

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

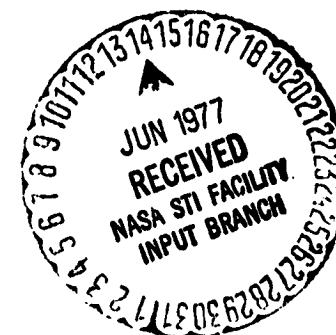
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**LIQUID ROCKET DISCONNECTS, COUPLINGS,  
FITTINGS, FIXED JOINTS, AND SEALS**



**SEPTEMBER 1976**

**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 Disconnects, Couplings, Fittings, Fixed Joints, and Seals," 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 D. E. Stuck, Rocketdyne Division, Rockwell International and was edited by Russell B. Keller, Jr. of Lewis. Significant contributions to the text were made by Mel Burr, Rocketdyne Division, Rockwell International Corporation. 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, Henry Hillbrath of The Boeing Company; R. M. Huls, Pratt & Whitney Aircraft Group, United Technologies Corporation; Charles Schroeder, McDonnell-Douglas Astronautics Company; Clifford Thompson, Aerojet Liquid Rocket Company; Louis Toth, Jet Propulsion Laboratory, California Institute of Technology; Philip Muller, Marshall Space Flight Center; and E. J. Fourney, Lewis Research Center 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.

September 1976

## 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 DISCONNECTS, COUPLINGS, FITTINGS, FIXED JOINTS, AND SEALS

## 1. INTRODUCTION

Disconnects, couplings, fittings, fixed joints, and seals are used in liquid-propellant rocket propulsion systems to contain and control the flow of the fluids involved. This monograph treats the design of these components for use in booster, upper-stage, and spacecraft propulsion systems; ground support equipment is excluded. Particular emphasis is placed on the design of components used on large-engine systems because of the high pressure and high vibration levels to which these components are exposed.

Disconnects are fluid-system connectors that provide quick-action separation of fluid-system interfaces between flight vehicles and ground systems or between stages of the flight vehicle. Couplings provide mechanical connection of elements of a fluid system and are capable of mechanical disassembly but not the quick-action separation characteristic of disconnects. Fittings are elements of a duct or duct assembly that change flow area or direction and provide control of internal flow geometry; they may also provide instrumentation access to the fluid, and attachment joints for auxiliary equipment. Fixed joints connect elements of a system when disassembly, removal, or replacement is not required; these joints are not capable of simple mechanical disassembly as are disconnects and couplings and are fabricated by methods such as welding or brazing. Seals within a disconnect or coupling are elements that conform to the retaining connector and thereby limit or control fluid leakage; seals are also utilized within components to limit or control leakage while permitting relative movement between two elements. Dynamic seals that permit both turning and sliding are covered herein; seals for high-speed rotating shafts are treated in reference 1.

The successful design of the components defined above presents a formidable array of technical problems to the designer; for example, temperature extremes from  $-423^{\circ}\text{F}^*$  (excluding thrust-chamber combustion products), pressure extremes from zero to 10 000 psi\*, high vibration and shock load environments, corrosive media, and the requirement that each part be within engineering tolerances and free of damage and contamination (ref. 2). This monograph draws on the design experience accumulated in development and operational programs in the last 20 years to present guidelines for overcoming these and other problems in achieving successful design.

\*Factors for converting U.S. customary units to the International System of Units (SI units) are given in Appendix A. Terms, symbols, and materials are defined or identified in Appendix B.

The problem most prevalent in valve-type disconnects is leakage, which may be caused by misalignment of components, extreme environment, extreme fluid requirement, or mechanical loads. Sealing elements within the valve as well as at the interfaces are exposed to some extent to ambient environmental conditions following separation. As a result, the sealing elements are especially vulnerable to damage caused by foreign particles and corrosion. Leakage also can be a result of self-induced damage resulting from inappropriate materials or to chatter or oscillation of the valve element (not mechanically held in fully open position) during prolonged exposure to flow. Excessive mechanical loads can cause malfunction of locking mechanisms or improper separation of the two halves. These excessive loads can be caused by handling damage, contamination, corrosion, mismatched parts, misalignment, improper installation, or inadequate design that did not give proper recognition to all operating parameters including dynamic conditions during separation.

The primary problem with couplings and seals also involves leakage, which may arise from structural deflection, differential thermal expansion, inadequate control of surface finish and dimensions, or handling damage. Structural problems include underdesign of the restraining load required to maintain a seal, flange deflection beyond the capability of the seal, and warpage of flanges in high-temperature applications. Seal problems sometimes are related to the interface coatings. Cryogenic seals with a Teflon coating can be subject to excessive cold flow if the bearing pressure is too great or if the processing controls are not maintained; some high-temperature seals have soft platings that may blister and stick to the mating interface if the plating is not processed properly. Both the coupling/seal interface and the seal are prone to handling damage because the critical surfaces normally are exposed when the coupling is not assembled. Threaded flared-tube couplings are plagued with leakage problems because of damage to metal-to-metal sealing surfaces and relaxation of the connecting-nut torque.

Problems with fittings that lead to leakage or inadequate functioning have been due primarily to initial design inadequacies such as improper flow levels and flow distribution, excessive pressure loss, and undesirable thermal gradients. Problems with fixed joints include leaks, corrosion due to improper cleaning and purging, poor dimensional control of parts to be joined, excessive weld droptthrough that restricts flow, and structural failures due to inadequate support or higher-than-predicted vibrations.

Since the primary problems related to the components discussed in the monograph are concerned with design details, emphasis is placed on experience with particular detail problems and on successful problem solutions. The monograph treats component design rather than the procedure or process by which a part is made. Each component is discussed separately in the following sequence: disconnects, couplings, fittings, fixed joints, and seals. Configuration design of the component is presented first, followed by discussion of the various design elements and their impact on the design. The monograph emphasizes hardware detail and is intended to serve as a summary of successful design as well as a useful tool for reviewing and evaluating new designs of the subject components.

## 2. STATE OF THE ART

Rocket engine and space vehicle programs have utilized disconnects, couplings, fittings, fixed joints, and seals to varying degrees. The current configurations for many of these components have resulted from solutions to operational problems with earlier designs and from the need to satisfy new and more stringent performance requirements. In general, the components have become more sophisticated to meet the ever-increasing requirements for performance and reliability.

Current aerospace technology utilizes only manually operated and breakaway disconnects; forced-separation disconnects, used in the past, do not appear in current engine and vehicle systems. Manually operated disconnects are predominant and have the greatest variety of applications. The manually operated disconnects used in the early programs such as Redstone were aircraft or industrial types, with essentially no modifications. Requirements for breakaway and forced-separation types were more severe than those for manual disconnects, and there were no readily available units of either type. New designs, therefore, were required to meet aerospace needs. As vehicle design and launching technology matured, the forced-separation disconnects have been replaced by breakaway disconnects, which in most cases are clustered together on umbilical panels. Manually operated disconnects still are basically aircraft and industrial designs, modified as necessary to meet the more exacting requirements of the aerospace industry for cleanliness, reliability, and material compatibility.

In current vehicles, flanged bolted couplings are used in high-pressure systems (10 000 psi) in sizes up to 9 in. in diameter and in low-pressure systems (600 psi) up to 30 in. in diameter; flanged couplings as small as 1/4 in. diameter have been used. Flared-tube and threaded-connector couplings are used primarily in the smaller sizes (< 1½-in. diam.) throughout a vehicle in a variety of applications.

On the more recent large engines and vehicles, much effort has been devoted to elimination of static-seal couplings by the use of in-place welding or brazing of joints. Welded and brazed joints are used extensively in spacecraft systems. The welds generally are butt welds and are accomplished with advanced, automated processes: electron-beam welding and gas-tungsten-arc welding. With the latter process, welds frequently are made in place on an assembly with diminutive welding devices. These butt joints are capable of being fully inspected for weld quality.

The technology of fittings for rockets presently is a rather specialized area. The majority of fittings are designed and fabricated by the vehicle or engine manufacturer or by specialty shops to accomplish a particular function. A wide variety of fittings such as elbows, tees, crosses, tapoffs, area transitions, flow straighteners, and flow-turning devices has been designed and successfully used in cryogenic, hot-gas, vacuum, and high-pressure applications. However, improvements in fitting designs still can be made in the areas of fluid-passage

designs for lower pressure drop, resistance to corrosion, resistance to thermal-stress cracking, and general structural integrity.

Static seals specified for current applications fall into three general groups:

- (1) Elastomeric O-rings or molded-in-place seals for use in limited-temperature-range systems ( $-80^{\circ}$  to  $+500^{\circ}\text{F}$ ) when such seals are compatible with the fluid.
- (2) Plastic pressure-assisted seals for limited-range systems ( $-80^{\circ}$  to  $+500^{\circ}\text{F}$ ) when the seals are compatible with the fluid.
- (3) Metal seals, usually with soft-metal plating or Teflon coating (depending on the environment), for temperature extremes (hot gas or cryogenic).

The elastomeric seal is the simplest type, is the most reliable (when the seal assembly is properly designed), and is inexpensive; it is usually the first choice if the elastomer is compatible with the environment. The plastic pressure-assisted seal is more expensive than the elastomeric O-ring, and covers about the same range of temperature and pressure; the advantage of the plastic pressure-assisted seal is its ability to seal with more joint separation than the O-ring seal. The metal seal is the most common cryogenic and hot-gas seal. The pressure-assisted seal concept was successfully developed and placed in service earlier than the non-pressure-assisted seal and, when integrated properly into a coupling, performed well within the design requirements. The flange interfaces are fairly simple in comparison with those for some of the non-pressure-assisted seals that require close-tolerance special interfaces.

## 2.1 DISCONNECTS

A disconnect assembly, also called a quick disconnect, is a specific type of separable connector characterized by two separable halves, an interface seal, and usually a latch-release locking mechanism; many breakaway or rise-off disconnects that are held together by gravity or friction do not have latch-release mechanisms. A check valve or shutoff valve usually is contained in at least one-half of the assembly. Connection or disconnection normally is performed quickly (in about 1 sec) and is implemented with a simple slide or rotary motion (or both) of the coupling ring. In vehicle-to-ground disconnects, the vehicle half is a flight item and as such requires flight reliability. Failure to disconnect properly can have a serious impact on the launch or even cause an abort.

The extensive use of disconnects in launch vehicles is illustrated by two examples. The Saturn V vehicle contained a total of 273 manual and breakaway disconnects (ref. 3). Approximately two-thirds of these were related to the propulsion systems; the remaining disconnects were associated with fuel cells and life-support systems. In the Atlas (MA-5)

vehicle, eight disconnects are used between the booster-engine package and the vehicle. The disconnects range in size from 1/2 in. to 1 1/2 in.; they are all breakaway types, with poppet-type check valves in the vehicle half. These staging disconnects are operative when the booster-engine package is jettisoned during flight.

### **2.1.1 Configuration Design**

In a disconnect assembly, the valve-operating mechanism, the latch-release mechanism, and the interface seal are intimately related. Therefore, the design of both halves is controlled by a single contractor, even though a subcontractor ultimately may be responsible for procurement of the ground half. Since the flight-half requirements are the more critical, the vehicle contractor normally is responsible for this half and for coordinating the entire design. Poor coordination can cause time-delaying and embarrassing problems when the two halves are mated for the first time (e.g., accumulation of manufacturing tolerances that could prevent engagement or permit seal leakage).

In the design of a disconnect, the very significant differences among all of the operating requirements including environmental conditions, contamination, and sealing in both the connected and the disconnected modes must be recognized. For example, in the engines in the Atlas vehicle, manually-operated disconnects are used to drain residual fuel from the system. The system pressure during this drain operation, the only time the two halves are connected, is static head only and is insignificant. Thus, a minimal latching mechanism could satisfy this requirement. However, during engine operation, when the ground halves are disconnected, the system pressure approaches 1000 psi. The disconnects, therefore, are designed to withstand the operating pressure.

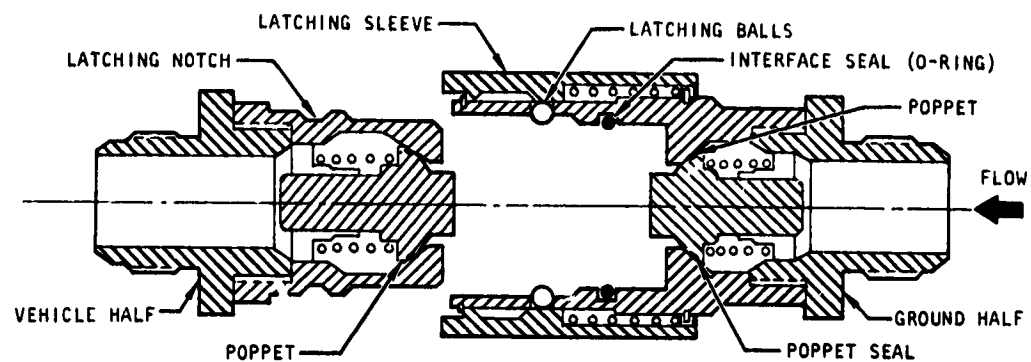
#### **2.1.1.1 MANUALLY OPERATED DISCONNECTS**

Manually operated disconnects are used in both liquid and gas systems to facilitate servicing and checkout operations of the flight systems while the vehicle is on the ground; for example, pressure testing and purging engine systems, and transferring life-support supplies (water, oxygen) and fuel-cell supplies to the vehicle. Manually operated disconnects larger than 1 1/2-in. seldom are used; 1/4-, 3/8-, and 1/2-in. line sizes are the most common.

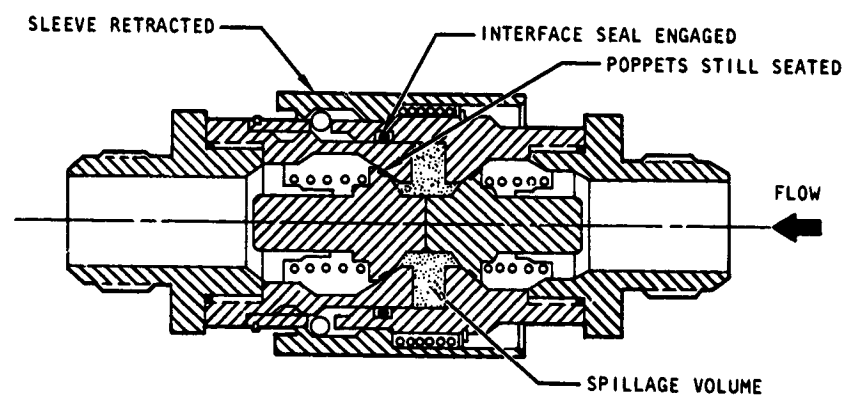
Manually operated disconnects normally contain a check or shutoff valve in each half; the valve is forced open mechanically by the connecting operation. Figure 1 illustrates this action. In some applications (e.g., purging from the ground), a pressure-actuated valve may be used in the vehicle half.

Safety of personnel is a prime consideration in design of manually operated disconnects, particularly with toxic or cryogenic propellants at high operating pressures. The propellants

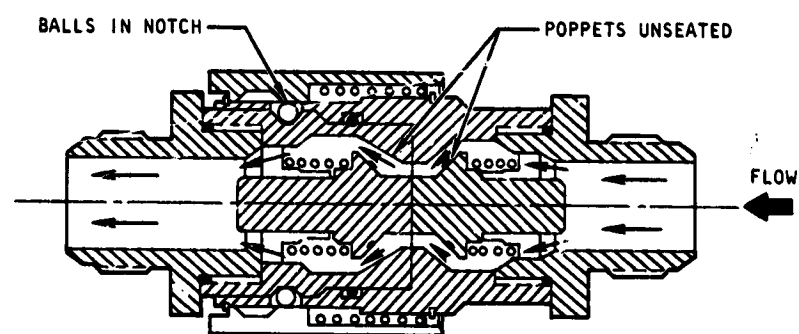




(a) Disconnected



(b) Partially connected (not latched)



(c) Fully connected (latched)

Figure 1. — Three stages of operation of a typical manually operated disconnect.



can cause bodily harm in the form of skin burns or lung-damaging inhalation. The high pressure produces high kinetic energy that, in the event of a structural failure or inadvertent disconnection, can induce high velocity into moving hardware. Therefore, during design careful attention must be given to the conditions under which the disconnect is to be separated (i.e., no pressure, low pressure, or high pressure).

Three basic types of latch-release mechanisms are used: screw threads, push-pull, and twist. While the threaded type offers maximum simplicity and reliability, it is the most cumbersome to use. In some applications, a latching mechanism that requires a specific action (e.g., an initial rotary motion) may be used as a safety feature. With very hazardous or high-pressure fluids, the check valves are not opened during connection until after the interface seal is engaged and the latching mechanism is secured. In a ball-valve type of disconnect, for example, this sequence is imposed by providing an interlock between the ball actuation and the latching mechanisms.

Manually operated disconnects rarely are used in cryogenic applications because of problems associated with handling temperatures, reliable interface seals, and contamination, principally moisture/ice.

