in intermediate open positions. Fabricate the seal to spherical geometry by first machining it to a conical shape and then using a controlled-load, rotating test ball to wear the seal into the proper spherical geometry. If inertial loads appear high, consider making the ball hollow to reduce shaft stress.

### 3.3.2 Seal Retention

*The seal retainer shall retain the seal, provide adequate preload, ensure effective sealing at the housing/retainer interface, and limit pressure loads in high-pressure service.*

Calculate the maximum load resulting from an initial preload (sec. 3.3.3) and maximum pressure acting on an assumed seat and retainer configuration. If this combination of loads results in excessive seal loads, consider use of a bellows-type seal retainer. Use a bellows-type retainer in cryogenic service also. Dimensions involved in effective seal loading are illustrated in figure 38.

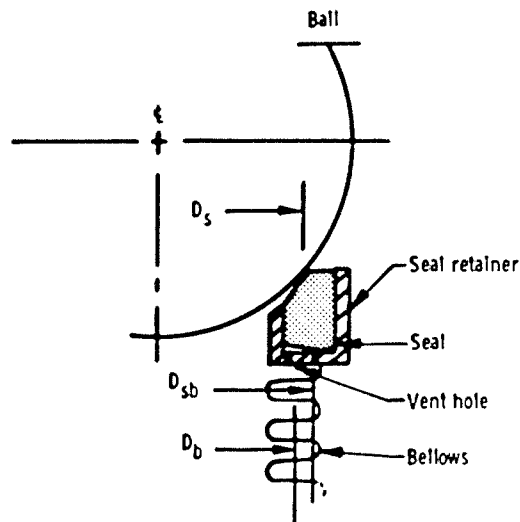


Figure 38. - Dimensions Involved in effective ball seal loading by a bellows.

Select the ball sealing diameter  $D_s$ , the diameter of the seal on the bellows side  $D_{sb}$ , and the effective diameter of the bellows  $D_b$  such that there is a positive force tending to hold the seal in its cup-shaped retainer and a positive force tending to load the bellows against the ball; e.g., with an externally pressurized configuration,  $D_{sb} > D_s > D_b$ .

If a spring-loaded seal retainer (fig. 21(a)) is chosen, then make provision for sealing the interface between the housing and the retainer with an O-ring or spring-loaded Teflon cup seal.

Provide vent holes of sufficient capacity to vent all allowable leakage. Assume all leakage is past the seal at its interface with the retainer.

### 3.3.3 Seat Stress

*The seat load shall be sufficient to prevent unacceptable leakage but not high enough to cause excessive wear.*

Determine the allowable seat stress by assuming a finite area of contact between the ball and seal, even though initial geometry may produce only line contact of the seal. For a finite area of contact with a glass-filled Teflon seal, use a seat load that will develop 190 psi (1.31 MN/m<sup>2</sup>); for unfilled-Teflon seals, a lower seat stress should be employed. If line contact is assumed, and the seal is made of Kel-F, a ball seal preload of 30 pounds per inch (52.5 N/cm) of seal circumference is recommended. If the ball seal is bellows mounted, limit the differential area that loads the seal against the ball to approximately 3 percent of the sealed area. To provide insight on the wear process, a wear analysis using methods presented in reference 72 should be made. Conduct cycle tests and leak checks to verify seal performance at the specified operating temperatures and pressures.

### 3.3.4 Alignment

*Alignment provisions shall ensure concentricity of the ball and seal adequate for effective sealing.*

To align the ball of an assembly that has integral shafts and is laterally pressure balanced, allow the ball to self align with the installed seal during assembly. Use the actuator attachments to the shaft to fix the ball in position.

With an assembly that is laterally unbalanced, the ball must be aligned during assembly, since the stackup of tolerances in the detail part of the valve will usually create a



misalignment condition. Devise a procedure for measuring the misalignment. On a ball valve with a bellows-mounted seal, for example, measure the out-of-parallelism between the flange of the bellows seal assembly and the mating interface on the ball valve housing, while the ball is positioned against the lateral thrust bearing. From this measurement, calculate a shim thickness. During subsequent assembly, add the shims between the ball and the thrust bearing until the ball is well-aligned. The maximum allowed misalignment is determined by cycle testing the valve with a known misalignment condition created by decreasing the shim thickness. As misalignment increases, excessive seal wear will occur. Shim thicknesses should be established to provide satisfactory adjustments of alignment.

Establish and enforce written assembly procedures because assembly operations with a laterally unbalanced ball valve configuration are complicated. Generally ball valves with large unbalanced lateral forces should be avoided because the misalignment problems are severe.

## 3.4 BLADE VALVING UNIT

### 3.4.1 Geometry and Surface Finish

*Geometry and surface finish of the blade and seal shall ensure effective sealing under all storage and operating conditions and provide the required cycle life.*

A blade that opens to permit flow by aligning a hole in the blade with the flow passage (fig. 23) is recommended. Use a blade surface finish of  $8 \mu$  in. ( $0.20 \mu\text{m}$ ) AA, flatness within 0.001 in. ( $25 \mu\text{m}$ ) TIR, and the sides of the blades parallel to each other within 0.001 in. ( $25 \mu\text{m}$ ) TIR. A seal surface finish of  $16 \mu$  in. ( $0.41 \mu\text{m}$ ) AA is recommended. If space considerations dictate use of the more conventional blade valving unit (fig. 4), wherein the blade slides off and on the seal, incorporate a  $0.035 \pm 0.005$  in. ( $0.89 \pm 0.13$  mm) radius on the blade edges and a  $7.5^\circ$  taper on the seal edge; in addition, provide the recommended surface finish, flatness, and parallelism for the blade top and bottom surfaces.

### 3.4.2 Seal Retention

*The seal retention method shall retain the seal and shall ensure effective sealing at the seal/housing interface.*

Use a machined piece of plastic that is installed in a close-fitting bore in the housing on the downstream side of the blade. Relieve the bottom of the seal to provide a narrow sealing surface at the interface of the seal and housing (fig. 22).

### 3.4.3 Seal Stress

Design criteria and recommended practices for seat stress are similar to those set forth for other soft-seal valves (sec. 3.1.2.3), the seat load being supplied by a wave washer or Belleville spring that loads a pressure plate on the upstream side of the blade (fig. 22).

### 3.4.4 Alignment

*The alignment between the blade and seal shall provide the parallelism necessary for effective sealing under all storage and operating conditions.*

Use clearances between the blade and shaft that will permit the blade to self align. Note that loose clearances may cause yielding of the drive elements of the blade and seal. Retain the pressure pad with a threaded retainer. An example of a recommended blade valve assembly is shown in figure 22.

## 3.5 SLEEVE VALVING UNIT

### 3.5.1 Rotary Sleeve Unit

*The rotary sleeve and its mating part shall not be subject to detrimental deflections and shall provide the required control of leakage rate under all storage and operating conditions.*

Since this valve is not used in shutoff applications, use the specified leakage rate permissible for the valve as the basis for calculation of the allowable clearance between the rotary sleeve and mating stationary part. Determine the minimum clearance by making a stress and deflection analysis of the rotary sleeve and stationary part under the extremes of temperature and pressure conditions to which the valve will be exposed. Perform a tolerance stackup to establish allowable eccentricities that will ensure that the clearance can be maintained with standard machining tolerances. Use a surface finish of 16  $\mu$  in. (0.41  $\mu$ m) AA on the rotary sleeve and its mating stationary part. Consider the use of a smaller diameter but longer gate sleeve to provide the flow area, since large sleeve diameters increase inertial effects and result in increased actuator shaft size.

## 3.5.2 Linear Sleeve Unit

Design criteria and recommended practices for linear sleeve valves are not included in this monograph because the current technology, which is limited to the development of the M-1 engine sleeve valve, was not verified by flight tests and operational use of the M-1 engine and valve.