Efforts to provide industry standards for manually operated disconnects are presented in reference 4.

#### **2.1.1.2 BREAKAWAY DISCONNECTS**

Breakaway disconnects (also called rise-off or staging disconnects) are disengaged by the separation force as the vehicle rises from the launch pad (or as an upper stage pulls away from a lower stage). Ranging in size up to 11 in. in diameter, these disconnects are used to maintain critical contacts with supply facilities such as propellant and pressurizing gas supply, hydraulic system supply, propellant fill and drain, and purge gas until vehicle liftoff (or stage separation). In the past, these disconnects also were used to supply propellants from engine start systems that were ground-mounted (Jupiter) or jettisoned in flight (Navaho); however, current technology employs self-contained start systems, which do not require breakaway disconnects. In most applications, both halves contain check valves, but it is not uncommon for the check valve to be deleted from one half.

Disconnects for cryogenic systems almost invariably have been the breakaway type. The ground carrier is mated and latched to the vehicle panel manually under ambient conditions, prior to activating the system; thus the danger of trapping frost or condensate with the disconnect is avoided. The complex mechanism normally required to make the connection remotely is unnecessary. Once mated and checked out, breakaway disconnects are not disconnected until vehicle liftoff.

Disengagement takes place after the latch mechanism is released automatically by pneumatic or mechanical actuation. Although latch mechanisms on the panels or the individual disconnects are similar to those in the manually operated disconnects that incorporate the ball or finger-locking concept, many other types have been used. The Centaur propellant fill and drain valves, for example, incorporate bolts that are fractured at rise-off by a lanyard-actuated cam system.

The most common configurations for breakaway disconnects are the slip and the ball and cone (figs. 2 and 3). Either of these types may be pressure balanced (figs. 4 and 5).

The slip disconnect (fig. 2) is characterized by cylindrical male and female halves that slip together and are sealed with self-forming lip seals, O-ring seals, or chevron seals. They are usually sealed with two seals, with the area between the seals vented for leakage measurement and for disposal of acceptable leakage of harmful or combustible gases. Disconnects on cryogenic lines usually are vacuum jacketed and may have a tertiary seal to permit purging the seal cavity with gaseous helium to prevent cryopumping and ice buildup on the sliding seal surface.

The ball-and-cone disconnect (fig. 3) is characterized by ball-shaped male half, usually the ground half, and a cone-shaped female half, usually the vehicle half. Disconnects for harmful or combustible fluids frequently have two ball-and-cone seals with vents between the seals for leakage measurement and disposal of acceptable leakage. The male ground half of the disconnect usually is loaded by a spring or bellows to maintain compression on the ball-and-cone seal.

In the pressure-balanced disconnect (figs. 4 and 5), the end of the male portion of the disconnect is closed, with holes in the side of the probe, and the female portion is a closed-end double-wall tube with holes in the inner wall. When the disconnect is mated, the flow of fluid from one portion to the other is essentially perpendicular to the centerline of the fittings and, with the closed-end fittings, the longitudinal forces are essentially balanced in each half. The lateral fluid forces are spread around the circumference of the male probe and thus are also balanced. Therefore, when this type of disconnect is separated, with fluid flowing under pressure, it imparts negligible lateral or longitudinal forces to the vehicle. For this reason, the pressure-balanced disconnect usually is used for lines that must be separated under pressure. As noted, pressure balancing may be incorporated in either the slip type or the ball-and-cone type.

Among the disconnects selected for the Space Shuttle, the rise-off disconnects were all slip types; for the Orbiter disconnects, the ball-and-cone disconnect with spring loading was selected for low pressures, the ball-and-cone with bellows loading for medium pressures, and pressure-balanced slip type for pressures over 500 psi.

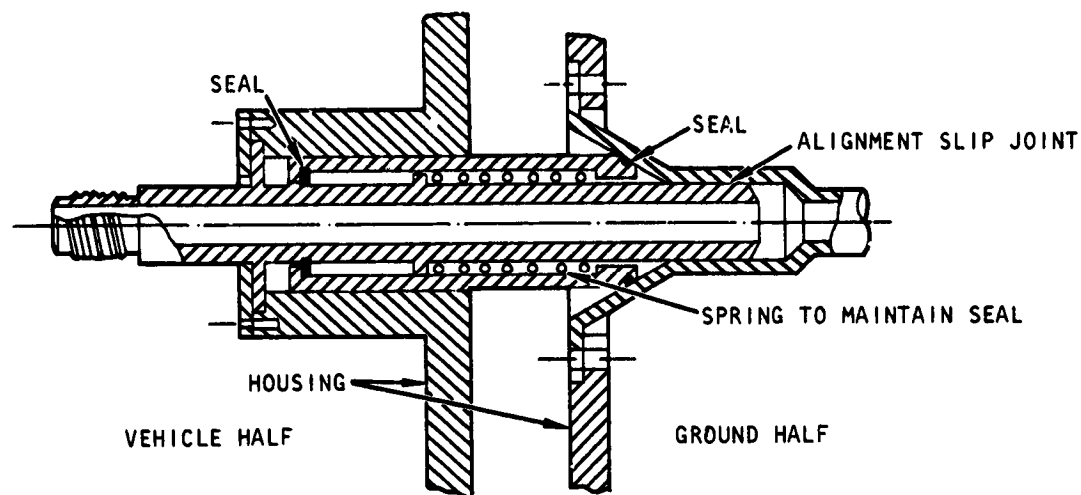


Figure 2. — Typical slip disconnect (engaged).

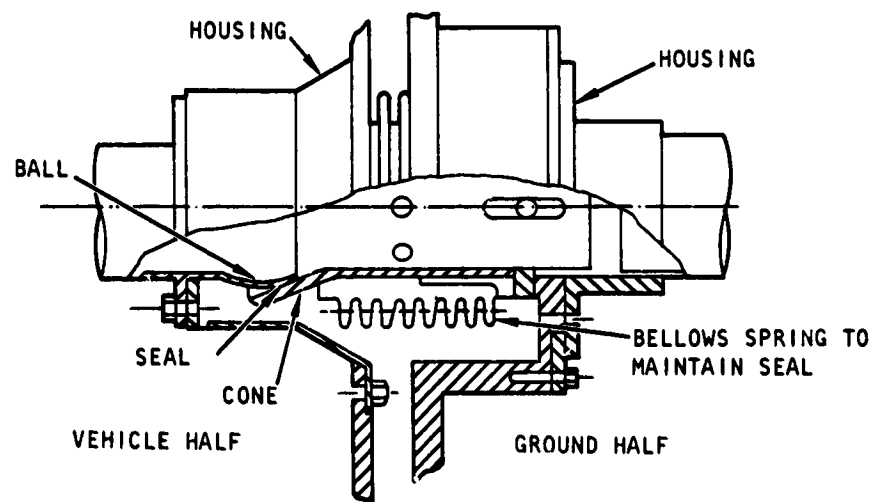


Figure 3. — Typical ball-and-cone disconnect (engaged).

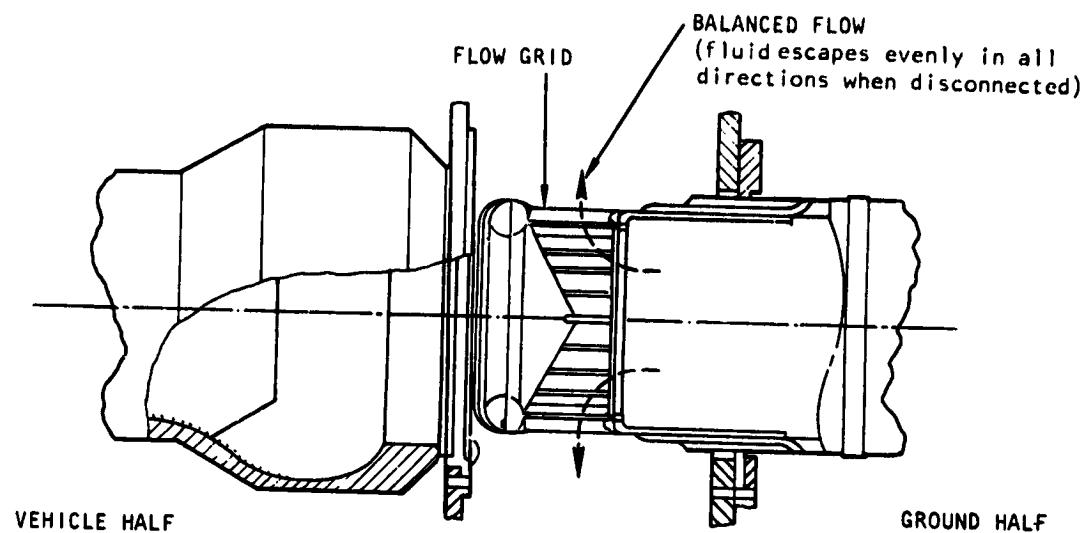


Figure 4. — Typical pressure-balanced slip disconnect (separated).

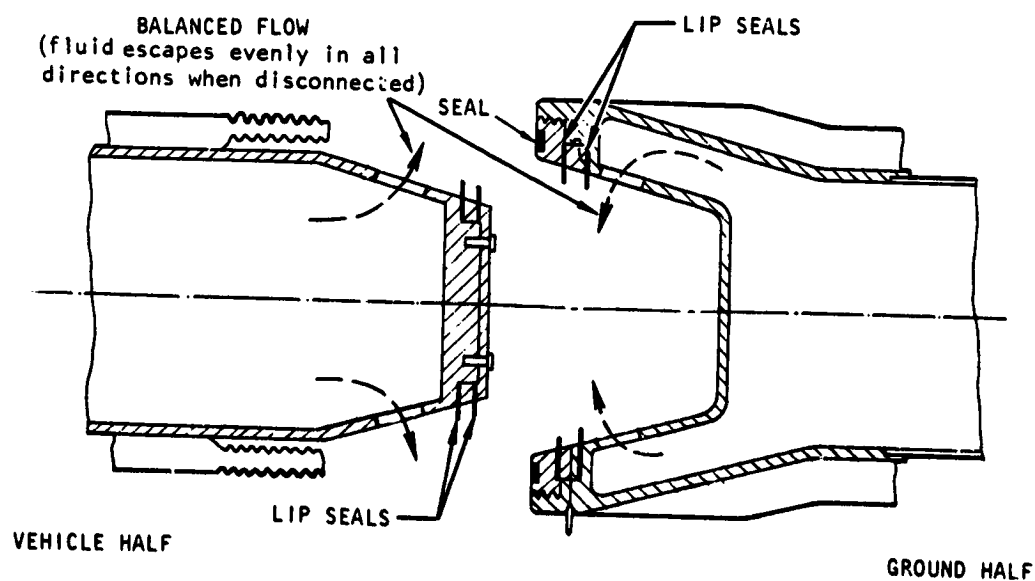


Figure 5. — Typical partially-pressure-balanced cone seal disconnect (separated).

### **2.1.1.3 FORCED-SEPARATION DISCONNECTS**

Applications of the forced-separation disconnects, which include both explosive- and pressure-actuated types, are very limited. Explosive-actuated disconnects are not used in current vehicle technology. Early in the Titan program, problems arose because the vehicle could not withstand the high forces generated by explosive-actuated disconnects, and these disconnects were replaced with breakaway disconnects.

Since most explosive-actuated disconnects make use of explosive-actuated valves having zero leakage, the primary advantages of these disconnects are zero-leakage capability, all-metal construction, rapid shutoff, small size, and simplicity. Disadvantages are (1) problems associated with handling hazardous explosive charges, providing structural capability to withstand the high forces, and handling the ejected ground half; (2) the inherent single-operation limitation; and (3) difficulties with providing a redundant release system. Design concepts for explosive valves are discussed in reference 5. In the disconnect design, the explosive energy that operates the valve acts simultaneously to cause failure of the latch and forcible ejection of the ground half. Thus, the latching mechanism can consist of a shear pin (or pins). In some designs, advantage has been taken of the high axial forces available to utilize a face gasket between the ground and vehicle halves of the disconnect.

## **2.1.2 Design Integration**

### **2.1.2.1 VALVES**

The basic valve function in disconnects is that of a check valve (ref. 5) modified as necessary to accommodate specific opening, latching, or closing requirements. Poppet valves normally are used, although ball valves (ref. 6) have been used where flow characteristics were sufficiently critical to justify their greater complexity or where a rotary-valve actuation coupled with the latching mechanism was desired because of hazardous fluid. Other considerations include spillage, leakage, possible ingestion of atmospheric air during separation, separation forces, and pressure drop. The factors that influence the choice of materials and the configuration details of valve seats, poppet guides, flow passages, and other design elements are not unique to disconnects and are treated in detail in references 5 and 6.

Except for those applications that require the valve to be pressure operated, the operating requirements may be considered to be relatively mild. For example, the valves are mechanically opened at low pressure and at ambient temperature and held open during exposure to flow and high pressure; the opening force is applied axially at a low travel rate and does not produce significant transverse loads on the poppet guide; a low cracking pressure and resultant low seat unit load is not a requirement; and the valves are not

subjected to rapid closing by sudden flow reversals while in the open position. Disconnect applications, however, do impose severe conditions in two areas with respect to valve design:

- (1) Widely varying flowrates may be applied for long periods of time. Pressure-operated valves thus are exposed to flow conditions that can cause chatter.
- (2) Frequent exposure to ambient conditions during the connection/disconnection phase of operation is conducive to damage of sealing surfaces by contamination or corrosion.

Chatter is minimized by using flow restrictors or lightly loaded springs that hold the poppet off the seat during low-flow operation (ref. 5). Contamination and corrosion are minimized by the proper selection of materials (sec. 2.6.1) and by installing protective covers (sec. 2.6.2) over the open disconnect. Figures 6 and 7 illustrate poppet valves that are held in the open position mechanically, one being externally actuated and the other internally actuated.

#### **2.1.2.2 INTERFACE SEALS**

Whenever the fluid and temperature requirements permit, an elastomeric O-ring is used for the interface seal. A typical installation is shown in figure 1. This configuration is simple and reliable.

In cryogenic disconnects, plastic lip seals generally are utilized between the two halves. Face seals have been used for some cryogenic applications; however, they have presented sealing and latching problems resulting from the valve opening prior to the seal engagement. A soft metal gasket between serrated flanges has been used successfully to contain chemically active fluids (e.g., liquid fluorine). One limitation on the use of the soft metal gasket is that it should not be used near a shock-producing device (e.g., an explosive-actuated valve or release device) because excessive shock levels may cause the gasket to yield beyond recovery and thus produce unacceptable leakage. Figure 8 shows this type of disconnect with a secondary seal and a method for venting and detecting dangerous leakage. When leakage is at a dangerous rate, safety precautions are initiated prior to seal engagement and detail tolerance studies are performed to verify seal function within tolerance limits. Another solution to this problem is the use of a separate operation to open and close the valve while the seal and latch mechanism is engaged.

Normally, the internal pressure is greater than external pressure and the sealing of internal pressure only is required. When the internal pressure is lower, or when the high-pressure potential alternates between internal and external, the interface seal must perform in either direction. Seal technology is discussed in section 2.5.

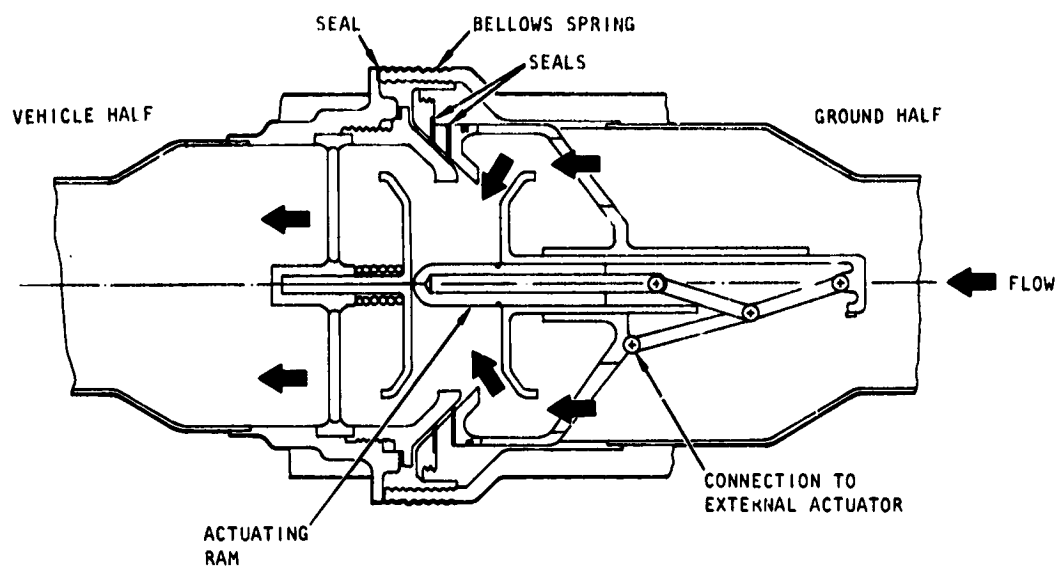


Figure 6. — Breakaway disconnect with poppet valves actuated externally (engaged).

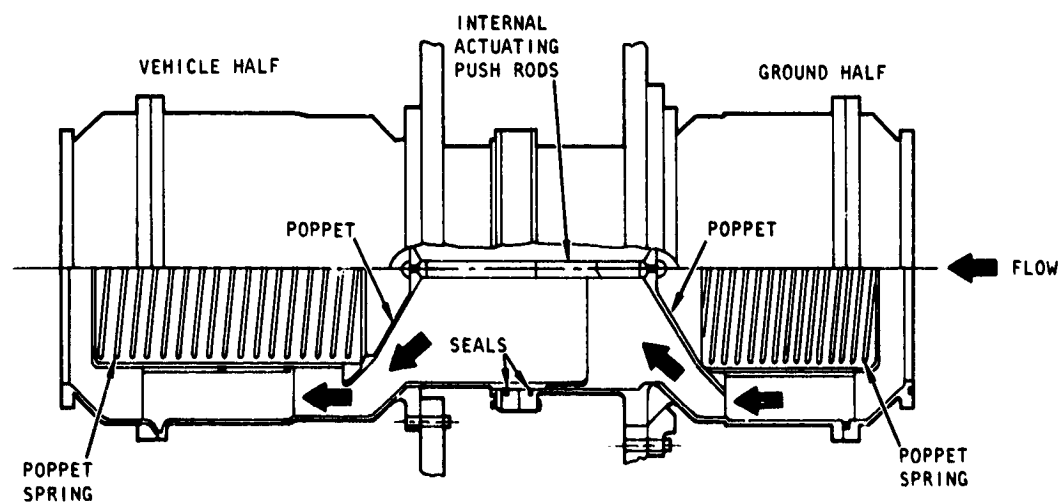


Figure 7. — Breakaway disconnect with poppet valves actuated internally (engaged).



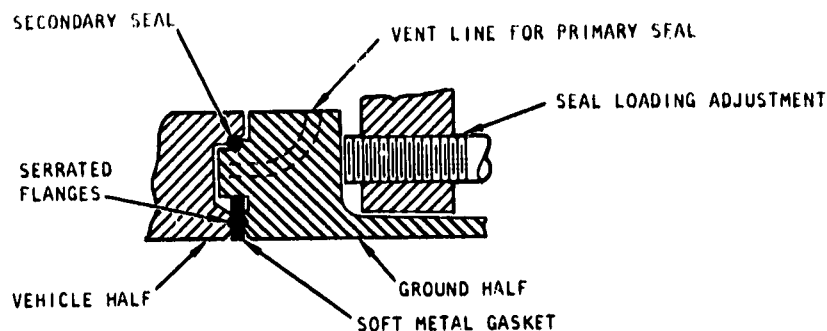


Figure 8. — Disconnect with metal-to-metal seal on serrated flanges.

### 2.1.2.3 LATCH/RELEASE MECHANISM

The primary function of the latch/release mechanism is to hold the two halves of the disconnect together when they are connected and to provide a clean-break separation of the two halves on command/operation. In addition to resisting separating forces due to pressure and equipment loads, the mechanism may also provide an axial load for the interface seal (ref. 7). In manually operated disconnects, the latch/release mechanism is relatively simple. The most common type employs a spring-loaded collar that, when connected, forces balls (or their equivalent) into a groove in the mating half, as shown in figure 1. When the collar is retracted, the balls are allowed to clear the groove, and the mating half then can be disengaged. When mated, the ground half is free to rotate to prevent twisting of the line. The flight half is rigidly mounted.

The motion required to operate the majority of manually operated disconnects is a push-pull motion. In some instances, push-pull designs are modified to require a twisting action to secure the latch after engagement, and to permit releasing the latch before disengagement. Such modifications are intended to prevent erroneous disengagement by requiring a deliberate specific action, or to provide additional resistance to loosening by vibration. A twisting motion also may be utilized to operate ball valves in conjunction with the latching and releasing operations.

The vehicle-half protective covers are used to provide protection for the exposed valve mechanism and, at times, to provide a redundant seal for the check valve. The protective covers utilize the same latching mechanism as the ground halves. When the protective cover is used to provide a redundant seal, the latching mechanism is designed to accept system operating pressures. These pressures may be in excess of any pressure that may exist when the ground half is engaged during ground operations. Thus, the structural requirements for



the latching mechanism design can be dictated by the protective cover requirements. Figure 9 illustrates a disconnect with flexible-cone and poppet debris valves, which protect both the vehicle and ground halves of the disconnect from debris after separation.

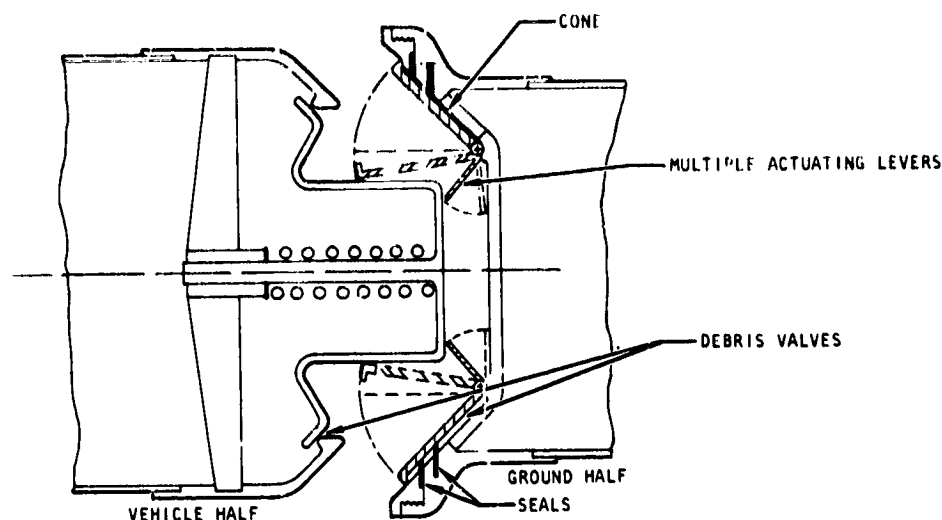


Figure 9. — Flexible-cone and poppet debris valves (coupling separated, valves closed).

Breakaway disconnects usually are panel mounted and do not always require individual latching or releasing mechanisms. For example, the latch/release mechanisms for the smaller disconnects on the Saturn V umbilical panels are mounted on the panels; the larger disconnects usually have self-contained latch/release mechanisms. However, the SIVB fill and drain or vent system disconnects do not have latch mechanisms. Ball-type locking devices, which are similar in concept to manually operated designs except that the locking member is pneumatically or mechanically actuated, are also used. The mechanical method normally is operated with a lanyard and can be used as a redundant system.

Latch/release mechanisms generally are trouble free; however, mechanism failures were a problem in early launches. For example, failure of a lanyard-operated disconnect device due to excessive loading prevented jettisoning of the fuel start system after launch in the second Navaho flight and resulted in destruction of the vehicle. Because of the disastrous consequences of failure to release and separate, current practice is to provide redundant release mechanisms in breakaway disconnects. On the Saturn V aft umbilicals, a pneumatic unlocking action is initiated by the launch commit signal, and when the vehicle rises 0.75 in., the ground-side panel is retracted hydraulically by the mobile-launcher service

arm. Should the primary pneumatic unlocking mechanism fail, a secondary lanyard-operated release mechanism is activated after a 2-in. rise. If both systems fail, the umbilical is disconnected by cam action from the vehicle after a rise of approximately 15 in. (ref. 8). Another potential problem is the vulnerability of the latch/release mechanisms to malfunction or permanent damage because of ice. To cope with this problem, shrouds frequently are used to prevent frost accumulation in critical areas. The shrouds are carefully designed to ensure that they do not interfere with latching mechanism operation. It is sometimes necessary to provide a purge in a shroud over a cryogenic disconnect to prevent the possibility of ice formation under the shroud.

#### **2.1.2.4 MOUNTING**

Disconnects of all types normally are attached individually to the vehicle or GSE at the location where required. In the case of the manually operated disconnects in particular, there is no significant advantage in combining more than one unit in a single housing; these disconnects serve their function better and simplify the system when they are located at the point of application. However, when a system requires use of closely grouped, manually operated disconnects of the same size, erroneous connections of incompatible subsystems are possible. Such connections can be avoided by making it physically impossible to interconnect ground halves and vehicle halves used in different systems. Several methods are used to accomplish this objective and still retain the advantages of common valve-latch and interface-seal elements. One method involves uniquely arranged keys and keyways or pins and slots. When for cost and spares reasons it is desirable to have a common design for disconnects of the same size, the panel and connecting lines are designed so that it is physically impossible to reach an incorrect disconnect of the same size.

The breakaway and forced-separation disconnects are mounted close to each other and occasionally on clustered panels to simplify launching operations. When mounted on panels, the configurations are designed to balance forces to avoid transverse loads. Whenever practicable, only completely compatible systems are clustered on a single panel; for example, oxidizer and fuel disconnects are mounted on different panels. Undesirable interactions are thus precluded. Proper alignment in clustered disconnects is ensured by providing an alignment adjustment, which is usually incorporated on the ground half to maintain simplicity on the vehicle side.

#### **2.1.2.5 SPILLAGE CONTROL**

Spillage is the amount of fluid in a double-valve disconnect that remains trapped within the partially disengaged assembly at the instant both valves have closed. Figure 1 depicts the spillage volume in a typical manually operated disconnect design. Since spillage also represents air entrapment during engagement, some systems handling liquids may be

sensitive to this volume from the standpoint of air inclusion. An extremely important aspect of spillage control is the control of large amounts of spillage that can cause "water hammer" in a fluid system.

In most aerospace systems handling nontoxic fluids, spillage is not critical. However, when it is critical, spillage control is directed toward minimizing the volume between valve seats as well as the displacement between the valves following valve closure during disconnect separation. Figure 10 shows a method of achieving minimum spillage volume. Through careful design, it is possible, for example, to reduce the spillage volume in a 1/2-in. disconnect to less than  $0.05 \text{ cm}^3$  (ref. 9).

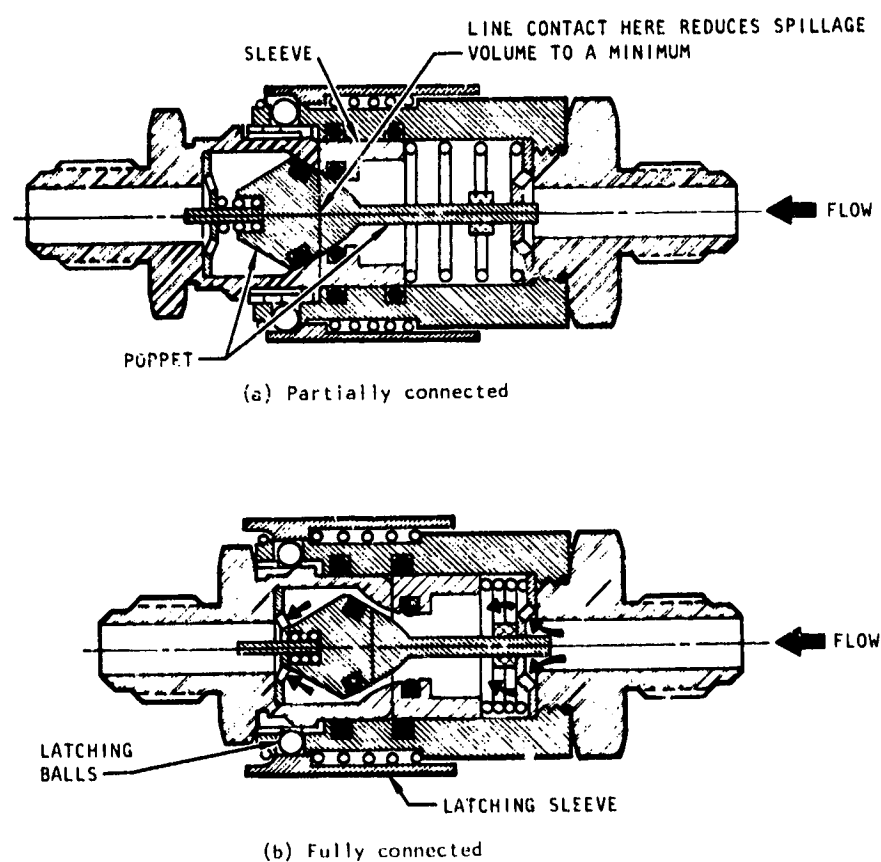


Figure 10. — Two-valve disconnect designed for minimum spillage.

## 2.2 COUPLINGS

Couplings are manually actuated separable connectors that require more than a few seconds for engagement or disengagement. Types of couplings include static flange, threaded connector, and slow-rotating or -oscillating dynamic swivel coupling. Couplings are used in fluid systems to connect lines or components in a "leakproof" connection that can be disconnected. This section covers the coupling load-carrying structure and the influence of the seal on coupling design; seals are covered in detail in section 2.5. The basic design information for fluid couplings is well documented in references 9 through 14.

A coupling design for use in a liquid rocket must satisfy the following requirements: (1) "zero leakage" for man-rated systems (a philosophy that has developed through the Saturn vehicle development and flights), (2) high reliability, (3) low weight, a consideration in direct conflict with high reliability, and (4) low cost, a requirement that must be compromised to meet the other requirements. A general discussion of the problems encountered in coupling design and the various tradeoff considerations are included in reference 15.

The need for zero leakage was established early in the Saturn program because of the large number of couplings involved and the impact of a fire or explosion that could result from leakage. The success of the Saturn vehicle series is strong evidence that the philosophy of "zero leak" is a good one, even though a firm definition of "zero leakage" is lacking (ref. 16).

In order to obtain the optimum coupling designs, a structural analysis is performed on the couplings; this analysis takes into account all internal pressure conditions, external loads, and thermal conditions. A comprehensive connector analysis intended to optimize the coupling design with respect to restraining forces and rigidity without over-designing has been prepared (ref. 17); however, the analysis has not been verified by test.

### 2.2.1 Configuration Design

#### 2.2.1.1 FLANGED COUPLINGS

Typical flanged-coupling configurations are shown in figure 11. These coupling configurations are modified as required to mate with the various types of seals. Flanged couplings are used where loads require the type of restraining force provided by bolts or vee clamps or where coupling reliability dictates the use of more than one threaded clamping fastener. Failure of a threaded coupling by thread disengagement can be catastrophic, whereas failure of one of a number of bolts on a flanged coupling may have less impact. This

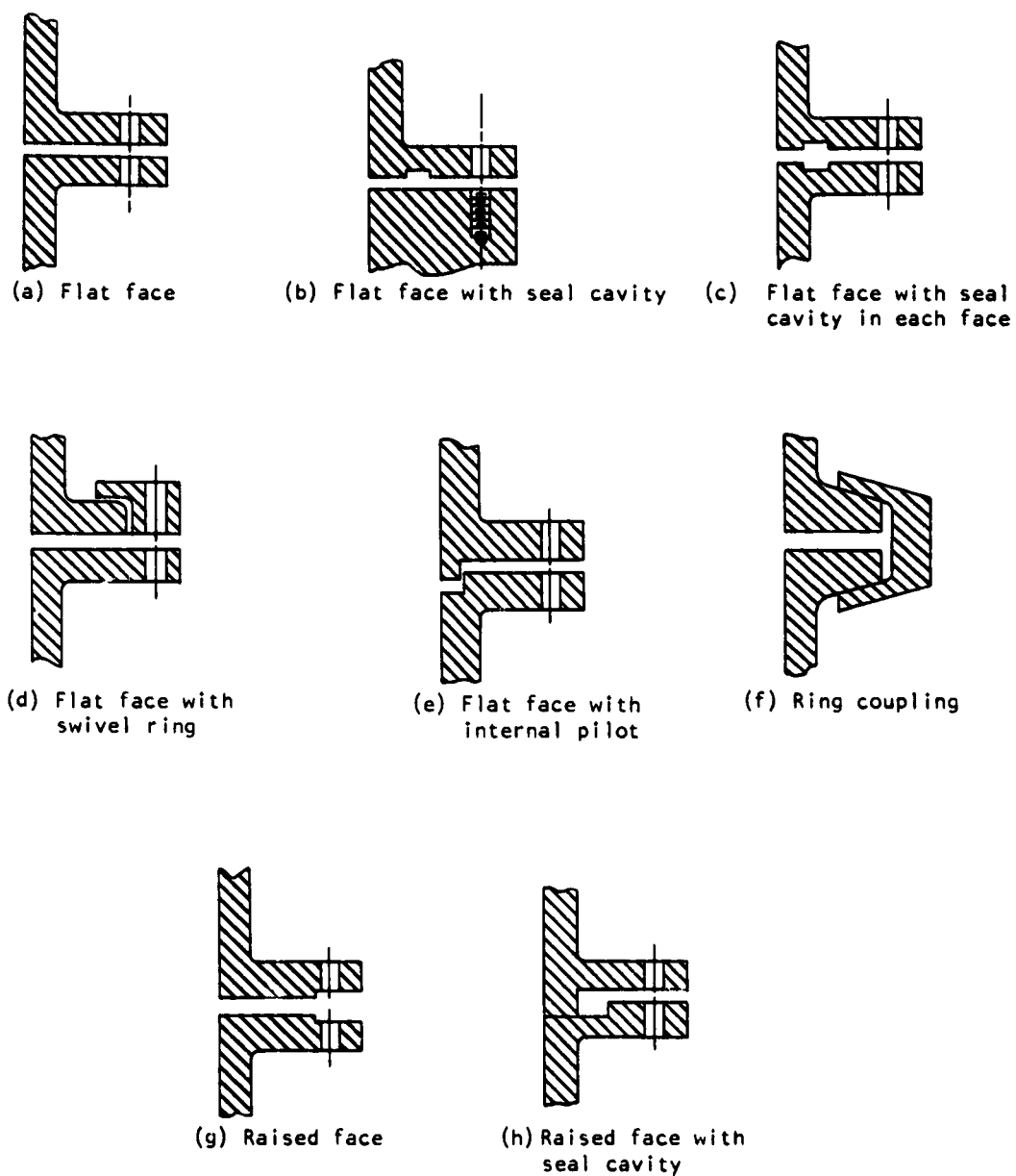


Figure 11. — Basic types of flanged couplings.

factor was one of the reasons that threaded couplings on the F-1 engine were limited to zero-pressure drain systems. The coupling configurations used on this engine consisted primarily of flat-face flanges, flat-face with seal cavity, and flat-face flanges with swivel ring. The swivel-ring design was used where alignment of bolt patterns presented a problem. Leakage-monitoring capability (sec. 2.2.2.4) was provided on joints that were considered critical or were representative of a family of joints of similar design and operational environment. On the J-2 engine, flat-face flanges, flat-face with swivel ring, and threaded boss couplings were used, with leakage-monitoring provisions at each coupling.