## 3.6 CYLINDRICAL SLIDE VALVING UNIT

### 3.6.1 Spool Unit

#### 3.6.1.1 GEOMETRY AND SURFACE FINISH

*The diametral clearance and surface finish of the spool outer diameter and its mating bore shall ensure smooth actuation and effective sealing under all storage and operating conditions.*

References 73 and 74 should be reviewed when establishing basic design values for a specific spool valve design. Recommended methods of fabrication are match lapping or selective fitting. Generally, these valves are utilized in environments with narrow temperature excursions.

### 3.6.2 Cylindrical Unit with Internal Seal

#### 3.6.2.1 GEOMETRY AND SURFACE FINISH

*The geometry and surface finish of the cylindrical slide and its seal shall ensure smooth operation and effective sealing under all storage and operating conditions.*

Recommended practices are limited to those used in the one known flight operational configuration (fig. 24). Use a 0.040-in. (1.0 mm)-thick Kel-F seal with a 32  $\mu$  in. (0.81  $\mu$ m) AA sealing-surface finish. Recommended radii and surface finishes for the slide are shown in figure 39.

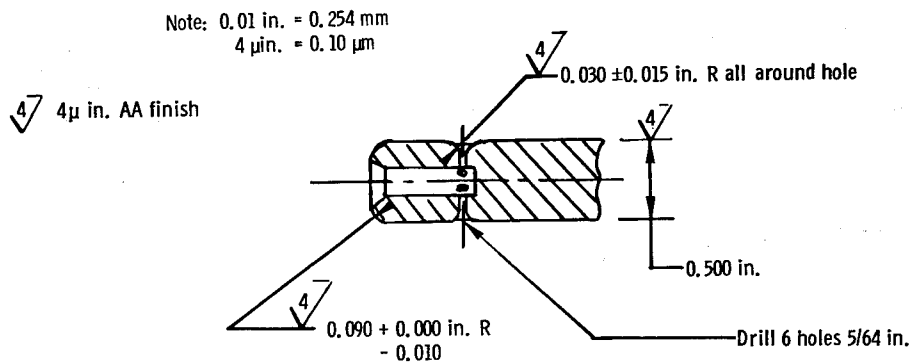


Figure 39. - Cross-section sketch showing geometry and surface finish recommended for an 0.5-in. cylindrical slide.

### 3.6.2.2 SEAL RETENTION

Design criteria and recommended practices are similar to those given in section 3.2.2.

## 3.7 GENERAL CONSIDERATIONS IN VALVING UNIT DESIGN

### 3.7.1 Vibration

*Relative motion between the valving element and seal due to vibration shall not result in excessive wear on the sealing surfaces or increased leakage.*

The recommended approach to design of a vibration-resistant valve element and seal in general is to minimize the available clearances in bearings and guides, avoid large overhung moments, and restrain any lateral motion of the poppet element that can cause scrubbing and wear (e.g., use flexure disks or Belleville springs on short-stroke poppets). Analytical approaches to evaluate a design for possible resonant-vibration problems are desirable; however, the complexity of most valve elements and connecting actuators makes such analysis extremely difficult. Section 13.3.5 of reference 4 should be consulted for equations that describe simple vibration systems. Analysis of analogous but simplified vibration systems that reasonably simulate the valve element mechanism is recommended where feasible. Such analyses can reduce the number of design changes that may be required later

during vibration testing. It is recommended that vibration tests be made early in the development of a valve design, since these tests provide a degree of design verification that is not possible through analysis. Early tests should be based on vibration levels predicted in an engine or vehicle design; later tests should use actual vibration levels measured during static and flight tests.

## 3.7.2 Contamination

### 3.7.2.1 CONTAMINATION TOLERANCE

*The valving element and seal shall seal effectively even when the system operating fluid contains contaminant particles up to the maximum specified level.*

Keep the propellants as clean as practicable. Use filters at the inlets of small valves.

When practicable, in a system containing appreciable contamination, use butterfly or ball valves. The rotating elements have a sliding contact at closure that tends to wipe off contaminants and thus increase the valve tolerance to contamination.

It is recommended that the seat load in a poppet valving unit be high enough to envelop or crush a trapped particle in addition to providing a satisfactory seat stress. Therefore, perform static particle-placement tests on the proposed configuration to establish the required overload value; the resulting seat stress should be in the range specified in sections 3.1.1.2 and 3.1.1.3. The stroke of a poppet valving element should be sufficient to prevent particles from being trapped between the poppet and seat while the valve is open. When valve size permits, use a flexure-mounted or Belleville-spring-mounted poppet instead of close-fitted sliding members.

### 3.7.2.2 CONTAMINATION CONTROL

*The design, construction, assembly, and handling of valve components shall not result in contamination in the valve.*

It is recommended that the information presented in references 4, 41, and 43 be used as a guide to minimize the quantity and effects of contaminants. Platings and threads should be avoided if possible and limited to applications where quality is very closely monitored. Use rolled threads instead of machined threads. Do not use like-on-like materials for components that move relative to each other. Use hardened materials with good surface finishes on sliding parts to avoid galling and wear and to reduce the size of wear particles.

All passages should have a straight-through flow, if possible. Dead-end passages, capillary passages, tees, bends, and fittings that could trap particles should be avoided. Eliminate small entrapped volumes by avoiding overlapping surfaces between mating components. In addition, mechanical attachments such as threads, rolled lip, snap rings, and overlapping retainers should not be used.

On detail parts, remove all burrs that are discernible by normal unaided vision. When clearances and sealing requirements are critical, remove all burrs discernible at 10X magnification.

### **3.7.2.2.1 Cleaning**

*Procedures for cleaning valve components shall be specific for the component and for the degree of cleanliness required.*

All valve components should be cleaned in accordance with a detailed specification. Recommended cleaning procedures for metallic valve components include a precleaning operation such as vapor degreasing to remove dirt, corrosion, and grease from metallic parts, excluding anodized or passivated parts and dry-film lubricated parts, and a final cleaning operation. Clean dry-film lubricated details by vapor degreasing or flushing with a clean halogenated solvent. Use the information in reference 47 and the procedures set forth in section 2.7.2.2.1 for precision cleaning of metallic components, general cleaning of metallic components, and cleaning of nonmetallic items.

### **3.7.2.2.2 Contamination Barrier**

*A contamination barrier shall protect the cleanliness of the detail part.*

Use a clean, heat-sealed 0.002-in. (51  $\mu\text{m}$ )-thick nylon and a 0.006-in. (152  $\mu\text{m}$ )-thick polyethylene exterior bag for small detail parts where sloughed off particle size and quantity may be critical. Use a 0.006-in. (152  $\mu\text{m}$ )-thick polyethylene bag for other detail parts. Heavy details or details having threads, sharp points and edges, or other puncture-making features should be enclosed in a 0.006-in. (152  $\mu\text{m}$ )-thick polyethylene bag and overwrapped with a sufficient amount of 0.006-in. (152  $\mu\text{m}$ )-thick film, secured with tape, and then placed in another heat sealed 0.006-in. (152  $\mu\text{m}$ )-thick polyethylene bag. For a LOX-compatible barrier, use 0.002-in. (51  $\mu\text{m}$ ) Aclar film.

Mark the full width of the bag, top and bottom, with appropriate lettering (e.g., the manufacturer's name). Apply tamper proof decals on the barrier to indicate the specification number for the cleaning process. Follow the guidelines presented in references 47 and 48.

## 3.8 HOUSING

### 3.8.1 Configuration

*The valve housing shall*

- (1) contain the service fluid and support the valve components*
- (2) permit installation and removal of the components*
- (3) satisfy the required constraints on envelope, size, and weight.*

Use a one-piece housing in preference to a two-piece housing to reduce weight. In ball valves, use a one-piece housing and insert the ball through the bore for the outboard bearing assembly. Use a one-piece housing for a butterfly valving unit and small poppet valves. Sleeve valves, large poppet valves, blade, and cylindrical slide valves will usually require two-piece bolted-together housings. Consider the available envelope when establishing the housing design; e.g., if ducting interfaces dictate an elbow, use a 90°-angle poppet. Utilize sections that are stressed to the design allowables so that size and weight are kept as low as possible.