### 2.2.1.2 THREADED COUPLINGS

Basic types of threaded couplings (or connectors) are shown in figure 12. The standard flared-tube coupling (fig. 12(a)) was developed before World War II, principally for aircraft

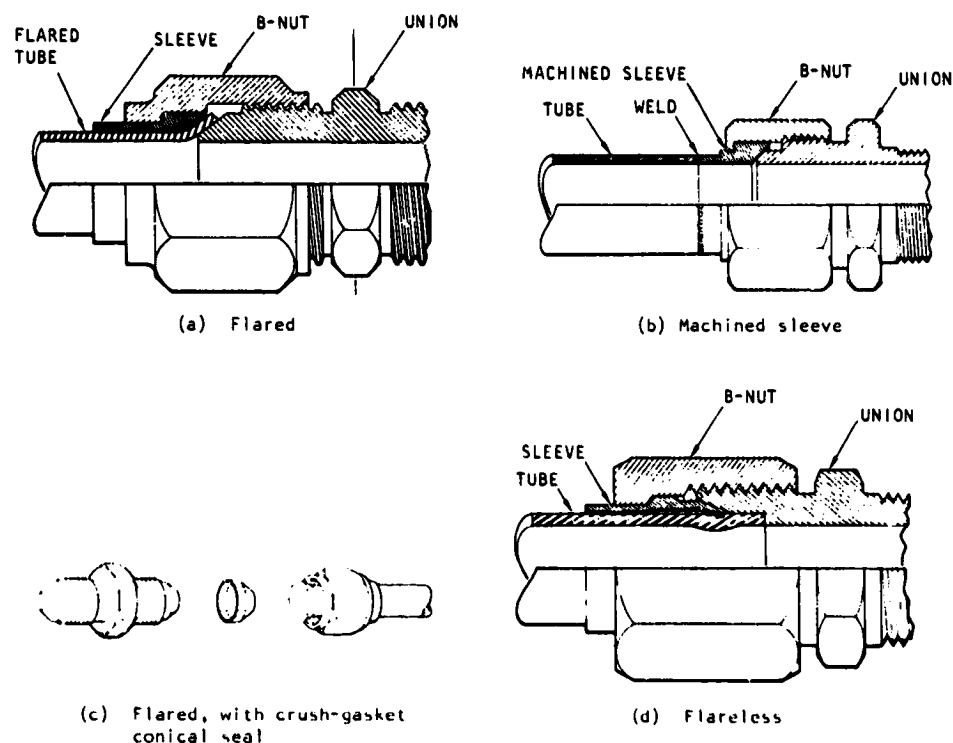


Figure 12. — Basic types of threaded couplings.

hydraulic systems, and was carried over into pneumatic and hydraulic installations on the early aerospace vehicle programs (Thor, Jupiter, Atlas). Aluminum components for connectors and lines proved adequate in sealing characteristics but had limited strength to contain the system fluid in high-temperature or high-vibration environments; further, the aluminum components were subject to seizing under over-torque conditions or in extreme environments. Stainless-steel components for connectors and lines proved difficult to seal because of the poor yield characteristics of the seal material and the difficulty in controlling the sealing surface of the components and tube flares involved. Stainless steel also work hardens and can crack during the flaring operation. Flared-tube couplings, both aluminum and stainless steel, are subject to relaxation of B-nut torque as a result of creep at the seal interface or within the threads and nut backoff under vibration. Leakage at normal installation torque can occur because of improper surface finish, misalignment, contamination, or inadequate seal interface load.

A search for a better flared coupling to solve leakage problems on the H-1 engine program led to the use of a machined sleeve (fig. 12(b)). The machined sleeve combined the flare and the ferrule, or sleeve, into one machined part, which was then welded into the tube assembly at a safe distance from the flare to prevent warpage. A variation of the butt-welded, machined-sleeve connector illustrated in figure 12(b) – a silver-brazed, machined-sleeve connector – is in general use in small plumbing on the RL 10 engine. The sealing surface on the tube assembly ferrule is finish-machined before the brazing operation.

The machined sleeve has the following advantages over the flared tube:

- (1) Tolerances in the conical-tube sealing surface can be closely controlled.
- (2) Increased rigidity of the sleeve resists distortion at sealing.
- (3) The weak point of the tubing at the base of the flare is eliminated.
- (4) Seizing between the tube and sleeve is eliminated.
- (5) Greater hoop strength helps prevent seizing between the sleeve (now the tube end) and coupling nut.

The machined sleeve retains two problems common to the flared coupling: a metal-to-metal seals, and the same type B-nut for effecting the coupling.

A different approach for achieving a seal on the flared connector (fig. 12(a)) is the use of a soft material generally copper or aluminum, between the steel interfaces (fig. 12(c)). This concept improved the likelihood of making a leak-free coupling; however, some of the above problems still exist (e.g., torque relaxation due to vibration, tube cracking during flaring



operation), and new problems were created (e.g., improper seal installation, which orificed the flow, and galling of the seal, which introduced metal particles into the fluid system).

Various flareless-tube threaded couplings (fig. 12(d)) have been developed to eliminate the flaring problems. These couplings still have the problem of sealing with metal-to-metal contact and, in some cases, have increased the number of potential leak paths. Other problems include yielding of the ferrule, which results in leakage, and the requirement that the external surface of the tubing must be damage-free to effect a seal between the ferrule and tube outside diameter. These couplings have very limited use on large rocket engines; however, they have been successfully used throughout spacecraft systems where the vibration is not severe and the pressures are not high. They have also been used successfully on thruster engines where the lines are small.

In order to overcome the weakness of the metal-to-metal-seal, brute-force couplings discussed above, connectors with a sophisticated sealing approach were designed (ref. 9, par. 5.12.3.14 and 5.12.3.15). These designs have overcome most of the problems with the metal-to-metal threaded couplings by providing an effective seal element and by using an improved loading method through a parallel load path to retain coupling tightness. In some of these designs, a separate seal is used, a provision that can result in installation problems (e.g., seal not installed or misaligned). Also, prior to assembly, sealing surfaces must be inspected visually with suitable aid (10 X enlargement) to detect flaws, scratches, pits, and other surface imperfections. Seal problems are minimized by performing leak checks that verify the seal integrity.

Another threaded-coupling configuration is the threaded boss coupling (fig. 13). The boss coupling is used for instrumentation ports with standard plugs or for a tapoff coupling for a fluid line.

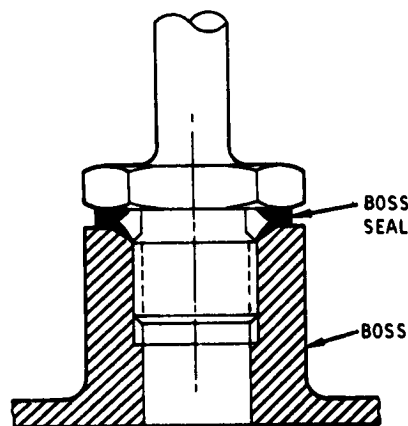


Figure 13. — Typical threaded boss coupling.



In the initial stages of the F-1 engine program, design specifications were established to restrict the use of threaded couplings in functional pressurized tubing to 1/4-in. diameter or less in order to achieve high reliability. For larger tubing and ducting, flanged couplings were used. These practices resulted in most of the tube-flared and machined-sleeve couplings being restricted to instrumentation and drain lines. To minimize the problem of B-nuts being loosened by vibration, a self-locking nut (deformed thread type) was used. Threaded-boss couplings with union and B-nut were used for test instrumentation ports, but a plug and seal were installed prior to engine delivery.

In the RL10 engine, large-diameter (1.625 in.) threaded couplings are used on low-pressure (< 5 psi) ducting. Flat Teflon-coated aluminum gaskets are employed under threaded instrumentation-boss couplings in pressure systems up to 1000 psi in LH<sub>2</sub> and 600 psi in LO<sub>2</sub>. Threaded instrumentation taps in high-pressure, thin-wall lines utilize set-in, butt-welded bosses. Conventional B-nuts without a self-locking feature are used throughout the RL10 engine; loosening of the nuts is prevented by the use of safety wire rather than by a self-locking feature in the nut.

In the Apollo spacecraft, various types of threaded couplings were used where access or installation dictated a separable connector. A tabulation of the couplings and the system in which they were used is presented in reference 18.

### **2.2.1.3 DYNAMIC SWIVEL COUPLINGS**

Swivel couplings, where the coupling serves as a dynamic interface, have been used on some rocket engines (e.g., on the E-1 and X-1 experimental engines, and on the Atlas vernier engines). However, because of leakage, limited life, reliability problems, and unsuitability for use with cryogenic systems, swivel couplings were not used in the ducting system on the Saturn engine. On the Saturn engines, design approaches included a rigid ducting system with alignment spacers for the pump discharge ducting, and flexible ducting with either bellows or flexible hoses for gimbaling ducts (ref. 19). Swivel couplings were used successfully between moving members of mechanical components (e.g., valve stems, actuator shafts, and pistons). However, provisions were made to dispose of leakage, and extensive testing was performed to determine the cycle-life limits and to establish reliability. Leakage is inherent in this type of coupling because of the sealing problems. For elastomeric seals in systems with liquids such as RP-1 or hydraulic oil, the leakage provides the lubricant necessary for long cycle life. The leakage is vented to a safe area to minimize accumulation of leakage products and prevent them from contributing to or igniting a fire.

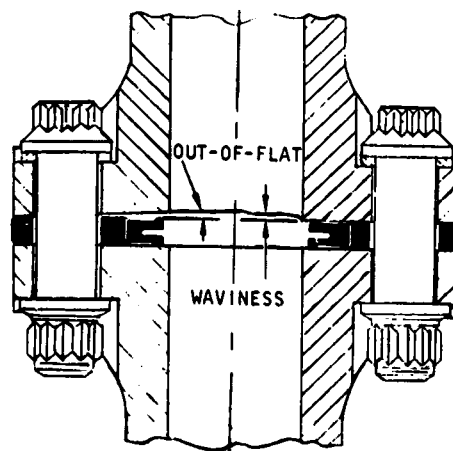
## **2.2.2 Design Integration**

### **2.2.2.1 SEAL CONTACT SURFACE**

The success of the coupling in limiting leakage depends on its capability for providing the necessary seal contact. The interface requirements involve the seal housing, surface finish,

machine lay, and surface flatness and waviness. A summary of the surface finish and flatness requirements for aerospace-type seals is included in section 2.5 (table I). The sealing ability of the coupling is directly related to the flange surface condition, since a surface that is smooth and approaches a true plane provides more seal contact area. This contact in turn creates a greater resistance to fluid flow. Reference 20 defines surface texture, surface roughness or finish, and waviness.

Surface flatness and waviness (fig. 14(a)) are factors in seal leakage either because these requirements are not specified on the drawing, or because manufacturing personnel do not understand the requirement. Flatness and waviness control is necessary to limit the surface contours to ensure their compatibility with the seals. Excessive out-of-flatness or waviness absorbs some of the seal's deflection capability without generating the required design sealing load. If flatness and waviness are excessive, the seal is unable to follow the flange surface contour in the radial or circumferential directions; this condition results in a coupling leak. A general practice for specifying flatness and waviness is to establish limits based on the surface finish requirement; and if a tighter flatness or waviness tolerance is required, it is specified on the drawing. Typical flatness and waviness requirements, as used on the F-1 and J-2 engine programs, are shown in figure 14(b) as a function of surface finish. When a seal requires a particular flange surface finish to seal a fluid, that finish usually is specified on the engineering drawing, and the flatness and waviness are automatically applied through the use of a machining specification (ref. 21).



(a) Flange flatness and waviness

Surface finish, $\mu$ in.	Tolerance	
	Flatness, in./in.	Waviness, in./1/2 in.
4	0.00008	0.00004
8	0.00016	0.00008
16	0.00032	0.00016
32	0.00063	0.00032
63	0.0012	0.00063
125	0.0025	0.00125

(b) Typical specifications

Figure 14. — Illustration of flange flatness and waviness and typical specifications.

An installation of a typical pressure-assisted plastic-coated seal for cryogenic fluid calls for a 32- $\mu$ in. surface finish with a circular lay (phonograph record profile), flatness within 0.0006 in./in., and waviness of 0.0003 in./0.500 in. An acceptable alternate to the 32- $\mu$ in. finish is a multidirectional finish of 8  $\mu$ in., which permits alternate fabrication methods and allows a damaged surface to be repaired without re-machining. For metal-plated seals, the same specifications for flatness and waviness are used, with the surface finishes varying from 32 to 8  $\mu$ in. the value depending on the leakage and interface loads. If the leakage requirements for the joint are extremely low, a 16- $\mu$ in. finish is used.

Another coupling/seal interface problem arises from surface imperfections. Surface imperfections may be inherent in the material (e.g., porosity in castings or stringers in wrought material) or, more frequently, they are incurred through handling abuse during fabrication, assembly, and installation. If these imperfections occur where the seal is made, they can impair sealing ability unless the seal unit load is sufficiently high to cause the seal material to flow into the voids. A general practice consists of specifying the critical flange sealing surface on the engineering drawing so that manufacturing personnel can take the steps necessary to obtain and retain the area free of imperfections. If a seal interface is damaged when the coupling is installed on an engine, the part may be removed for rework or it may be resurfaced in place, provided that the design requirements for waviness and flatness are attained.

Finally, tolerance accumulation can also present a seal contact problem and result in coupling leakage under extreme conditions. Tolerance conditions uncovered on J-2 and F-1 engines included (1) seal/flange lateral shift that permitted the seal to overlap the inside of the duct or overlap the leakage monitoring cavity, (2) boss/seal misalignment, and (3) bolts bottoming in a blind hole or running out of thread. Tolerance problems currently are minimized or eliminated by tolerance-extreme studies as outlined in reference 22. On the RL10 engine, piloted flanges are used to align tubes to bosses and to keep seals and bosses aligned and out of the flow stream.

#### **2.2.2.2 RIGIDITY**

Coupling rigidity enables the coupling to resist deflections and distortions due to mechanical or thermal loading. The degree of rigidity required for a coupling is related directly to the seal element that must compensate for deflections and distortions. Typical deflections and distortions that can cause leakage problems with flanged couplings are shown in figure 15.

Flange bowing is the tendency for the flanges to separate between bolts as a consequence of thin flanges or an inadequate number of bolts. This problem is prevented by increasing the flange thickness or the number of bolts or both. Reference 23 provides a detailed analysis of bolt spacing for flanged couplings. Flange rotation denotes the tendency of the coupling to separate at the seal surface as a consequence of (1) thin duct walls that have too little

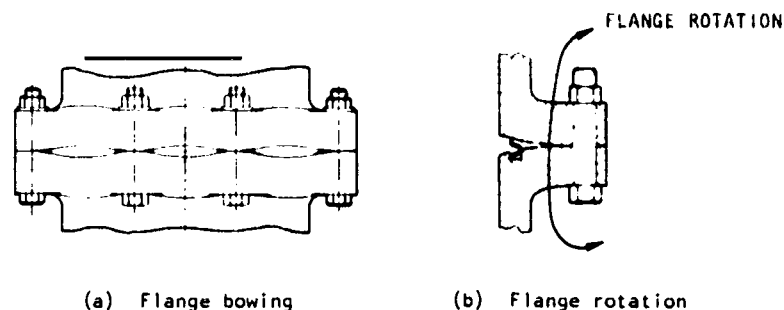


Figure 15. — Two kinds of flange deflection resulting from lack of rigidity.

resistance to the rotating moment, (2) flanges that lack sufficient rigidity and consequently deflect under high separating loads, (3) insufficient flange contact outside the bolt circle, (4) insufficient flange thickness, and (5) an adverse thermal gradient that prevents the flange from expanding without constraint. The flange rotation problem normally is resolved either by increasing the flange rigidity or by increasing the ability of the seal to follow the flange separation. The current coupling design practice permits deflections within the capability of one of the high-resilience type of seals (metal or elastomer), so that coupling size and weight are kept to a minimum. Flange deflections are analyzed by methods presented in reference 24.

In the coupling design concept used on the Space Shuttle Main Engine (SSME), the point of rotation during flange rotation is placed just on the pressure side of the seal and as near to the seal as possible; a gap between the two flanges is provided (fig. 16(a)). Any rotation of the flange, therefore, has little effect on the seal. Figure 16(b) shows the design generally used on the large Saturn engines. The two figures illustrate the difference in size that can be achieved in sealing the same fluid pressure by using the SSME design with high-strength material (Inconel 718) for the flanges and the bolts and by measuring the fastener preload with the ultrasonic method described in section 2.2.2.3. The weight saving on the SSME with this coupling design, in comparison with the weight required by the Saturn concept, was over 1200 lbm per engine. This new design has been proven successful on SSME qualification tests for very-high-pressure applications: 10 000 psi operating pressure, 18 000 psi test pressure.

### 2.2.2.3 RESTRAINING MECHANISM

The restraining device provides the force required to keep the coupling together under all operating loads. The loads on a coupling are both internal and external. Internal loads are

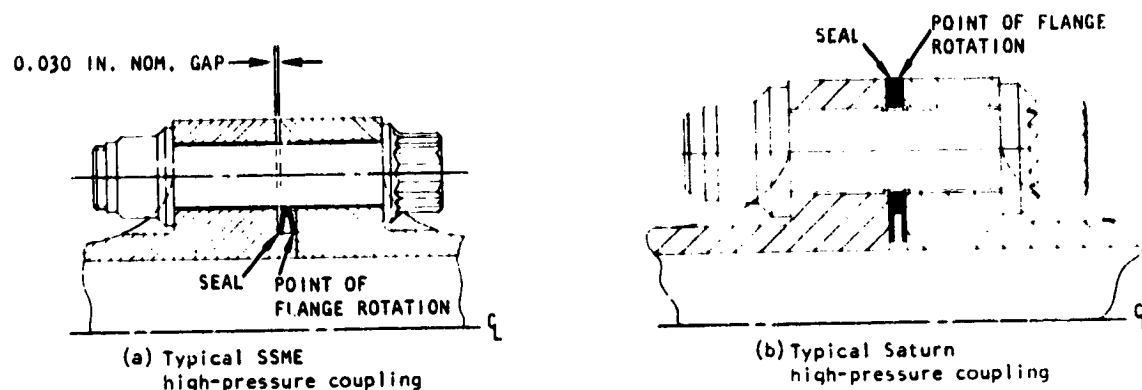


Figure 16.— Comparison of flanged-coupling designs used on SSME and Saturn engines.

due to the system pressure; these loads include the effects of surges generated by operation of components in the system and by the system operation itself. External loads include vibration and misalignment loads transmitted through the ducting or components, torque from rotating machinery, thermally-induced loads, seal loads, acceleration loads, and the load from the coupling restraining mechanism itself.

To provide the required restraint, the restraining device is preloaded to seat the seal at a compressive load greater than the separating loads. Insufficient preload will result in leakage if the resilience of the seal is unable to compensate for the separation that occurs in operation. Also, a gap between the flanges can result in seal abrasion in an installation subjected to high vibration. Coupling restraining problems have occurred on both cryogenic and hot-gas applications as a result of installation preload, loss of preload because of a thermal gradient, and bolt yielding at high temperatures. Such problems are resolved by analyzing the coupling and specifying the proper bolt loading. References 9 and 25 provide detailed information on coupling preload and preloaded bolts. Flange or coupling nuts and bolts not designed for the required levels of shock or vibration allow flange or coupling deflections, and intermittent leakage results. This point is specifically addressed in reference 26.

The restraining mechanisms for couplings consists of the nut and tube flare on threaded couplings, and either bolts or ring clamps on flanged couplings. Most liquid rocket engine couplings are restrained by bolts because of high clamping-load requirements and the large duct diameters, which prohibit the use of threaded couplings. Ring clamps, in comparison with bolted flanges, have limited use because of the large weight and envelope requirement for high-pressure applications. Another problem with ring clamps is difficulty in determining

the clamping load; the friction between the ring and flange results in an uneven load around the circumference and causes a wide tolerance band in clamping load.

A significant problem with bolted systems has been the inherent inaccuracies in the common methods for preloading bolts. The usual method of torquing with the standard torque wrench is considered to be only about 50% accurate because of frictional variations from unit to unit. This variation limits the design bolt load to the minimum expected value and does not permit the optimum utilization of the bolt ultimate strength. The result is that the coupling requires thicker flanges, larger bolts, and more bolts than if a more accurate preload method were used. Because of the high pressures required on the SSME, large, heavy couplings were required, and these couplings became a significant weight problem. As noted, the coupling design was changed as shown in figure 16. In addition, the preload for the SSME coupling bolts is determined by measuring bolt-length change with an ultrasonic device. A transducer mounted on the bolt head sends a sound pulse down the length of the bolt. The wave reflects from the end of the bolt back to the transducer. The time span for the wave to traverse the length of the bolt and return is determined. The instrument is calibrated to read changes in length as a function of time. The method, however, involves more than bolt-length change, since material density due to strain, bolt bending, and twist also affect the ultrasonic-wave travel through the bolt material. With the use of ultrasonics, the coupling-bolt-preload errors have been reduced to approximately 10%, permitting greater utilization of bolt ultimate strength. The use of this method of preloading bolts combined with high-strength material and improved seal and coupling design has resulted in a smaller, lighter coupling for the SSME than was used on the Saturn (fig. 16).

#### 2.2.2.4 LEAKAGE MONITORING

A requirement for monitoring coupling leakage was implemented early on the Saturn program, so that coupling leakage could be monitored on bench, subsystem, static engine firing tests and, if desired, even on critical couplings in flight. Leakage monitoring was provided by incorporating a low-pressure secondary seal concentric with the primary seal and a bleedoff passage in one of the mating flanges between the two seals. The secondary seal provides sufficient resistance to leakage past the primary seal to force flow out through the bleed passage and an attached flowmeter. These flow passages may also be used to collect leakage past the seal, so that combustible fluids may be separated and directed to a suitable venting area. Figure 17 shows typical monitoring provisions for couplings.

Design penalties are incurred in providing leakage-monitoring capability. The coupling diameter must be increased to permit incorporating a secondary seal inside the bolt circle. This change increases the pressure separating load, which in turn requires larger bolts and results in increased weight. Monitoring provisions in general have utilized a boss coupling, which requires an increase in flange thickness on some of the thin flanges. In some instances, the seals overlap the flange monitoring hole; to prevent this condition, a tolerance-accumulation study is performed.



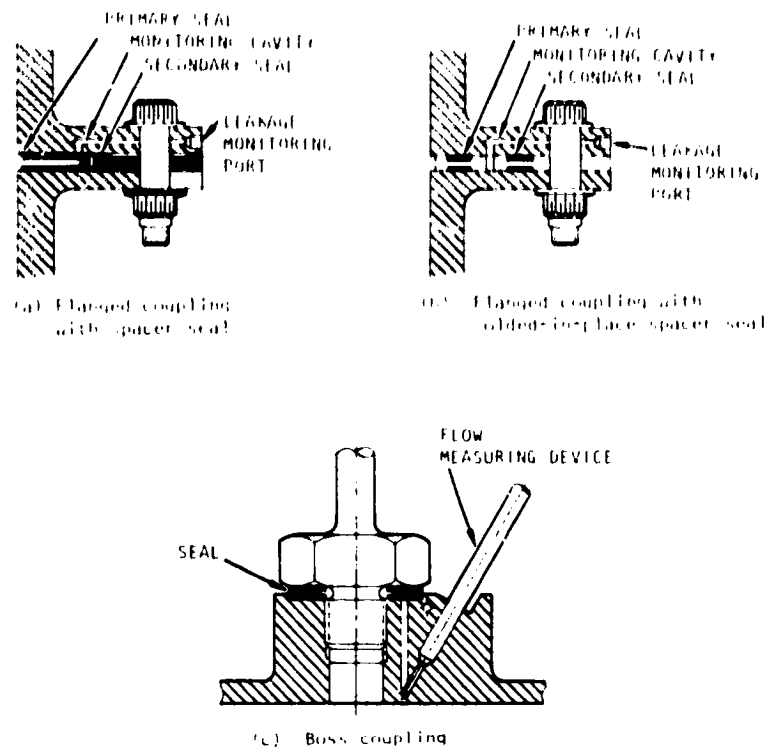


Figure 17. — Provisions for monitoring leakage at a coupling.

On the J-2S engine program, attempts were made to minimize the increased flange weight required by leakage monitoring. The objective was achieved by incorporating monitoring provisions in the seal and by replacing the large boss with a small-diameter tube (fig. 18). Use of the small-diameter tube in the seal provides a monitoring capability that can be readily deleted from the design after the coupling configurations have been verified. This feature results in a less expensive component. References 27 through 30 provide information on leakage-monitoring techniques and leakage-monitoring data from test programs conducted on the F-1 and J-2 rocket engines.

## 2.3 FITTINGS

Fittings are devices used to change flow area or direction while connecting two or more straight elements in a tubing, line, or ducting assembly; these are the T's, Y's, etc., used

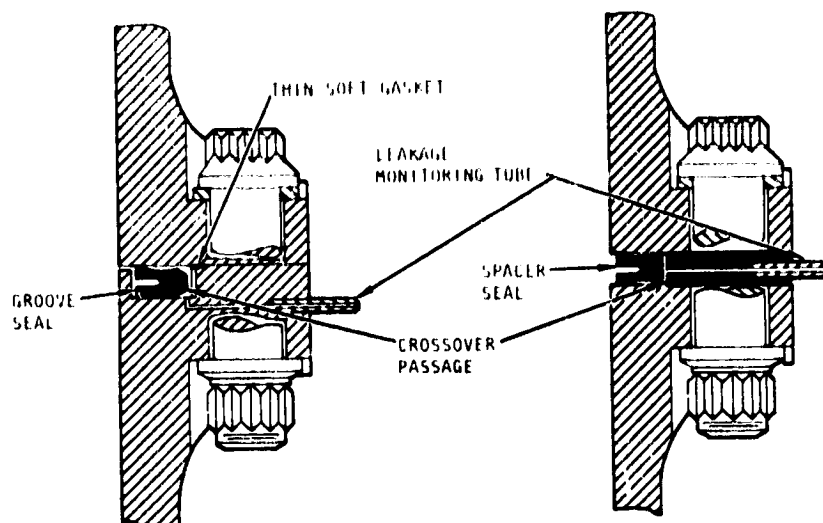


Figure 18. — Lightweight leakage-monitoring provisions.

to route fluids to required areas. Fittings also include the flow-control devices (e.g., flow straighteners) installed inside line or duct assemblies. Standard commercially available fittings or standard tube fittings are not discussed in detail because design information is readily available in supplier's handbooks and catalogs. These fittings usually are specified as originally designed.

## 2.3.1 Configuration Design

### 2.3.1.1 TUBE FITTINGS

Conduits or lines less than 1½-in. in diameter generally are referred to as tubes or tubing. Tube fittings used in recent rocket propulsion systems are designed mostly for simplicity in routing of the tubes, and to a lesser degree for ease of manufacturing the fitting. Flow efficiency of the fitting itself seldom is a consideration unless pressure loss of the entire tubing system is excessive. The most successful approach to tube fitting design has been utilization of a one-piece fitting machined from a bar or a forging. This design characteristically has cylindrical projections machined to the size of the tube outside diameter through which are bored holes equal in size to the tube inside diameter. The tubes



are welded to these stubs with a radiographically inspectable butt weld. The internal flow passages then are the intersections of drilled holes that connect the tubes along a rather tortuous flow path. This type of fitting has obvious advantages of compactness, structural adequacy, and leak tightness. Often mounting lugs are provided in the one-piece construction of the fitting for supporting the tubes on the engine structure. This practice provides mounting rigidity unequaled by most clamping methods.

Fittings are used in place of simple tube bends only where the routing requires a bend radius sharper than that available by tube bending techniques. Fittings are also required when more than two tubes are connected. Simple tube transitions between two tubes of different diameters have been accomplished with one-piece machined fittings as previously described, but have been made by other means such as swaging the larger tube diameter to meet the smaller in a butt-welded joint. Sometimes an area transition is made abruptly at a flanged interface.

Fittings designed for low pressure drop are more costly to manufacture than those that simply intersect cylindrical passages. These fittings typically are made in more than one piece and require internal machining to smooth the passages for a more desirable flow path. One example of this type of fitting is in the J-2S heat exchanger where a cross fitting is made in an "H" pattern to connect one small tube efficiently with three larger tubes. One problem discovered in this application was that the orificing from weld droptrough at the tube joints added significant, unnecessary pressure loss. This condition was corrected by improving the welding technique to eliminate droptrough during welding. Low pressure loss usually necessitates close concentricity at the tube joint; otherwise, a change in flow area and direction occurs abruptly at the mismatch and causes unnecessary pressure loss.

Elbow tube fittings are designed for low pressure loss by specifying proper geometric configurations (bend radius) where space permits. References 31 and 32 present data of the type used for optimum sizing of elbows.

Where tubes are required to tap flow from high-velocity ducts at low pressure loss, special takeoff fittings have been designed to protrude upstream into the main flow of the duct. This design allows fluid to enter the tapoff tube at the main flow total pressure and take an optimized turning loss to the duct wall. Typical applications of these taps are propellant supply lines for gas generators or other secondary combustors where it is desirable to inject propellants at a pressure as close to the pump discharge pressure as possible.

Fluid traps in internal flow passages of fittings can collect cleaning fluids that can promote corrosion. Entrapped impurities, chemically incompatible with oxidizers, have caused violent reaction or explosion when contacting oxidizer during filling or engine operation. Trapped toxic propellants have vaporized after engine shutdown and caused bodily harm to personnel working in the area. Fittings designed for self-draining solve these problems.

### 2.3.1.2 DUCT FITTINGS

Fittings used in rocket engine ducts (conduits larger than 1½ in. in diameter and currently up to 35-in. diam.) usually are given more consideration for minimizing pressure loss than is given to tube fittings. In pump inlet and discharge ducts or turbine drive gas ducts, pressure losses reduce engine performance, whereas pressure losses in small tubing usually do not affect engine performance to any measurable degree.

Five fundamental rules for avoiding excessive pressure loss in a duct are followed: (1) maintain a constant flow area, (2) use smooth flow surfaces, (3) utilize a minimum length duct, (4) keep flow directional changes to a minimum, and (5) make area and directional changes as gradual as possible. In addition, the overall duct assembly is designed to minimize length and directional and area changes. The same rules apply to the detail design of the fittings themselves.

A common method for designing a duct fitting is to call for the fitting to be made from two or more formed sheet metal pieces joined by butt welding. This approach allows considerable flexibility in design of the flow paths within the limits of formability, as long as at least one plane of symmetry in which to make a weld seam exists. If no wrinkling occurs during forming, the sheet metal surfaces as formed are of acceptable smoothness, and flow paths can be optimized to satisfy the other rules for minimizing pressure loss.

The most common fitting is the radiused elbow. Space permitting, it can be designed to be a low-pressure-loss fitting with proper geometry (refs. 31 and 32); however, space limitations often do not permit such optimization. Expanding and reducing elbows have been built for several engines, and even a cast elbow, with one bend followed by a "Y" juncture into two vaned bends, has been used successfully for the fuel pump inlet on the F-1 engine. Vaned elbows have been successfully used on the Titan III propulsion system. "T" and "Y" fittings have been built in many sizes and shapes, as have constant-flow-velocity manifolds for providing even flow distribution circumferentially around a thrust chamber. Pressure-loss data for "T" fittings are given in reference 33.