### 3.8.2 Strength

#### 3.8.2.1 INTERNAL LOADS

*The housing shall withstand specified and predicted steady-state and transient pressure loads and thermal stresses without structural failure or detrimental deformation.*

In the design of the housing, use the largest practicable internal radii wherever possible.

Verify structural integrity by a stress analysis based on the maximum pressure, either steady-state or transient, that the housing will be exposed to in operation; use minimum guaranteed material properties for ductile materials but use only 90 percent of the minimum guaranteed properties of brittle materials (those having an elongation of less than 3 percent). Use appropriate stress concentration factors as given in section 14.6 of reference 4.

Perform a proof pressure test at 1.2 to 1.5 times the maximum operational pressure for a minimum duration of 2 minutes. Repeat the proof pressure test for a total of 5 cycles. This procedure will ensure yielding in ductile materials, or it will propagate a failure due to crack formation in brittle material. Use a safety factor of 1.1 to 1.5 between the stress at the proof pressure and the material yield stress (0.2 percent offset). Use a safety factor of 1.5 to 2.0 between the stress at the proof pressure and the material ultimate stress.

If stresses are very high, a preliminary proof pressure that is slightly higher than the design proof pressure can be applied to a preliminary machined housing to cause distortion, if possible. Then finish machine the housing and perform the design proof-pressure test.

Perform all proof tests after the valve is assembled but before conducting valve functional checks.

When the housing operates at a temperature higher than test temperature, the proof pressure at test temperature should be increased by multiplying it by the largest of the following three ratios:

- (1) 
$$\frac{\text{Ultimate strength at test temperature}}{\text{Ultimate strength at maximum operating temperature}}$$
- (2) 
$$\frac{\text{Yield strength at test temperature}}{\text{Yield strength at maximum operating temperature}}$$
- (3) 
$$\frac{\text{Modulus of elasticity at test temperature}}{\text{Modulus of elasticity at maximum operating temperature}}$$

Use a weld efficiency (ratio of yield strength of the annealed parent material to the allowable yield strength of the weld) of 100 percent for a weld that is inspected visually for freedom from voids, cracks, and other flaws; checked for fillet dimensions; inspected by magnetic particle or penetrant; and inspected by X-ray. Use a weld efficiency of 80 percent for a weld that is inspected visually and checked dimensionally; is inspected by magnetic particles or penetrant; and is proof-pressure tested. Use a weld efficiency of 60 percent for a weld that is inspected visually, checked dimensionally, and inspected by magnetic particle or penetrant. Use a weld efficiency of 40 percent for a weld that is inspected visually and checked dimensionally only.

Perform a thermal stress analysis on all housings that may be subject to significant thermal shock; make all section thicknesses as uniform as possible in these housings.

### 3.8.2.2 EXTERNAL LOADS

*The housing shall withstand the stresses that result from external loads acting singly or in combination.*

The external loads of concern are identified at the beginning of section 2.8.2.2.

Calculate the maximum misalignment of the ducting allowed by the specified tolerance extremes and by thermal contraction or expansion; then calculate the loads due to the misalignment. Calculate the loads on the housing due to (1) maximum vibration, shock, and



acceleration loads on the mating ducting, (2) mounting-bolt torque, and (3) static-seal compression force. Do not overlook the load due to the actuator.

Perform a stress analysis on the housing using the worst combination of loads and a safety factor of 1.32 on yield strength of the housing and 1.68 on ultimate strength of the housing, whichever is the more conservative. Use material strength properties in accordance with section 3.8.2.1.

Consider use of an integral actuator and housing or section of housing with a butterfly, ball, or blade valve to reduce alignment problems. Use only rigid connections to the valve inlet and outlet. Use steel inserts to provide threads in an aluminum housing.

### 3.8.3 Static Seals

*Static seals in the housing shall seal effectively against external leakage under all storage and operating conditions*

Use elastomeric O-rings or other types of elastomeric seals for hydrocarbon and storable fuels; consult section 6.3.3 of reference 4 for O-ring gland design. For cryogenic service, use a pressure-actuated U-shaped metal seal with a Teflon coating or a C-shaped Teflon seal with an internal flat helical or segmented C-shaped spring; consult reference 51 for the detail design of these seals. Use 32  $\mu$  in. (0.81  $\mu$ m) AA finishes with a circular lay for the sealing surfaces of the seal and the housing.

## 3.9 SHAFT

### 3.9.1 Dimensional Tolerances and Surface Finish

*Shaft dimensions and clearances shall allow the shaft to move and position the valving element reliably, and the shaft finish shall provide a satisfactory surface for seals and bearings.*

Analyze the clearances between the shaft and mating parts; ensure that the clearances provide freedom from binding, but limit maximum clearances that would adversely affect the control function and induce fretting. Clearance variables must include manufacturing and assembly tolerances, deflections due to structural loads, changes due to thermal gradients, and differences in thermal expansion.

When the shaft provides the inner race for a rolling-contact bearing or a surface for a sliding-contact bearing, specify surface finishes, texture, and profile in accordance with the applicable function as delineated in section 3.10; when a separate inner race is utilized, finish and texture on the shaft is less critical. When seals contact the shaft, specify the surface finish and texture as given in section 3.11 for the applicable seal configuration.

### 3.9.2 Strength

*The shaft shall withstand the stresses from specified and predicted loads over the range of operating conditions.*

Minimize stress concentrations by avoiding abrupt changes in cross section and by providing generous fillets with surface finishes of  $32 \mu$  in. ( $0.81 \mu\text{m}$ ) AA or better. This practice is particularly important where bending or shear stresses are significant. Shot peening may be used to relieve surface stresses in critical areas. In rotary valves where torsional stress levels are high, use a hollow shaft.

Specify manufacturing process control procedures as required to ensure fabrication of a shaft that meets the design requirements for strength and dimensions. Heat treatment, for instance, must be closely controlled to develop full strength without excessive residual stress, and grinding must be closely controlled to prevent grinding checks and excessive residual stress.

Perform a stress analysis to ensure that the shaft has adequate strength. For a rotating shaft, analyze the operating conditions to find the combination of bending, shear, and torsion that produces the maximum stress condition. For a translating shaft, analyze the operating mode that produces the maximum bending and tensile or compressive loads to find the maximum combined stress. Calculate the bending load (if any) by determining the flow force acting at right angles to the shaft and valving element line-of-motion. Establish the loads due to pressure surges on the closed valve element and the loads imposed by the actuator output when the valving element is locked at any full or partially open position. Perform an analysis of impact and vibration loads at the full-open and full-closed positions. Evaluate the effects of temperature on material strength and ductility.

## 3.10 BEARINGS

*The bearing type shall be suitable for supporting the direction (radial, axial, or combination) and magnitude of specified and predicted loads and shall satisfy the friction requirements of the application.*

Perform a tradeoff study to establish the bearing configuration. When friction, alignment, compatibility, and load requirements allow, select a plastic for guide bearing or journal bearing applications, since such bearings are the least complex. Utilize metal guides or journals when the bearing must satisfy load requirements higher than plastics can withstand. However, examine both journal and rolling-contact bearings for rotary motion applications; compare possible increased actuator force and size requirements that may arise from the higher friction of the journal with the requirements for a rolling-contact bearing. When load requirements exceed the bearing supplier's recommended values for journal bearings, utilize a rolling-contact bearing.

### **3.10.1 Guide Bearings**

#### **3.10.1.1 GEOMETRY**

*The guide-bearing geometry shall provide the required alignment of the valving element and the seal.*

Provide the maximum possible overall guide length and minimum length between the guide and poppet, so that angular freedom of motion between poppet and seat centerlines is minimized. An L/D ratio greater than 1.0 is recommended for a one-piece guide. Establish the minimum clearances that will permit operation without interference from temperature effects and deflections due to pressure and structural loads, but make the clearance larger than the maximum-size wear particle in unlubricated bearings. Establish the maximum clearances consistent with the required valving element-to-seat alignment at contact.