Ducting runs are sized as much as possible for constant area, and transition requirements are minimized to minimize pressure drop. Available space frequently limits area transitions to abrupt conical or bell-mouthed transitions. References 34 and 35 contain data pertinent to diffuser design showing the necessity of small divergence angles (approximately 10° total angle) for best efficiency; however, a small angle normally requires a cumbersome transition too long for rocket engine use unless the area change is very small. However, even where it is necessary to make a short, abrupt transition in line area, the advantages of a diffuser are not to be disregarded. For very short transitions, the optimum divergence angle is quite small and even a short diffuser can greatly improve pressure recovery if properly designed. The optimum length-limited diffuser of area ratio 4.0 has a half angle of approximately 4° and is approximately 7 throat diameters in length. For purposes of standardization, 4° is preferred.

for all lengths. An  $L/d$  of 1.0 is a minimum and an  $L/d$  up to 7.0 is desirable; diffusers of larger ratios are not beneficial. Detailed design information on diffusers can be found in reference 36.

Problems associated with duct fittings other than that of meeting pressure-loss expectations are largely structural. Proper aligning of material at weld joints is a problem that generally is solved with adequate fixturing during welding. Instrumentation bosses welded to duct fittings, unless properly designed, can crack at welds or provide cleaning-residue traps that later contaminate the system. Weld or material cracking is likely to occur at a sharp change in duct wall thickness (e.g., at instrumentation bosses or mounting lugs), in hot-gas ducting, and to a lesser degree in cryogenic ducting. Load-spreading doublers or skirts can be used on heavy fittings of thin-wall ducts or vacuum jackets to avoid cracking by vibration.

One problem with duct fittings often overlooked is their effect on downstream components. Bellows failures have been traced to high-velocity fluid in skewed velocity profile coming out of an elbow immediately upstream of the bellows. Turning vanes in the elbow (sec. 2.3.2.1) have been used successfully to even out the velocity distribution and eliminate the destructive high local velocities. Flow-induced resonance of bellows (leading to failure) is treated in reference 19.

A fitting with another fitting closely coupled to it can have pressure-loss characteristics grossly different from those it exhibits by itself with long straight entrance and exit runs. This phenomenon often leads to erroneous pressure loss predictions when the individual losses of close-coupled fittings in a train are summed in the calculations. This problem is minimized by performing pressure-loss flow tests on the complete duct assembly.

### 2.3.1.3 VACUUM-JACKETED FITTINGS

Insulated lines are used extensively in rocket propulsion systems. The majority of these lines are insulated by an external covering; i.e., the lines are nonvacuum type. The attachment of external insulation has little or no influence on the detail design of the line itself or its fittings, unless support bracketry must be provided on the line to hold the insulation in place. Vacuum-jacketed lines, however, do require consideration of fittings in the design of the line.

The qualities of a successful fitting design for a vacuum-jacketed line are the same as for the other elements of the vacuum-jacketed line. These are leak tightness, adequate structural support of jacket around the pressure carrier, minimum heat-conduction paths from jacket to carrier, minimum radiation heat-transfer capabilities from jacket to carrier, and cleanliness of vacuum-side surfaces.

Leak tightness is achieved through all-welded construction with materials of high corrosion resistance, and leakage is verified by mass spectrometer leak test. Corrosion is one of the major causes of leakage in lines of all types. Cleanliness of the material surfaces is important in reducing corrosion that can cause leak paths through the walls. Configuration features that trap fluids (cleaning solvents or operational moisture condensation) are nearly always the most severely corroded. Examples of such features are doublers, lap joints, brackets, or other protrusions with an undrained cup-like shape. These features therefore are avoided whenever possible.

Reference 37 contains useful information for design of vacuum-jacketed lines and means for estimating their possible heat-leakage rates. Many specific details of successful designs are presented in this reference, but it must be noted that the ruggedness required for rocket propulsion system use was not an objective of these designs. The subject of vacuum-jacketed lines for rockets is presented in more detail in reference 19.

#### **2.3.1.4 FLOW-CONTROL DEVICES**

##### **2.3.1.4.1 Turning Vanes**

In most of the ducting for liquid propellant rocket systems, the use of turns (elbows) is unavoidable; and since the space available for the duct assembly is limited, it is not always possible to use elbows of the radius-to-diameter ratio that is optimum for minimum pressure loss. Since the geometry of these elbows has a considerable effect on the overall pressure drop of a duct assembly, it becomes necessary to utilize every known design technique to evolve the most efficient configuration. Where a sharp turning elbow is necessary, turning vanes are very effective in minimizing pressure loss. The turning or guide vanes in effect break the elbow up into smaller elbows with more favorable radius-to-diameter ratios than the entire elbow.

The pressure-loss coefficient for a mitered 90° elbow is reduced by 80 percent or more by the addition of a cascade of vanes (ref. 38). A 65 percent reduction of pressure-loss coefficient has been achieved by adding splitter-type (concentric) vanes to curved elbows (ref. 39). The subject of turning vanes is presented in more detail in reference 19.

##### **2.3.1.4.2 Flow Straighteners**

Flow straighteners have had rather limited application in rocket propulsion systems. The primary purpose of a flow straightener is to produce a uniformly distributed nonrotating flow in a duct. This type of flow sometimes is desirable in order to make an accurate pressure measurement, lower the pressure loss in fittings downstream, or evenly distribute

flow across an injector face. Most flow straighteners consist of flow-guiding surfaces arranged parallel to the longitudinal axis of the duct. Usually they have no curvature into the oncoming flow because its nature is unknown. The subject of flow straighteners is presented in detail in referene 19.

## 2.4 FIXED JOINTS

Fixed joints are nonseparable connections of fluid-system components. Since it is impractical to make line assemblies in one piece from end to end because of assembly and installation clearance problems and the insertion of components such as valves and filters, fixed joints are a necessary design element. The number of such joints is kept to a minimum because of complexity, cost, and effect on reliability. In the sections that follow, fixed joints are discussed according to joining method in the following order: welding, brazing, diffusion bonding, soldering, and interference fit.

### 2.4.1 Welded Joints

Welding is one of the common methods for joining tubular components in a propellant system. Welded joints normally exhibit mechanical properties and environmental- and propellant-compatibility characteristics similar to those of the components being joined. A welded joint can be constructed so that it contributes little or nothing to system contamination. Welded joints can be made at a subassembly stage on a bench or they can be made as final installation joints in the propellant system. The types of welded joints currently being used in fluid system lines and tubing are illustrated in figure 19. The butt, fillet, and sleeve welds (figs. 19(a)-(c)) are most common; these types are sometimes used in combination, the most frequent being fillet and sleeve (fig. 19(d)).

Butt welds are used for bench-fabricated joints and are the most desirable because of ease of radiographic inspection and lack of builtin crevices. Butt-welded tube joints were used throughout the J-2S engine, with flange couplings on the components and no in-place welding. The J-2S, a simplified version of the J-2, had fewer lines with less complicated routing and thus accommodated installation of complete line assemblies without interference. Butt-welded joints also were used on the larger lines ( $> 1$  in.) on the F-1 engine.

For in-place joints, sleeve welds appear to have the greatest acceptance because of the self-aligning feature of this joint. Also, the sleeve-weld design has proven advantageous for engine modifications in the field. Sections of tubing can be cut out of a line assembly and new components or fittings such as T's inserted through use of the in-place sleeves. The sleeve weld eliminates the need for trimming a line to an exact length during final

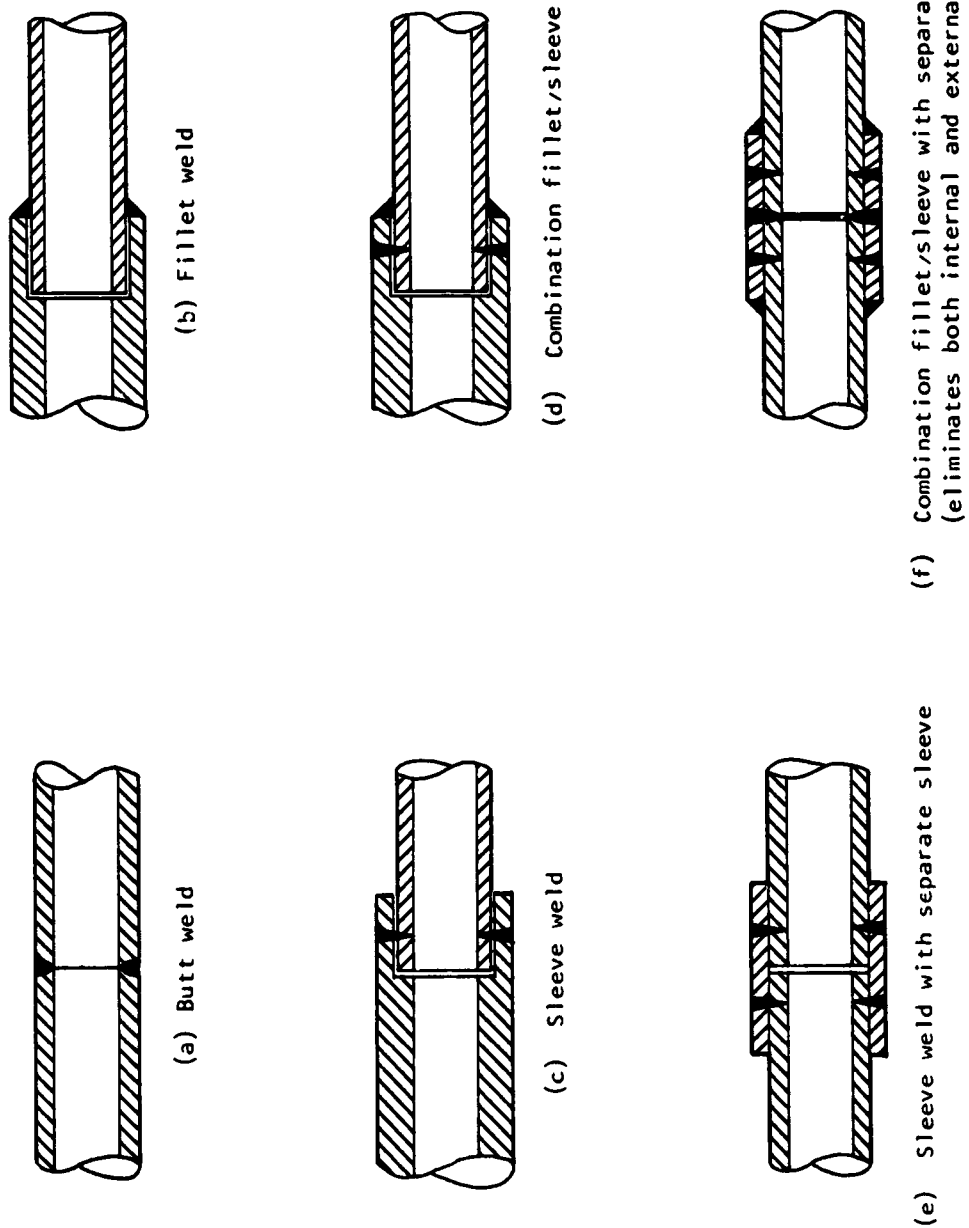


Figure 19. — Types of welded joints used in fluid systems.



installation and permits the use of a fixed joint in lieu of a coupling. Sleeve welds are used to join thin-wall tubing where the sleeve must be added to achieve the thickness for practical welding. The disadvantages of the sleeve weld are as follows:

- (1) Unless all crevices can be eliminated by design, as in figure 19(f), the resulting built-in crack will act as a stress raiser and result in lower fatigue life.
- (2) The longitudinal forces in the tube will load the weld in shear.
- (3) Unless all crevices can be eliminated by design, as in figure 19(f), the resulting crevices provide a collector for contaminants and corrosive materials.
- (4) The radiographic inspection of the weld is relatively difficult.

In the F-1 engine, the combination fillet/sleeve weld (fig. 19(d)) was used for line sizes of 1-in. diameter or less. In the J-2 engine the sleeve weld with separate sleeve (fig. 19(e)) was predominant; in some instances, the combination fillet/sleeve with separate sleeve (fig. 19(f)) was used for lines 1-in. in diameter or less. When the B-nut type of tube joint was used (for instrumentation lines), machined-sleeve tube stubs were attached to tubing assemblies with the welded-sleeve design of figure 19(e).

Weld joints are located in areas free from vibration when possible; in particular, locations with vibrations normal to the tube axis are avoided. Clearance around the exterior of the joint is provided for the in-place welding equipment. The number of joints is kept to a minimum consistent with overall system requirements. Additional information on weld joint design may be found in references 40 and 41.

#### **2.4.1.1 WELDING METHODS**

Inert-gas tungsten-arc (GTA) welding is the most widely used joining method. Electron-beam (EB) and inert-gas metal-arc (GMA) welding also are used when the unique characteristics of these welding methods are required. Resistance seam welding was employed on the Atlas and Thor engines. However, use of this method of joining currently is limited because of the close tolerance required on component parts, and because the unfused seam remaining in the completed joint causes stress risers and provides cavities for contamination. Individual components must be designed to achieve a tight or even interference fit; otherwise unwelded areas, which become leak paths, will occur in the completed joint. Even if a leak-free joint is made, the crack-like seam of the unfused area in the joint (1) is subject to both stress corrosion and crevice corrosion, (2) presents a system contamination problem because it cannot be cleaned thoroughly, and (3) is subject to fatigue failure from vibration.



The chief attributes of GTA welding are as follows:

- Can be utilized with most weld-joint designs
- Capable of joining all tubular components in a propellant system with the exception of tubes with walls less than 0.020-in. in thickness
- Adaptable to manual or automatic welding equipment
- Adaptable to shop or field use
- Capable of joining some dissimilar metal combinations with the use of proper filler wire addition
- Capable of joining components of almost any size in a propellant system (not always practical)
- Capable of joining all materials classified as weldable.

The flexibility of GTA welding is best exemplified by its use for the tube joints of both the F-1 and J-2 engines (sec. 2.4.1).

Electron-beam welding generally is used to join materials where a narrow weld bead or minimum heat input to the part is required. Because of vacuum-chamber requirements, EB welding is confined to shop operation, and the size of parts that can be joined is limited by the vacuum chamber available. Electron-beam welding can be controlled manually, but more often is automated, at least to the extent of movement of the joint with respect to the electron beam. Completely automatic welds are also made. Electron-beam welding generally is accomplished without the use of filler material, and is limited to butt or sleeve-type tubular joints. Because of the resulting desirable narrow weld bead produced by EB welding, component parts are designed with tighter tolerances than those required of a similar GTA weld joint (ref. 42). Electron-beam weld joints are designed to contain rootside weld spatter, or are designed to allow removal of weld spatter after completion of the weld.

Electron-beam welding was used on the pump-discharge ducts on the F-1 engine to attach the coupling flanges and the internal linkage tying across the bellows to the duct wall. Since these ducts were large in diameter (9-in. ID) and heavy in construction, EB welding provided a minimum heat-input method for welding that kept warpage and distortion within tolerable limits.

Inert-gas metal-arc welding is used to join heavy sections when it is desirable to minimize the number of weld passes needed to complete the joint. This weld technique is adaptable to both shop and field use. The process is also adaptable to manual or automatic welding. By

its very nature, GMA welding is always a filler-type weld. Because the process is used largely on the thick wall sections, joint design requires preparation for acceptance of filler material (e.g., a "V" or "U" groove). Joint tolerance is similar to that required for a GTA weld joint. The bolted couplings of the pump discharge ducts for the Atlas, Thor, H-1, F-1, and J-2 engines are butt welded to their respective duct assemblies with the GMA method.

#### **2.4.1.2 PROBLEM AREAS**

In-service failures with the F-1 tube sleeve-weld configuration developed in the form of fatigue cracks in the heat-affected zone (HAZ) of the weld. Cause was considered to be the high vibration environment. Fatigue failure has also been attributed to partial-penetration welds. Addition of the fillet weld to the joint for increased fatigue strength and vibration damping clamps on the tubing, located to reduce amplitudes, resolved this problem. The improved design was evaluated through extensive laboratory testing (ref. 41) and research and development engine testing.

In-service leaks in the weld area occurred with the J-2 sleeve-weld design. Examination indicated that weld oxidation porosity in the weld root and adjacent to the weld on the inside of the tubular joint permitted leakage. The solution to this problem was the careful application of backup purge gas during the welding operation. Use of a gas sniffer to sample purge gas purity during welding proved very successful. The proper application of backup purge gas to accommodate manufacturing of the part must be considered during initial design.

Crevice contamination was a contributing factor in some of the fatigue failures of weld joints on early F-1 engines. The sleeve-joint crevice harbored corrosive contaminants that led to fatigue failures in the tube heat-affected zone. The addition of fillet welds eliminated the external crevices and strengthened the joint.

In the F-1 engine program, adverse tolerance stackup on the automatic-weld joints presented processing problems. Without controlled joint dimensions, inconsistent weld quality resulted from diametrical mismatch, abnormal surfaces, or eccentric fits between a sleeve and inserted component. The allowance on dimensional variation was determined by testing sample welds that were made with production equipment. After welds that satisfied the design requirement were made, the allowable tolerances were determined.

Another design problem that contributed to weld-joint inadequacy in the F-1 engine was nonuniformity of wall thickness at the joint. This nonuniformity created poor heat distribution during welding and resulted in partial penetration of the weld. This problem was overcome by designing uniform mating walls and using special heat-sink adapters where required for the GTA method.