#### **3.10.1.2 LOAD CAPACITY**

*The guide bearing shall possess adequate load capacity over the range of operating conditions.*

Analyze all the steady-state and transient operating forces, including flow, vibration, and acceleration. Establish the load capacity required of the bearing; select a bearing material that is adequate for the load and that in combination with the shaft material will meet the friction limits imposed on the design. If the bearing is propellant lubricated, the design must be capable of being dry cycled without excessive friction. Recommended PV factors for plastic guide bearings are given in section 6.8.3.2 of reference 4.

### 3.10.1.3 SURFACE FINISH

*The guide-bearing surface finish shall minimize friction, wear, and generation of contamination.*

Sliding bearing surfaces should be highly finished. The use of Teflon guide bushings is recommended. Plastics or soft bearing alloys should have surface finishes of  $32 \mu$  in. ( $0.81 \mu\text{m}$ ) AA or less. When harder bearing materials are used, surface finishes must be finer. Well-beveled or radiused edges that simplify assembly and minimize edge loading during valve operation must be provided.

## 3.10.2 Journal Bearings

### 3.10.2.1 GEOMETRY

*The journal-bearing geometry shall provide the valving element-to-seat relationship necessary for reliable opening and closing of the valve.*

Analyze bearing clearance and alignment to ensure freedom from binding and compliance with bearing supplier's recommendations. Clearance variables must include manufacturing and assembly tolerances, deflections due to structural loads, changes due to thermal gradients, and differences in thermal expansion. Designs that involve an overhanging load must limit misalignment so that edge loading does not generate excessive bearing pressure. Limit out-of-roundness to 0.0005 in. ( $13 \mu\text{m}$ ).

### 3.10.2.2 LOAD CAPACITY

*The journal bearing shall possess adequate load capacity over the range of operating conditions.*

Analyze static and dynamic loads to ensure that the static bearing capacity is adequate and that the PV factor of the material combination is satisfactory for the required operating life of the valve. Consult bearing suppliers for allowable operating bearing pressures for the application.

### 3.10.2.3 SURFACE FINISH

*The journal bearing finish shall minimize friction, wear, and generation of contamination.*

Specify surface finishes in accordance with the bearing or material suppliers' recommendations. Finishes should not exceed 32  $\mu$  in. (0.81  $\mu$ m) AA for the bearing and 16  $\mu$  in. (0.41  $\mu$ m) AA for the journal.

### 3.10.3 Rolling-Contact Bearings

*The bearing configuration shall be simple, compact, and appropriate for the intended use.*

Use caged bearings for minimum friction and wear generated by ball-to-ball or roller-to-roller contact and for minimum skewing of roller-type bearings. Use full-complement bearings only if the maximum capacity is required, increased friction is acceptable, and skewing can be kept to the minimum necessary for reliable operation. Reliable operation of full-complement bearings should be confirmed by a design verification test performed before acceptance for production.

Define complete design requirements for bearing procurement: load (including the method of load application, rotational speed, and life); mounting (including housing material, clearances, and retention); dimensions (including allowable clearances and finishes); materials (including acceptable substitutes); operational temperatures; lubrication; and cleanliness of the bearing. Follow the guidelines presented in references 54 and 55.

#### 3.10.3.1 GEOMETRY

*The bearing geometry shall provide the valving element-to-seat relation necessary for reliable opening and closing.*

Analyze bearing clearance and alignment to ensure freedom from binding and compliance with bearing suppliers' recommendations. Clearance variables must include manufacturing and assembly tolerances, deflections due to structural loads, changes due to thermal gradients, and differences in thermal expansion. Perform a tradeoff study to determine the advantages of operating the bearing in the operational fluid versus sealing the bearing from the operational fluid and atmospheric moisture.

#### 3.10.3.2 LOAD CAPACITY

*The bearing shall possess adequate load capacity over the range of operating conditions.*

Analyze static and dynamic loads and the effects of maximum misalignment including that due to load. Avoid overhanging loads. Consult suppliers of rolling-contact bearings for information on allowable load capacity of the bearing based on the applicable requirements for load, service environment, and service life. Compile complete information pertaining to the bearing details and perform a detailed load and stress analysis.

### **3.10.4 Thrust Bearings**

#### **3.10.4.1 LOAD CAPACITY**

*The bearing shall possess adequate load capacity over the range of operating conditions.*

Perform a stress analysis to ensure that the thrust bearing has adequate capacity to absorb the loads from the forces acting on the valve. Consider static and dynamic loads and the effects of maximum misalignment including that due to overload.

Consult the bearing material supplier for allowable operating bearing pressure and PV factor of a journal thrust bearing. Use ball bearings for rolling-contact elements when the load can be accepted without exceeding the capacity of the bearing. Consult bearing suppliers for allowable load capacity of a rolling-contact bearing.

#### **3.10.4.2 SURFACE FINISH**

*The surface finish and flatness of the bearing shall be adequate for the application.*

Specify surface finish and flatness of a journal-type thrust bearing in accordance with the bearing supplier's recommendations. Finishes should be on the order of 32  $\mu$  in. (0.81  $\mu$ m) AA for the bearing, 16  $\mu$  in. (0.41  $\mu$ m) AA for the journal; flatness, 0.0005 in./in. Corresponding values for a rolling-contact thrust bearing should be those specified by the bearing supplier.

### **3.11 SECONDARY SEALS**

*The secondary seal shall (1) be suitable for service conditions, (2) limit leakage past the valve shaft over the range of storage and operating conditions, and (3) accommodate allowable radial or transverse shaft motion.*

Perform a tradeoff study to evaluate the advantages and disadvantages of the various types of secondary seals. Use O-rings in pneumatic or hydraulic service and in hydrocarbon and storable fuel service. Use plastic lip seals in cryogenic propellant service. Use face seals in high-temperature or highly reactive propellant service where an all-metal seal is required. Use a bellows when essentially zero leakage is required. Use the U-shaped Teflon seal shown in figure 5 for sleeve seals of approximately 8-in. (20 cm) diameter or larger.

Follow the recommendations on intended application and design details presented in references 3, 4, 12, 13, and 51.

### 3.11.1 O-Rings

*The O-ring shall be suitable for the application and shall not extrude into the clearances or be subject to twisting failure.*

O-ring selection and gland design should be in accordance with the military standard (ref. 56). However, for oscillatory stop-and-start motion, use the large-cross-section O-ring recommended for slow-speed rotary applications, rather than O-rings of small cross section that are utilized in continuous rotary applications. Use the information presented in reference 57 to determine optimum hardness to avoid extrusion. Follow references 56 and 57 to determine if backup rings should be utilized. Provide generous radii on corners and 10° to 20° chamfers to provide a "shoehorn" for ease of installing the O-ring. Where possible, lubricate the O-ring to reduce chance of installation damage and to reduce friction and wear during dry cycling of the valve.

### 3.11.2 Lip Seals

*The lip seal shall provide effective sealing, withstand operating pressure forces, and satisfy the cycle life requirement.*

Select a lip length that will provide the maximum-length leakage path consistent with the bearing pressure between seal and shaft. Lip length on cold-formed seals may be established by the amount of stretch possible without tearing or cracking. Select a seal thickness or combination of multi-ply thicknesses that will provide adequate strength and maximum flexibility. Provide adequate material thickness (or supplementary support such as a garter spring) at the end of the lip to ensure that it will maintain pressure on the shaft. Consider an additional overlapping seal to provide additional lip pressure. In high-pressure service, provide support for the low-pressure side of the seal between the housing and shaft; this support may be molded or machined as an integral part of the seal or as a separate piece. A

plastic seal for cryogenic service should be installed in the housing so that shrinkage at low temperature does not reduce seal-to-shaft bearing pressure.

Use a shaft-seal tester to verify the design of prototype seals. Test at design limits to ensure that leakage requirements will be met under all conditions of shaft deflection, pressure, temperature, and life. If necessary, investigate by test the effect of lip length and thickness on seal performance.

Maximum surface roughnesses of 32  $\mu$  in. (0.81  $\mu$ m) AA on seals and 16  $\mu$  in. (0.40  $\mu$ m) AA on shafts is recommended. Monitor fabrication processes or control them, if possible, to avoid processes that produce undesirable surface textures. Avoid grinding ductile metals with a hard wheel at high pressures. Monitor lapping processes to minimize lapping compound embedment in a sealing surface. A carefully controlled lapping or polishing process that will produce a scratch-free circular lay surface whose texture is composed of rounded, smooth asperities is recommended for minimum leakage and long life. If hard anodized surfaces are used, it is recommended that the anodized surface be ground to obtain the desired surface finish.

Provide the lip seal with an adequate flange sealing area and a loading device such as serrations or raised lands to prevent leakage between seal and housing. Do not permit excessive cold flow. Consider the variation in installed load and variations due to pressure and temperature. If a valve assembly with an aluminum housing is to be operated in cryogenic service, consider the use of a CRES washer to compensate for the thermal shrinkage of the plastic lip seal flange.

### 3.11.3 Face Seals

*The face seal shall control leakage as required and attain the specified cycle life.*

Analyze static and dynamic loads under all operating conditions including installation variables. Obtain the PV factor for the material combination from the seal material supplier and verify that the materials will satisfy cycle life requirements. Analyze seal alignment; ensure acceptable seal loading, due account being given to the effect on alignment of installation variables and deflection under structural and pressure loads and temperature change.

Determine seal/shaft interface roughness in accordance with the analytical procedure outlined by reference 8. It is recommended that a surface somewhat finer than the roughness requirement resulting from this analysis be specified so that some degradation during cycling can be tolerated. Lapped surfaces 4  $\mu$  in. (0.10  $\mu$ m) AA or better are recommended.



## 3.11.4 Bellows

### 3.11.4.1 OPERATING CHARACTERISTICS

*The operating characteristics of the bellows shall result in reliable operation over the specified cycle life.*

Perform a stress analysis to ensure that flexural and pressure stresses for static and dynamic operation are within the allowable operating stresses for the required life. Use the analytical techniques in references 63 and 64 to ensure an adequate design. Encapsulate a representative sample bellows in plastic and then section the capsule to determine the actual bend radii and wall thickness. Use these values in the stress analysis.

Select single-ply or multi-ply construction on the basis of advantages and disadvantages previously noted. Use a single-ply welded or formed bellows in applications where differential pressures are low. Consider the use of the welded bellows to keep the spring rate low when maximum flexibility is a requirement. For reactive propellants, use a single-ply formed bellows, since contamination in convolutions of a welded bellows or between plies of a multi-ply bellows could induce failure. For applications where the spring rate must be kept low, long cycle life is required, and resistance to high differential pressure must be obtained without sacrificing flexibility, the multi-ply formed bellows is recommended.

Provide adequate venting for cryogenic installation to prevent the possibility of over-pressurization during venting or boil-off.

### 3.11.4.2 FABRICATION

*The method of fabrication shall provide acceptable bellows at lowest possible cost.*

Perform a tradeoff study to evaluate the advantages and disadvantages of hydroformed, welded, machined, or metal-deposited bellows. In general, hydroformed bellows are recommended. Consider welded bellows for low-cycle applications where their integrity can be predicted by a simple functional test. When welding of end fittings is undesirable, consider a machined bellows. When small size or low pressure permits a thin wall, consider a bellows formed by metal deposition.

## 3.11.5 Pressure-Balancing-Piston Seal

*The pressure-balancing-piston seal (sleeve seal) shall withstand the maximum applied pressure and shall limit the leakage as required over the range of operating conditions.*

Use an O-ring type of seal in hydrocarbon- or storable-fuel service. Lip seals are recommended only for small sleeve seals (<1.5 in. [3.8 cm] diameter) in cryogenic service. For large valves (>1.5 in. [3.8 cm]) in cryogenic service, use a U-shaped Teflon seal with an Invar expander ring and an aluminum compressor ring, as shown in figure 30. Restrain the leg of the seal that contacts the sliding surface so that during valve motion or exposure to high differential pressures relative displacement of the legs of the seal in the axial direction is restricted to 0.10 in. (2.54 mm) maximum.

## 3.12 MATERIALS

*The materials for valve components shall*

- (1) have an acceptable blend of strength and ductility*
- (2) be compatible with the service fluid*
- (3) be corrosion resistant and wear resistant*
- (4) possess acceptable hardness, weight, thermal expansion properties, and dimensional stability*
- (5) be low in cost.*

Perform a tradeoff study to evaluate the advantages of various materials. Use the information presented in references 41, 42, 65, and 66. Passivate CRES alloys and anodize aluminum alloys to give them corrosion protection. For CRES alloys, use the guide lines presented in reference 75. Specific material recommendations for specific valve design requirements are presented in sections that follow.

### 3.12.1 Poppet Valving Unit

#### 3.12.1.1 HARD-ON-HARD UNIT

For flat and conical poppets and seats in pneumatic and hydraulic service, in liquid oxygen, and in hydrocarbon and hydrazine-type fuels, use 440C steel hardened to Rockwell C 58 to 62. For a spherical seat in any of these fluids, use tungsten carbide or sapphire balls with a 440C seat hardened to Rockwell C 58 to 62. For a spherical poppet and seat in reaction control systems using  $N_2O_4$ , where the  $N_2O_4$  can be kept within the maximum specified water content limit of 0.1 percent during system operation, use a ball fabricated of Stellite No. 3 and a seat of 440C (Rockwell C 58 to 62). For service in operational  $N_2O_4$  systems that are subject to contamination by atmospheric moisture, however, use a Stellite ball seating on a seat of a slightly softer Stellite, or a flat poppet of Stellite No. 3 with a flat seat of Stellite No. 3. Recommendations for materials for service with other propellants are presented in references 41 and 42.

For a flexible metal disk valve used in pneumatic service down to  $-300^{\circ}\text{F}$  (89 K), use 440C heat treated to Rockwell C 58 to 62 for the seats. For the disk, use Elgiloy as rolled, cold reduced 85 percent, and heat treated for 5 hours at  $950^{\circ}\text{F}$  (783 K) in an argon atmosphere. Test the hardness of sample parts to ensure correct heat treatment.

### **3.12.1.2 HARD-ON-SOFT UNIT**

For poppet materials in hydrocarbon fuel or pneumatic service, use either aluminum alloys that are not subject to stress corrosion (e.g., 6061-T6 and 7075-T73) or corrosion-resistant alloys such as A286, 304L, and vacuum-melt 347. Inconel 718 also is recommended for use, except in pure gaseous hydrogen at pressures above 5000 psi (34 MN/m<sup>2</sup>) at room temperature or above. For poppet materials in service in storable propellants, use 6061-T6 and Tens-50 aluminum castings if impact stresses are low; use Stellite, 17-4PH, and 17-7PH when the stress level during storage is very low. Use Inconel 718, A286, or 6061-T6 for poppet materials in cryogenic service.

## **3.12.2 Butterfly Disk Valving Element**

Use Inconel 718 or A286 for a disk in cryogenic service. For a disk in hydrocarbon fuel service, use 7075-T73 aluminum for minimum weight or Inconel 718 or A286 for simplified procurement and logistics (e.g., where a steel disk is required also for the oxidizer valve). In hot-gas service, use Rene 41.