Cleanliness of welded surfaces is necessary for complete penetration. Removal of foreign matter (e.g., cutting oil, grease) or metal burrs and chips prior to welding is made mandatory by procedures included in the drawing specifications.

### **2.4.1.3 WELD REPAIR**

Weld repair of a discrepant joint involves either complete restoration of the component parts and rejoining by the same method used for the original joint, or a local repair. Little or no difficulty can be expected with the part-restoration-and-reweld procedure. However, a localized repair causes (1) an enlarged heat-affected zone that may not be tolerated by the joint design, (2) an increase in weld droptrough that may greatly restrict propellant flow, (3) localized distortion that may result in sufficient residual stress to affect joint integrity adversely, and (4) aggravation of a crack-sensitive joint. To avoid the problems with local repairs, the component usually is restored and rejoined.

## **2.4.2 Brazed Joints**

Brazing is a thermal process of joining metal parts by diffusion bonding with a metal filler that has a flow point over 800°F but less than the melting point of the parts to be joined. Brazing generally is used when welding is not desirable or practical or when minimum distortion or minimum residual stress is required. Brazed joints are an acceptable method for making fixed joints in many propellant systems; for example, in the Apollo spacecraft, over 770 brazed joints were used on the various fluid systems (ref. 18). The basic design requirements for brazed joints are documented in references 9 and 43.

The required brazed-joint mechanical properties and environmental- and propellant-compatibility characteristics determine the braze alloy and the method of brazing. A brazed joint can be made so that it will not contribute to system contamination. Components with considerably different wall thicknesses can be joined by brazing, an advantage over welding.

Three different brazing methods are in use: (1) furnace brazing, (2) induction brazing, and (3) dip brazing. Most materials suitable for a propellant system can be joined by either of the first two methods, which are the preferred brazing methods. To protect the metal from oxidation during the braze cycle, furnace brazing is done either in vacuum or in a hydrogen atmosphere. Similarly, in induction brazing, an argon shield is used on both the inside and the outside of the joint. Furnace and dip brazing are shop operations only; induction brazing may be either a shop or field operation.

Dip brazing is the only suitable method for making a brazed aluminum joint. The salt bath used in the dip-braze process controls oxidation of aluminum joints during brazing. The sizes of the parts being brazed are limited also by the availability of large salt baths.

A grooved sleeve or insert (one component inserted into the other) is normally used for the brazed-joint design. This design serves two purposes: (1) loads on the braze alloy are in shear, and (2) the joint is self-aligning and minimizes problems with braze alloy flow. Braze alloy can be placed either inside or on the outside of the joint. Braze alloy placed in braze grooves inside the joint is preferred, because if sufficient braze alloy has been used, the presence of braze alloy at inspection holes or a fillet on the outside of the joint can be used to judge joint integrity.

References 9, 41, and 44 provide information on braze alloys, wettability and flow information for various alloys, joint clearances, and various brazing parameters for induction brazing of tubing. Reference 41 provides applied research and development information on brazed tube joints.

Failures of brazed joints have occurred mainly as a result of inadequate cleaning (including handling) of components prior to brazing, insufficient supply of braze alloy available in the joint, and inadequate control of dimensional tolerances of component parts. Failure to achieve a good brazed joint may also result from marginal wettability of the braze alloy on the material being brazed. In some cases, this problem has been eliminated by shot peening or nickel plating the surface to be brazed. Failures due to poorly matched dimensions and inadequate braze alloy are avoided by performing dimensional studies of the joint components and of the required volume of braze alloy. Then samples of joints with extreme-tolerance parts are tested destructively to verify the design. Nondestructive tests (X-ray or ultrasonic inspection) are performed on production joints to verify joint integrity. Since an effective brazed joint cannot be obtained unless parts are adequately cleaned, the nondestructive test provides a continuous method for verifying the cleaning process.

Unlike welded joints, repair of a brazed joint usually is not a difficult operation unless the joint becomes contaminated by foreign matter. Generally, preplacement of additional braze alloy and rebrazing will repair a leaking brazed joint. If necessary, the joint can be taken apart by reheating; however, temperatures higher than the braze temperature usually are required for taking the joint apart. The increase is due to the loss of one or more elements of the alloy by diffusion into the base metal or by boiling off; either loss increases the melting point of the brazing material. This problem is solved by changing the braze alloy or by cutting out the tube section with the joint and replacing the tube section with new brazed joints.

### **2.4.3 Diffusion-Bonded Joints**

Diffusion bonding is a unique method for joining some widely differing metals by the use of controlled temperature and pressure. To date, the only widely accepted joint produced by this technique is a titanium-to-stainless-steel tubular transition section. Other material

combinations have been joined; however, they have not been used in a rocket propulsion system. The titanium-to-stainless-steel tubular transition joints were developed to provide a fixed joint between titanium propellant tanks and stainless steel lines in the MOL, Gemini, Apollo, and Surveyor programs. Materials commonly diffusion bonded are Ti-5Al-2.5Sn or Ti-6Al-4V alloys with 304L, 321, and 347 stainless steel. All titanium alloy-to-stainless steel combinations have been used successfully.

Because of limited use, diffusion-bonded transition joints have been developed by only a few suppliers, and the techniques and controls used in fabrication are proprietary. Some items involving tube-to-fitting joints have been made; however, in-line tubular sections provide the major use. Joint diameters range up to 6 in. with any wall thickness. Tubing lengths on each side of the joint are governed by the tube outer diameter; the larger the diameter, the shorter the tube length.

In use, the titanium end of the transition joint is welded to the titanium tank during tank fabrication. At a later stage of system fabrication, the stainless steel end of the transition joint is welded or brazed to the stainless steel propellant line. Therefore, while not a shop- or field-producible item itself, the diffusion-bonded transition joint does permit shop or field installation of the propellant tank. The main problem with the diffusion-bonded joint is that of locating the bond line between the two materials. However, careful inspection will reveal a slight color variation between the two materials, thereby indicating the bond line. Location of the bond line is necessary to ensure that subsequent joining operations do not exceed the temperature limitation of the joint. Diffusion bonding of dissimilar metal combinations other than titanium and stainless steel have not been developed for rocket engine propulsion systems because they provide little or no advantage over other types of joints.

#### **2.4.4 Soldered Joints**

Soldered joints have been successful in low-pressure applications. For example, approximately 150 of the more than 1500 plumbing joints on the Apollo spacecraft were soldered aluminum joints. The basic design incorporated sleeves with two grooves into which the rosin-core solder was preplaced. The aluminum tube ends were vapor honed and caustic etched prior to plating with nickel and tin. It was also necessary to preplate the inside surface of the sleeve before placing the solder. The joint was made by heating the assembly with either an induction tool or a standard resistance-heating gun.

Soldered joints are light, require simple heating tools, can be assembled in a minimum envelope, do not need inert-gas shielding, and are readily made on in-place hardware. However, solder will often flow into the flow passage of small-diameter tubing, creating a significant flow restriction. This problem was solved on the Apollo spacecraft by incorporating an interference-fit Teflon sleeve. This sleeve was placed inside the tube and

prevented solder or corrosive flux from entering the line. Solder has low creep strength and low efficiency for carrying axial loads. Preloading a soldered joint by tightening adjacent B-nuts after soldering was avoided on the Apollo spacecraft by tightening all nuts before a joint was soldered.

## **2.4.5 Interference-Fit Joints**

Interference-fit joints without the use of some other joining method are not used in propellant systems but are used occasionally in pneumatic and hydraulic systems. This type of joint is attractive because of the apparent cost saving over other fixed joints. However, attempts at fabricating a leak-free interference-fit joint for propellant systems have generally met with failure.

## **2.5 SEALS**

Two types of seals are discussed herein: static seals, and rotating and sliding dynamic seals. High-speed-shaft seals required in turbomachinery, valve closures that must move off and onto a seal, or hermetic seals such as diaphragms and bellows are treated in references 1, 6, and 45. The basic design principles for static and dynamic seals are documented in references 9 and 10.

Static seals are utilized chiefly at the interfaces of interconnected engine components; they are also used at the interfaces of detail parts or subassemblies within a large, major engine assembly such as a turbopump or a thrust chamber. Dynamic seals, as defined herein, are used at the moving interfaces of mechanical components such as reciprocating and rotating shafts and piston-cylinder applications.

A designer, faced with selection of a static or dynamic seal for a given application, must first establish the design requirements, viz., seal size, weight limits, fluid to be sealed, operational pressure and temperature, thermal gradients, permissible leakage rates, duty cycle, operational loads imposed by vibration, pressure surges, and seal life. Once the requirements are established, he screens the various seal concepts available for those that may be capable of meeting the requirements or he may even evolve a new concept. Tradeoff studies on performance, cost, availability, maintainability, and other factors are then made to select the optimum design.

### **2.5.1 Static-Seal Configuration**

Static seals are devices used to limit or control leakage of fluid in a separable connector in which there is no relative motion of the mating surfaces other than that induced by changes



in the operating environment. The present basis for successful static-seal design is a complete analysis of the leakage requirement and the integration of the seal into the coupling. This analysis takes into consideration the fluid, environments, loads, and the seal requirements imposed on the coupling design. Sealing is accomplished by achieving and maintaining a continuous barrier against the transfer of fluid between the seal element and the mating coupling surfaces. In general, this fluid barrier occurs when the plastic flow at the sealing interface is sufficient to fill all the voids; the seal is effective as long as this interface is not disturbed. An effective sealing barrier can also be achieved without a gasket between metal flanges with very flat and very smooth surfaces (ref. 46). Sealing is accomplished within the elastic limits of the flange material; however, the surface finishes required are difficult to obtain and are very sensitive to damage.

The wide variety of static-seal configurations that has been used in rocket engines and vehicles can be grouped into five basic descriptive categories: gaskets, pressure-assisted seals, plastic spring-loaded seals, radial or toggle seals, and metallic boss seals (table I\*). The usefulness of any seal configuration is enhanced considerably by the kind of coating or plating applied to it; a summary of basic characteristics of seal coatings and platings is presented in table II\*.

The gasket types were adapted from commercial applications and used in the earliest rocket engine designs. Though unsophisticated and requiring brute force by means of bolts and flanges to maintain a seal, they served the purpose of restricting leakage sufficiently to permit concentration on the development of major engine components. All-metal gaskets were used for cryogenic and high-temperature applications, while elastomeric O-rings, developed during World War II, were used in limited-temperature-range applications.

The pressure-assisted types, although having some commercial heritage, were basically a product of the space age. By their nature, they did not require the brute-force axial loads to effect a seal; thus lightweight flanges could deflect under pressure but would not leak because the pressure-assisted seal was able to follow the deflection. This type of seal is predominant in high-pressure applications in current propulsion systems.

The plastic spring-loaded seals employed fluorocarbon plastics (e.g., Teflon, Mylar, and Kel-F) that were suitable for use at cryogenic temperature to act as a soft, conforming interface with the hard-surface metallic flanges. Cold flow of the plastic into flange asperities was produced by use of an integral metallic spring. This type of seal when used at cryogenic temperature is limited to low-pressure (up to 500 psi) operation because of the low hoop strength of the plastic materials at cryogenic temperature.

The radial or toggle seals effect a seal by imposing localized, high-unit-pressure loading against the mating flanges. The sealing tip of the seal actually is yielded by the harder surface of the mating flange. Although these seals require a higher axial load than the pressure-assisted seals, they also have good ability to follow flange deflection.

\* Table I appears on pages 46 and 47. Table II appears on page 48.



The metallic boss seals were developed for use on bosses for applications beyond the temperature limits of elastomers. These seals generally are spring- or pressure-assisted types with a soft plating for conformance to the mating contact surfaces. They are used extensively on temperature- and pressure-instrumentation bosses and at boss couplings for connecting small-diameter tubing to components.

The factors involved in the selection of the seal configuration for a given application are the operational pressure and temperature, fluid/material compatibility, loading, flange requirements, leakage requirements, and cost. Elastomeric O-rings or molded-in-place seals are the first choice for static-seal applications within their compatibility and environmental limitations. Various static-seal configurations for applications beyond the elastomer limitations are presented in table I. The nonelastomeric seals are either all metal or a combination of metals and nonmetals, (e.g., Teflon coating on stainless steel). These seal configurations can seal cryogenic fluids, hot gases, and fluids that are not compatible with elastomers. The use of the various seal configurations on large-diameter applications is limited only by the tooling and machining capability to fabricate the seals and related flange surfaces.

## 2.5.1.1 GASKETS

### 2.5.1.1.1 Elastomeric O-Rings

Elastomeric O-rings are made from natural or synthetic rubber and are used extensively in components, between components, and on standard bosses. Standard installations include face seal, circumferential seal, and boss seal (fig. 20). The O-ring installations generally are controlled by the requirements of the O-ring gland design specification (ref. 48) and the

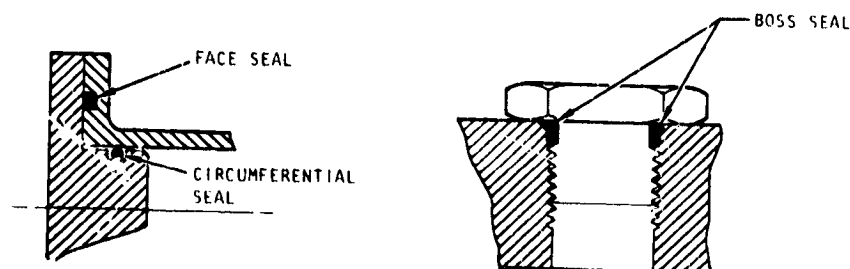
























Figure 20. — Three types of O-ring static seals.

Table I. — Chief Design Features of Commonly Used Static Seals

SEAL TYPE	CONFIGURATION OR TRADE NAME	MANUFACTURER	MATERIAL	COATING	GLAND DESIGN REQUIREMENTS						PRESSURE LIMIT WITHOUT SPECIAL GLAND OR SEAL DESIGN, PSI
GASKET	O-RING 	VARIOUS	ELASTOMERS AND PLASTICS	NONE	SURFACE FINISH, $\mu$ IN.	FLATNESS, IN. IN.	ALLOWABLE SEPARATION, IN.	UNIT SEALING LOAD, LBF IN.	SEAL COMPRESSION CONTROL REQUIRED	YES	1500
	MOLDED SHAPES 	PARKER SEAL CO STILLMAN RUBBER CO	ELASTOMER MOLDED TO METAL BASE	NONE	32 to 64	0.003	0.003	30 to 50	NO		5000
	HOLLOW O-RING 	UNITED AIRCRAFT PRODUCTS, INC ADVANCED PRODUCTS DSD MFG CO	VARIOUS METALS	TABLE II	8 to 32	0.0007	1/2	1/2	YES		5000
	SPIRAL WOUND 	JOHNS MANVILLE FLEXITALLIC GASKET CO GARLOCK, INC	VARIOUS METALS AND FILLERS	NONE	64 to 125	0.0007	0.0005	1/2	YES		100
	FLAT 	VARIOUS	ELASTOMERS AND PLASTICS	NONE	32 to 64	0.010	0.010	1/2	1/2		100
	FLAT PLAIN AND GROOVED 	VARIOUS	VARIOUS METALS	NONE	8 to 32	0.001	NONE	1/2	1/2		100
	SOLID 	VARIOUS	VARIOUS METALS	NONE	32	0.001	NONE	1/2	1/2		15,000
	SEAL SPACER TYPE AND GROOVED TYPE 	VARIOUS	INCONEL 718	TABLE II	32 (CIRCULAR LAY)	0.0006	0.008	200 to 300	SPACER TYPE NO GROOVED TYPE YES		10,000
	K SEAL 	HARRISON MFG CO	CRES	TABLE II	8 to 32	0.0007	0.002	30 to 60	NO		3000
	HASKEL 	HASKEL ENGRG	CRES	TABLE II	2 to 32	0.0005	0.002	20 to 100	NO		100
METALLIC PRESSURE ASSISTED											

METALLIC PRESSURE ASSISTED	HYDROLYNE	DONALDSON CO	VARIOUS METALS	TABLE II	8	0.0002	0.0002	70	YES	1000
SPRING LOADED		PARKER SEAL CO HI TEMP RINGS TETRAFLUOR INC	VARIOUS METALS	TABLE II	16 to 32	0.0005	0.008	100 to 250	YES	(b)
		PARKER SEAL CO	INCONEL 718	TABLE II	32 to 64	0.0005	0.008	300 to 400	YES	(b)
		KOPFERS CO NAVAN PRODUCTS	CRES	NONE	8 to 32	0.0015	0.007	220 to 550	YES	4000
		AERODUP CORP	TEFLON JACKET OVER FLAT STAINLESS HELICAL SPRING	NOT APPLICABLE	63	0.005	0.020	50 to 100	YES	1200
RADIAL OR TOGGLE		RACO MFG CO	TEFLON JACKET OVER STAINLESS FINGER SPRING	NOT APPLICABLE	32	0.005	0.015	50 to 100	YES	1200
		HASKEL ENGRG	TEFLON TUBE OVER STAINLESS STEEL COIL SPRING	NOT APPLICABLE	32	0.005	0.010	50 to 100	YES	1500
		VARIOUS	CRES	NICKEL	32	0.020	0.005	500	YES	(b)
		NATIONAL UTILITIES	CRES OR ALUMINUM ALLOY	NONE	64	0.005	0.005	20 to 80	YES	(b)
METALLIC ROSS		AERODUP CORP	CRES OR ALUMINUM ALLOY	NONE	32	0.020	0.020	500 to 600	YES	(b)
		GAMAH CORP	CRES OR ALUMINUM ALLOY	NONE	32	0.010	0.005	500	YES	(b)
		HARRISON MFG CO	CRES	TABLE II	16 to 32	0.001	0.0005	(a)	NO	(b)
		NAVAN PRODUCTS	CRES	SILVER AND NICKEL	32	0.001	0.0005	1000	NO	(b)