## **3.12.3 Ball Valving Element**

For cryogenic service, fabricate the ball of high-strength alloys such as Inconel 718 and A286. For hydrocarbon fuel service, use Inconel 718, A286, or heat-treated 17-4PH or 17-7PH. Recommended materials for storable-propellant service are the high-nickel alloys, 300-series steel, and heat-treated 17-4PH and 17-7PH steels. Apply dense chrome plating 0.0001 to 0.0003 in. (2.54 to 7.62  $\mu\text{m}$ ) thick on the ball to increase wear resistance.

## **3.12.4 Blade Valving Element**

Recommended materials and practices for ball valving elements are applicable also to blade valving elements.

### **3.12.5 Sleeve Valving Element**

With a Kel-F lip seal, use an Inconel X-750 sleeve with a hard flame-sprayed coating. With a Teflon sleeve seal, a 6061-T6 sleeve that has been chromic-acid anodized is recommended.

### **3.12.6 Cylindrical Slide Valving Unit**

Use 440C hardened to Rockwell C 58 to 62 for the spool and its mating part. Consider the use of a dry-film lubricant if the fluid is other than hydraulic fluid.

### **3.12.7 Elastomeric or Plastic Seal for Valving Unit**

Use Buna-N for O-rings and other molded seals for hydraulic oil and hydrocarbon fuels in the temperature range  $-65^{\circ}$  to  $+150^{\circ}$ F (219 to 339 K). Consult material suppliers and military specifications for a recommended compound for a specific service. For fluids not compatible with Buna-N or for extended-service temperature, consult material suppliers for recommended materials. Consider silicone rubber or Teflon for low-temperature service, and silicone rubber or Viton A for high-temperature service. Use resin-cured butyl rubber for short-term (<15 min.) exposure to storable oxidizers. Use ethylene propylene rubber or resin-cured butyl rubber in hydrazine-type fuels.

Use Kel-F, Mylar, and Kynar as soft-seal seat materials for hydrocarbon-fuel and pneumatic service. Use Teflon when bonding the soft seal to the metal poppet is required. In general, use Teflon for soft seals in storable-propellant service; Kel-F is recommended for use only where the duration of exposure is limited to a maximum of 2 hours. Use Kel-F, Vespel, and Teflon seats for service in liquid oxygen, liquid hydrogen, and low-temperature gas; Kel-F is generally preferred. Use Kel-F for cold-formed lip seals intended for liquid-oxygen service. Use Mylar seats and lip seals only for liquid-hydrogen service and low-temperature gaseous nitrogen and gaseous helium service; do not use Mylar in liquid oxygen.

With a butterfly disk valving unit, use a Kel-F disk seal in cryogenic service and polypropylene in hydrazine service. Use Teflon as the seat material in a ball valving unit in storable-propellant service or where low friction is important; otherwise, use Kel-F.

### **3.12.8 Housing**

The housing material should be suitable for fabrication into complex shapes by conventional techniques. Make a tradeoff study of the advantages and disadvantages of various materials

and fabrication processes as extrusions, castings, and forgings. Use die forgings for large, complex shapes. Recommended aluminum alloys are 6061-T6 extrusions, Tens-50 and 356-T6 castings, and 7075-T73 extrusions and forgings. Chromic-acid anodize these materials for corrosion protection. Aluminum alloys 2024-T6 and 7075-T6 are not recommended. Recommended steels are 302 or 347 castings or 321 or 347 consumable-electrode-melted bar stock; 304L bar stock is recommended if a lower yield strength can be tolerated.

Join a steel bellows to an aluminum housing by brazing a 304L steel sleeve into a 6061-T651 aluminum alloy housing and then weld the bellows to the sleeve. Do not exceed a bellows or housing temperature of 300°F (422 K) during the welding operation. Avoid joining these dissimilar metals unless operation in a dry environment can be ensured; otherwise, corrosion effects can be serious.

### **3.12.9 Shaft**

Use a high-strength nickel alloy such as Inconel 718 or A286 steel for cryogenic service in a fast-actuating valve where ductility and corrosion resistance are required. For a slow-actuating valve where the impact is low and high ductility is not required, and where the design does not induce areas of high stress concentration and corrosion is minimal, use a heat-treatable 400-series steel. For hot-gas service, use a high-temperature alloy such as Rene 41; this alloy also provides good service in corrosive atmospheres. For service in corrosive fluids (e.g., many high-performance storable propellants), use a high-nickel alloy such as Inconel 718. If the shaft is not subjected to sustained stresses likely to induce stress corrosion, 17-4PH or 17-7PH alloys are acceptable. A 300-series steel is recommended for a design that can utilize the high-strength properties of cold working. For ordinary service temperatures where corrosion is not a problem (e.g., in a hydraulic metering valve), Nitralloy or 52100 bearing alloy is recommended. Where there is a minor corrosion problem, use 440C steel. Areas requiring additional corrosion protection, such as the end of a shaft that is exposed to atmospheric moisture, may be plated with chrome or nickel.

Shafts used in cryogenic service and operating with near-zero clearance must be processed to stabilize grain growth. This stabilization should be accomplished by chilling the shaft to liquid-nitrogen temperature after heat treatment (ref. 75).

### **3.12.10 Bearings**

For guides and journal bearings in normal or cryogenic service, use Teflon or a filled Teflon or a soft metal plating (e.g., silver or gold) against a hard surface. Dry-film-type lubricants

are not recommended because of the contamination generated by wear of the dry film. For extremely corrosive service such as in liquid fluorine, beryllium-copper against flame-plated tungsten carbide is recommended. Use 440C steel for rolling-contact bearings in normal or cryogenic service. For guides and journal bearings in high-temperature service, use a high-temperature alloy such as Haynes 25 against a hard surface such as flame-plated tungsten carbide or aluminum oxide. For high-temperature rolling-contact bearings, use Haynes 25.

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## GLOSSARY

<u>Term or Symbol</u>	<u>Definition</u>
AA	arithmetical average deviation of a surface from a mean line
actuator	device that converts control energy into mechanical motion to operate a valving element
ambient temperature	the temperature of the environment surrounding a body
anodize	form a protective oxide coating on a metal by electrochemical means
balance chamber or piston	chamber used to balance the pressure forces exerted on a poppet valving element and thereby decrease force necessary to actuate the element
ball flow tube	tube inserted into a hollow ball to reduce pressure drop through a ball valve
ball valve	rotary-action valve using a ball with a flow passage that rotates to align the mating upstream and downstream lines
bearing pressure	applied load divided by projected area of a bearing
Belleville spring	truncated, conical, metal spring washer that can provide a negative mechanical spring rate
bellows	thin-walled, circumferentially corrugated cylinder that can be elongated or compressed longitudinally
biasing spring	spring installed in a valve actuator to obtain a preferred position when power is removed
bipropellant valve	valve incorporating both fuel and oxidizer valving units driven by a common actuator
bleed	remove or draw off fluid from a system
circular lay	circular direction of the predominant pattern of a machined surface; ordinarily determined by the production method used
coupling	mechanism used to attach a rotating shaft to its valving element; e.g., splines or drive flats

<u>Term or Symbol</u>	<u>Definition</u>
creep	gradual flow of a material, either metallic or nonmetallic, under application of constant load, even when stresses are under the proportional limit
cryogenic	fluids or conditions at low temperatures, usually at or below $-150^{\circ}\text{C}$ (123 K)
cycle life	the number of times a valving unit may be opened and closed and still possess sufficiently low internal leakage to meet specified leakage rate
cyclic vibration	vibration mode, induced by rough combustion in a rocket engine, that causes severe g loads on a periodic basis at one predominant frequency
cylindrical slide or piston valve	valve utilizing a cylindrical bore with annular holes that are covered or uncovered by a piston sliding through the bore; used primarily as a sequence valve
decontamination	cleaning process to ensure that all corrosive fluids remaining in a valve assembly after test are flushed out or neutralized, so that damage to metallic and nonmetallic materials does not occur
disk	valving element of a butterfly valve
dry cycle mode	functional testing of the valve without propellant or test fluid in the flow passages
dynamic seal	mechanical device used to minimize leakage of a fluid from the flow-stream region of a valving unit when there is relative motion between the sealing interfaces
earth-storable propellant	propellant with a vapor pressure such that the propellant can be stored on earth as a liquid in a moderate-pressure container
elastomer	polymeric material that at room temperature can be stretched to approximately twice its original length and on release return immediately to its original length
epoxy	thermosetting resin widely utilized in the fabrication of glass-filament/resin composites and as an adhesive
fail safe	philosophy in the design of propulsion system valves and associated hardware that seeks to avoid the compounding of failures; fail-safe design provisions ensure that the valve element will move to a predetermined "SAFE" position if electrical, pneumatic, or hydraulic power is lost

<u>Term or Symbol</u>	<u>Definition</u>
flat-face armature	solenoid actuator utilizing a flat armature that is flexure mounted to avoid sliding fits in the valve assembly
flexure disk	supporting member that allows poppet axial motion but restrains rotation and prevents misalignment of concentric fits
flexure tube	interconnecting member between the valving element and a dc torque motor that transmits motion and seals the flow stream; the tube is rigidly attached to the valve body at one end and acts as a spring
four-way valve	valve having four controlled working passages such that there are two simultaneous flow paths through the valve; commonly used to control a double-acting actuator
fretting	mechanism of wear that acts on mated metallic materials to produce surface damage when one surface moves relative to the other; vibration and cryogenic temperatures are aggravating factors
full-complement bearing	needle or roller bearing that does not incorporate an inner race; the needles or rollers ride directly on the shaft
galling	progressive damage of mating surfaces resulting in increased friction and possible seizure
gland	cavity in which an O-ring is installed; includes the groove and cooperating surface of the mated part that together confine the O-ring
hard poppet	poppet with a sealing surface of metal, ceramic, or a cermet
hard seat	seat with a sealing surface of metal, ceramic, or a cermet
Hertz stress	maximum compressive stress caused by the pressure between elastic bodies, at least one of which is a curved body
hydraulic	operated, moved, or effected by liquid used to transmit energy
hydraulic dashpot	device used to reduce the velocity of the actuator as it approaches a fixed stop, so that impact energy levels are reduced
journal bearing	sliding-surface bearing in which combinations of metals and nonmetals are used to achieve low friction, fluid compatibility, and wear resistance
land	actual sealing surface in poppet valve housing
L/D	length-to-diameter ratio

<u>Term or Symbol</u>	<u>Definition</u>
lever link	mechanical linkage between the actuator and the valving element of a rotary valve that consists of a lever or crank on the rotary member and a link with clevis connections from the lever to the actuator shaft
master lap	lapping of the work piece with a master tool
match lap	lapping together of two mating detail parts so that they are matched in contour
metal-to-metal seal	an internal seal in a poppet type valve achieved with hard-on-hard or hard-on-soft metallic seats
OAMS	orbital attitude and maneuvering system
overtravel tolerance	feature provided by rotary valves such as the ball or blade whereby shutoff can be achieved even when the valving element is not rotated to exactly the same closure position each time
pintle valve	flow-control unit utilizing a translating pointed shaft to change flow area of an orifice or flow passage
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
plunger solenoid	solenoid that pulls an armature into the center of a coil when the coil is energized
pneumatic	operated, moved, or effected by gas used to transmit energy
poppet valve	valve constructed to close off flow by translating a ball, cone, or disk against a seat in the housing; translation of the poppet away from the seat can result in essentially orifice flow
primary leakage	leakage from the upstream side to the downstream side of a valving unit
primary seal	seal intended to limit primary leakage
proof test	pressure test to prove the structural integrity of a valve assembly without exceeding allowable stresses
PV	product of bearing pressure $P$ times rubbing or sliding velocity $V$
random vibration	vibration characterized by a wide continuous band of multiple frequencies

<u>Term or Symbol</u>	<u>Definition</u>
RCS	reaction control system
redundant	incorporating duplicate, identical components to achieve increased reliability
repeatability	capability of a valving unit and actuator to operate in the same way and in the same time each time the assembly is actuated
response time	the length of time from first signal to full-open or full-closed valve position, a total comprised of electrical delay plus pneumatic or hydraulic control system delays plus valve travel time
rms	root mean square
Rockwell C	hardness scale
rolling element	ball, needle, or tapered roller in a rolling-contact bearing
scch	standard cubic centimeters per hour
scim	standard cubic inches per minute
seat	surface in the valve housing that the valving element contacts to shut off flow and limit primary leakage
secondary leakage	leakage from the valve interior to the exterior
secondary seal	seal intended to limit secondary leakage, e.g., a shaft seal
self-aligning bearing	journal bearing with a spherical joint to provide alignment of the axis
shutoff valve	valve that terminates the flow of fluid; usually a two-way valve that is either fully open or fully closed
sleeve valve (linear)	valve utilizing a cylindrical sleeve element that reciprocates in the cylinder bore to open or close the flow area by uncovering or covering annular slots in the bore
sleeve valve (rotary)	valve utilizing concentrically mated slotted cylinders that open and close the flow area by rotation of one cylinder relative to the other
soft poppet	poppet that has a sealing surface of elastomer or plastic
soft seat	seat that is fabricated of an elastomer or plastic



<u>Term or Symbol</u>	<u>Definition</u>
space-storable propellant	propellant with a vapor pressure such that the propellant can be stored in the space environment at moderate ullage pressure without significant loss over the mission duration
spool valve	valve utilizing a solid cylindrical valving element having two or more lands that fit closely within the bore of the housing; the valve opens or closes by translating the spool within the bore
spring rate	change in spring force per unit of deflection of a flexure element
squeeze (O-ring)	compression of the O-ring cross section between opposite surfaces of a gland
static seal	device used to prevent leakage of fluid through a mechanical joint in which there is no relative motion of the mating surfaces other than that induced by changes in the operating environment
sterilization	process in which a propulsion system package is rendered sterile or free from micro-organisms and bacteria by the application of heat or by the use of a special sterilization fluid or both
three-way valve	valve having three controlled ports, usually one inlet and two outlet ports
throttle valve	valve to control flow rate of a fluid by means of a variable-area flow restriction; this kind of valve may have an infinite number of operating positions as contrasted to a shutoff valve that is either fully open or fully closed
TIR	total indicated runout
valving element	the moving portion of the valving unit that translates or rotates to vary or to shut off the flow of fluid
valving unit	the combination of the movable valving element and the valve seat contained in a suitable housing
valving unit throat	flow area between the valving element and seat of the valving unit
visor valve	ball valve constructed with only a segmented shell of the ball; design is used to reduce weight
wet cycle mode	a functional test in which the valve is operated with propellant or test fluid in the flow passage

<u>Material</u>	<u>Identification</u>
<u>Metals<sup>1</sup> and other solids</u>	
A286	austenitic iron-base alloy
AM-350, -355	semi-austenitic precipitation hardening steels
Buna-N	tradename for copolymer of butadiene and acrylonitrile
cermet	material composed of metals and ceramics, e.g., tungsten carbide with a cobalt binder
CRES	corrosion resistant steel
Elgiloy	cobalt-base spring alloy manufactured by Elgin National Watch Co.
EPR	ethylene propylene rubber
Havar	cobalt-base spring alloy
Haynes 25	tradename of Haynes Stellite Corp. for a cobalt-chromium-nickel alloy
Haynes Stellite	see Stellite
Inconel 718, X-750	tradenames of International Nickel Co. for austenitic nickel-base alloys
Invar	tradename of International Nickel Co. for a nickel-base alloy with a very low coefficient of thermal expansion
Kel-F	tradename of 3 M Corp. for a high molecular weight polymer of chlorotrifluoroethylene
Kynar	tradename of Pennsalt Chemical Corp. for polyvinylidene fluoride
Meehanite	tradename of Meehanite Metal Corp. for cast iron
Mylar	tradename of E. I. duPont, Inc. for polyethylene terephthalate film
Nitralloy	steel alloyed with nitride-forming elements; exposure of the alloy to active nitrogen results in a thin hard case that is especially wear resistant
nitroso rubber	1:1 copolymer of trifluoronitrosomethane and tetrafluoroethylene
nylon	a polyamide

<sup>1</sup> Additional information on metals herein can be found in references 52, 65, and 66.

<u>Material</u>	<u>Identification</u>
Rene 41	tradename of General Electric Co. for austenitic nickel-base alloy
Stellite	designation for a series of cobalt-tungsten-chromium-carbon alloys manufactured by Haynes Stellite Corp.
Teflon (also Teflon TFE)	trademark of E. I. duPont, Inc. for tetrafluoroethylene polymer
Teflon FEP	trademark of E. I. duPont, Inc. for fluorinated ethylene propylene polymer
Tens-50	tradename of Rockwell International for high-strength cast aluminum alloy
T4, T6, T73, T351, T651	designations for heat-treating and tempering processes for aluminum alloys
Vespel	trademark of E. I. duPont, Inc. for a polyimide film
Viton A	trademark of E. I. duPont, Inc. for copolymer of vinylidene fluoride and hexafluoropropylene
17-4PH, -7PH	semi-austenitic precipitation-hardening stainless steels
302	austenitic stainless steels
304L	
321	
347	
356	cast aluminum alloy with Mg and Si as principal alloying elements
430	ferritic stainless steel
431	martensitic stainless steel
440C	martensitic stainless steel
446	ferritic stainless steel
1100	wrought aluminum (99% Al)
2024	wrought aluminum alloy with copper as the principal alloying element.
4340	low-alloy standard steel

<u>Material</u>	<u>Identification</u>
6061 6151	wrought Al alloys with Mg and Si as principal alloying elements
7075	wrought Al alloy with Zn as principal alloying element
52100	electric-furnace steel
<u>Fluids</u>	<u>Identification</u>
Aerozine 50; A-50	mixture of 50% hydrazine and 50% unsymmetrical dimethylhydrazine, propellant grade per MIL-P-27402
FLOX	mixture of $LF_2$ and LOX
fluorine	elemental fluorine ( $F_2$ ) in its liquid form ( $LF_2$ ) used as a cryogenic propellant
Freon MF	trademark of E. I. duPont, Inc. for trichloromonofluoroethane
Freon TF	trademark of E. I. duPont, Inc. for trichlorotrifluoroethane
$GH_2$	gaseous hydrogen
$GN_2$	gaseous nitrogen
helium, He	pressurant helium per MIL-P-27407
hydrazine	$N_2H_4$ , propellant grade per MIL-P-26536
IRFNA	inhibited red fuming nitric acid per MIL-P-7254
$LH_2$	liquid hydrogen, propellant grade per MIL-P-27201A
$LO_2$ ; LOX	liquid oxygen, propellant grade per MIL-P-25508D
MMH	monomethylhydrazine, propellant grade per MIL-P-27404
$N_2O_4$	nitrogen tetroxide, propellant grade per MIL-P-26539
nitrogen	gaseous nitrogen per MIL-P-27401A
RP-1	kerosene-base hydrocarbon fuel, propellant grade per MIL-P-25576
UDMH	unsymmetrical dimethylhydrazine, propellant grade per MIL-P-25604D

<u>Vehicle, Propulsion System, or Engine Designation</u>	<u>Identification</u>
Atlas	launch vehicle using MA-5 engine system
Centaur	upper stage on Atlas or Titan; uses RL10 engine system
F-1	engine for S-IC; 1 500 000 lbf (6.67 MN) thrust; uses RP-1/LOX; manufactured by Rocketdyne
Gemini	manned spacecraft for extended earth orbital missions
H-1	engine for S-IB; 200 000 lbf (890 kN) thrust; uses RP-1/LOX; manufactured by Rocketdyne
Intelsat III thruster	engine for Intelsat III; 3.5 lbf (15.6 N) thrust; uses N <sub>2</sub> H <sub>4</sub> as monopropellant; manufactured by TRW Systems
J-2	engine for S-II; 200 000 lbf (890 kN) thrust; uses; LH <sub>2</sub> /LOX; manufactured by Rocketdyne
Lance	intermediate-range tactical missile manufactured by Ling-Temco-Vought; uses UDMH/IRFNA
LEM	Lunar Excursion Module
LMAE	Lunar Module Ascent Engine; 3500 lbf (15.6 kN) thrust; uses A-50/N <sub>2</sub> O <sub>4</sub> ; manufactured by Rocketdyne
LMDE	Lunar Module Descent Engine; 9850 lbf (43.81 kN) thrust; uses A-50/N <sub>2</sub> O <sub>4</sub> ; manufactured by TRW Systems
LR-87-AJ-5	engine for Titan 1 <sup>st</sup> stage; 215 000 lbf (956 kN) thrust; uses A-50/N <sub>2</sub> O <sub>4</sub> ; manufactured by Aerojet Liquid Rocket Co.
LR-91-AJ-5	engine for Titan 2 <sup>nd</sup> stage; 100 000 lbf (445 kN) thrust; uses A-50/N <sub>2</sub> O <sub>4</sub> ; manufactured by Aerojet Liquid Rocket Co.
M-1	engine designed and developed by Aerojet-General but not flightproven; 1 500 000 lbf (6.67 MN) thrust; used LH <sub>2</sub> /LOX
MA-5	five-engine system for Atlas containing 2 booster, 2 vernier, and 1 sustainer engines; boosters provide 330 000 to 370 000 lbf (1468 to 1646 kN) thrust; uses RP-1/LOX manufactured by Rocketdyne
M-50-M	engine for MM69; 50 lbf (222 N) thrust; used N <sub>2</sub> H <sub>4</sub> (monopropellant); manufactured by TRW, Inc.

<u>Vehicle, Propulsion System, or Engine Designation</u>	<u>Identification</u>
RL10	engine for Centaur; 15 000 lbf (66.7 kN) thrust; uses LH <sub>2</sub> /LOX; manufactured by Pratt & Whitney Aircraft Division of United Aircraft
RS-21	engine for MM71; 300 lbf (1.3 kN) thrust; uses MMH/N <sub>2</sub> O <sub>4</sub> ; manufactured by Rocketdyne
S-1B	booster using a cluster of eight H-1 engines
S-1C	first stage (booster) of the Apollo Saturn V vehicle; uses five F-1 engines
S-II	second stage of the Apollo Saturn V vehicle; uses a cluster of five J-2 engines
SE-5	engine for attitude control; 50 lbf (222 N) thrust; uses MMH/N <sub>2</sub> O <sub>4</sub> ; manufactured by Rocketdyne
SE-6	engine for Gemini RCS; 25 lbf (111 N) thrust; uses MMH/N <sub>2</sub> O <sub>4</sub> ; manufactured by Rocketdyne
SE-7	engine for Gemini OAMS; 85 to 100 lbf (378 to 445 N) thrust; uses MMH/N <sub>2</sub> O <sub>4</sub> ; manufactured by Rocketdyne
SE-8	engine for Apollo Command Module RCS; 93 lbf (414 N) thrust; uses MMH/N <sub>2</sub> O <sub>4</sub> ; manufactured by Rocketdyne
SE-9	engines on Titan Transtage; 25 and 45 lbf (111 and 200 N) thrust; uses A-50/N <sub>2</sub> O <sub>4</sub> ; manufactured by Rocketdyne
Service Module Propulsion System	AJ10-137 engine for Apollo Service Module; 22 000 lbf (98 kN) thrust; uses A-50/N <sub>2</sub> O <sub>4</sub> ; manufactured by Aerojet
Titan I, II, and III	family of launch vehicles using the LR-87-AJ and LR-91-AJ series of rocket engines developed by Aerojet-General Corporation

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