(a) TOTAL LOAD APPLIED SHOULD NOT CRUSH GASKET

(b) PRESSURE LIMIT DEPENDS ON SEAL DIAMETER AND CROSS SECTION (REFER TO MANUFACTURERS' RECOMMENDATIONS)

(c) DEPENDENT ON BASE MATERIAL AND SEAL CROSS-SECTIONAL CONFIGURATION (SEE TABLES 6.3.2 b, c, d and 6.3.3.4c IN REF 9)

Table II. — Basic Characteristics of Materials Used as Platings or Coatings on Static Seals (adptd. from ref. 47)

Status	Plating or Coating	Useful Temperature Range, °F	Hardness		Remarks
			Brinell or as noted	Rockwell	
Operational	Copper	-452 to +1900	Vickers 41 to 220	< 97 R <sub>B</sub>	Suitable for vacuum applications; resistant to fluorine and certain other corrosive chemicals.
	Gold	-452 to +1850	65 to 125	26 to 77 R <sub>B</sub>	Similar to silver but somewhat better resistance to certain corrosive fluids; very wide temperature range but higher in cost than silver.
	Kel-F	-452 to +300	Rockwell R 80 to 85	80 to 85 R <sub>R</sub>	Similar to Teflon, but more resilient and plastic at low temperatures; generally higher in cost than Teflon.
	*Rhodium	-452 to +3500	400 to 800	43 to 72 R <sub>C</sub>	Similar to silver, but useful at much higher temperatures; higher in cost than either silver or gold.
	Silver	-452 to +1650	50 to 150	88 R <sub>B</sub>	Excellent general purpose plating for high-temperature use, but generally less suitable for cryogenic temperatures than gold or Teflon; excellent chemical and radiation resistance.
	Teflon (FEP)	-452 to +400	Shore D 59	55 R <sub>D</sub>	Similar to Teflon (TFE), but somewhat softer and denser; useful high-temperature limit lower than that of Teflon (TFE).
	Teflon (TFE)	-452 to +500	Shore D 52	75 to 95 R <sub>J</sub>	Excellent coating for applications up to +500°F; excellent chemical resistance; particularly suitable for cryogenic applications.
Research and Development	Aluminum	-452 to +900	Vickers 30 to 90	< 56 R <sub>B</sub>	Compatible with most oxidizers and fuels, and particularly suitable for use with liquid and gaseous fluorine; however, it is very difficult (and costly) to obtain quality of plating required.
	Indium	-452 to +300	Soft	NC*	Very soft plating, limited to moderate temperatures; suitable for cryogenic applications.
	Lead	-65 to +450	5	NC*	Very soft plating with limited temperature range; excellent radiation resistance.
	Nickel (soft)	-452 to +2500	140 to 200	76 to 93 R <sub>B</sub>	Withstands high temperatures but slightly softer and less ductile than other platings.
	Platinum	-452 to +3100	Knoop 280 to 290	NC*	Withstands higher temperatures than other platings; normally limited to use with ultra-high-temperature base metals such as TZM.
	Tin (pure)	-32 to +350	5	NC*	Very ductile, but very limited temperature range; used only with a few corrosive chemicals.

\*Used primarily as flash coating to prevent sticking

NC = no correlation to Rockwell scales.

TZM = molybdenum alloy containing 0.5Ti-0.08Zr-0.3C.

handbooks of O-ring suppliers (e.g., refs. 49 and 50). O-rings are used over a temperature range of  $-80^{\circ}\text{F}$  to  $+450^{\circ}\text{F}$  for long periods of time, and at higher temperatures for shorter periods of time; useful lifetimes at various temperatures are given in references 49 and 50. The initial seal is accomplished by compressing the elastomer to its installed position. The system pressure then compresses the elastomer so that it conforms completely to the flanges at the leak path. O-rings can be used in a variety of configurations. The flange surfaces that contact the O-ring are uncomplicated, with reasonably toleranced grooves that confine the O-ring in a compact area. O-rings seal over a wide range of pressures (limited only by supporting structure) and provide high reliability at nominal expense. Problems associated with O-rings include damage to the O-ring during installation, improper squeeze, compression set, and O-ring extrusion; leakage can result from each condition.

Forcing an O-ring over sharp corners or threads can result in pinching or, if the force is of sufficient magnitude, cutting the O-ring. This damage can result in leakage and can also contaminate the system. The pinching condition is more severe if there is excessive O-ring squeeze due to improper gland depth. However, if the squeeze is too low, leakage can result, because the load is lower than that required to establish an initial seal. These conditions are prevented by the use of proper hardware design to eliminate pinching and use of a squeeze range that has been proven successful. Figure 21 shows two common conditions that can

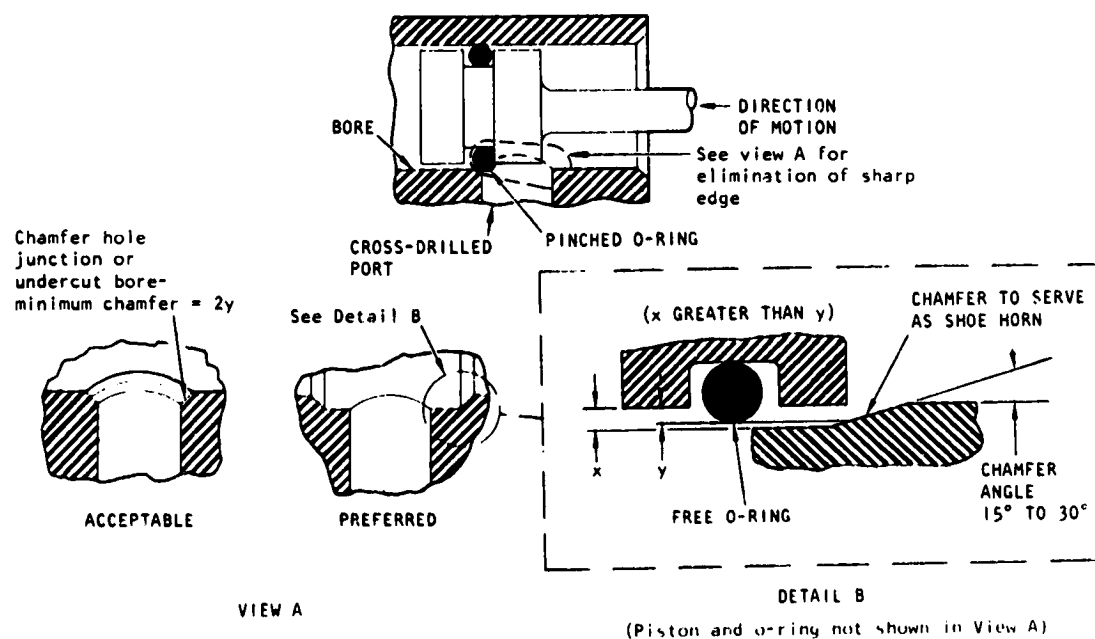


Figure 21. — Gland chamfer techniques to prevent O-ring damage.

cause pinching and the design approaches that have eliminated the problem. The squeeze required to prevent damage while still preventing leakage ranges from 8 to 32%, depending on the O-ring cross-sectional diameter, type of application (face or radial), and the coupling tolerances. The squeeze also is influenced by the change in O-ring cross section due to temperature changes and installation stretch (ref. 49). Changes in O-ring cross section are considered when sizing the O-ring grooves for the squeeze required to prevent leakage. Squeeze requirements for standard O-ring sizes, installation configurations, and operating conditions are included in references 48 through 51.

O-ring extrusion and subsequent nibbling of the elastomer occur when the clearance between the retaining members permits the elastomer to be forced into the clearance; and if the elastomer becomes trapped, it is nibbled or sheared off when the pressure decays. Extrusion is prevented by proper combinations of clearances and relatively hard elastomers. Figure 22 (ref. 49) illustrates the clearance, pressure, and hardness relations used to avoid

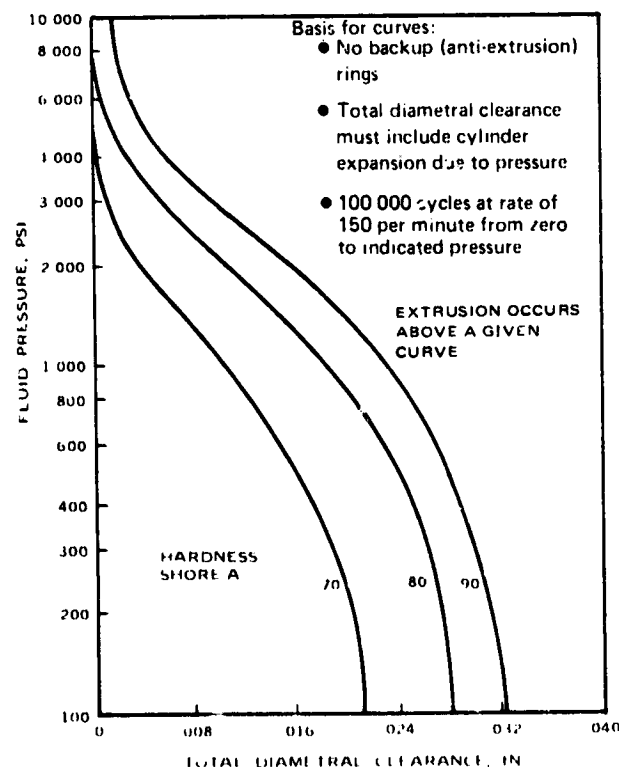


Figure 22. — O-ring extrusion related to diametral clearance, fluid pressure, and O-ring hardness (ref. 49).

extrusion in successful designs. Cyclical deflections during operation and loss of elastomer hardness at high temperature are also factors that must be used in conjunction with the relations shown in figure 22. In some instances, backup rings are used to reduce the clearance in circumferential O-ring installations. To minimize the problem of installing the backup ring on the wrong side, backup rings are installed on each side of the O-ring. Clearances due to operational deflections are reduced by using more rigid structures (e.g., increased wall or flange thickness or increased bolting to reduce coupling deflections). Another method for reducing clearance on extreme-pressure applications ( $> 10\,000$  psi) is the use of a clearance-compensating device such as that discussed in reference 52.

#### 2.5.1.1.2 Molded-In-Place Seals

Molded-in-place seals generally are flat plates with elastomeric inserts molded into machined grooves (fig. 23). These seals function like O-rings, with the initial seal being made by

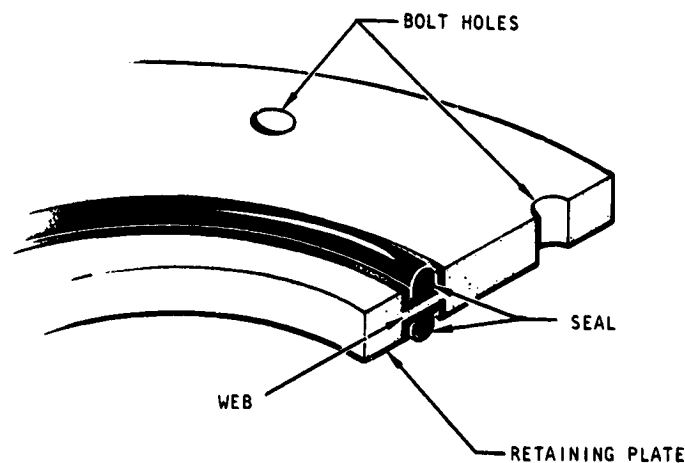


Figure 23. — Molded-in-place seal configuration.

compression of the elastomer and system pressure forcing the elastomer against the flange at the leak path; the seals are used in applications where the designer needs freedom from restrictions imposed by the circular O-ring. The retaining plate provides both the cavity for positioning the molded-in-place elastomer with respect to the flanges and the structural backup to prevent the seal from blowing out of the joint. The plate is part of the elastomer mold and is subjected to high temperatures and high forces while the elastomer is molded.



This condition adds some design requirements to the seal plate (e.g., the distance from the elastomer edge to the bolt holes must be sufficient to enable the plate to withstand forces imposed during molding).

The advantages of the in-place seal are as follows:

- The thickness of the metal plate controls the amount of squeeze on the elastomeric portions of the seal; this condition allows the use of flat-faced flanges on both sides of the seal, thus obviating the need for machining an O-ring groove in one of the flanges with its attendant increased dimensional tolerance buildup and cost.
- The in-place seal permits easy installation in large, heavy, and cumbersome hardware, since it can be installed laterally between two flanges with only enough clearance to slip the seal in and out. This feature also makes seal and component replacement easier in the field.

The in-place seal was used successfully on the fuel system on the F-1 engine. The various configurations of in-place seals used on the F-1 include dual-elastomer inserts for leakage monitoring, noncircular elastomer inserts, in-place seals with integral orifices, and multipressurized areas. In one instance in the F-1 engine program, the web between the two elastomer inserts (fig. 23) buckled because of inadequate web thickness. At engine cutoff, the very rapid decrease in system pressure caused the flanges to be released from their deflected position, and the fluid was trapped in the void. The trapped fluid pressure was high enough to shear the web. Web weakness is avoided by conducting a stress analysis (at operating pressure), and then specifying the plate thickness and elastomer groove depth that ensure an adequate web thickness.

Problems associated with molded-in-place seals include extrusion on high-pressure systems, thinning in branch and overlapping seals, and structural failures of the plate. Extrusion and subsequent nibbling of the elastomer insert is similar to that which occurs with O-rings. This damage is eliminated by increasing the coupling rigidity or reducing the rate of pressure decay.

The branch and overlapping types of seals shown in figure 24 allow additional design possibilities in a single-seal unit that can save space and weight, and can permit a design that otherwise may not be accomplished. However, leaks have occurred because the system pressure was higher than the elastomer's supporting capability. This capability is limited in this application (in comparison with a single-pressure zone) because of the tendency for the elastomer to move around the circumference of the groove and not in the normal radial direction (fig. 24). The leakage can be eliminated only by reducing the system pressure to less than 150 psi or by eliminating the branch or overlap seal concept.

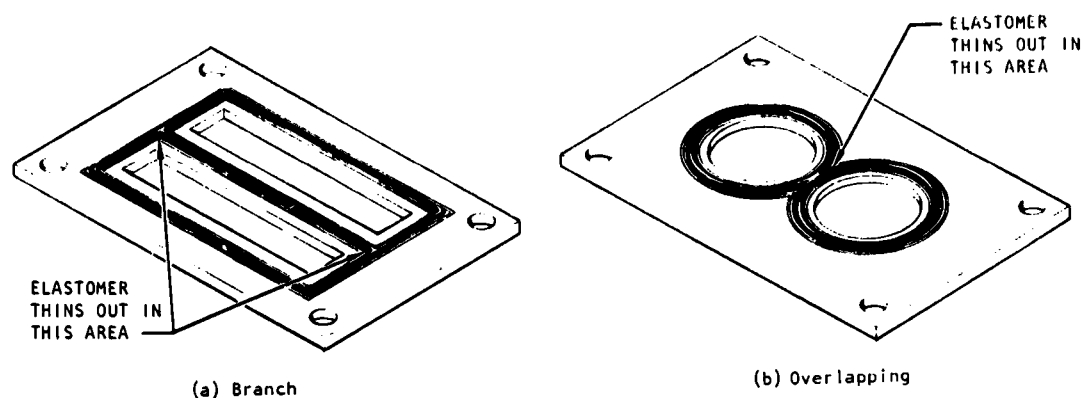


Figure 24. — Thinning of branch and overlapping seals.

### 2.5.1.1.3 Metallic Gaskets

A variety of metallic seals of the "brute-force" species can be used at both cryogenic and high-temperature conditions. In general, these seals require high seating loads and have very little capability for following flange deflection.

Metal O-rings are used in some special couplings in rocket propulsion systems, particularly where the flanges or mating surface are very rigid and are connected with ample bolting. This type of seal will perform satisfactorily under these conditions and offers the advantage of a small cross section for use where space is limited. They can be Teflon coated for use in cryogenic service or soft-metal plated for high-temperature or cryogenic usage.

Spiral-wound gaskets consist of a V-shaped ribbon of stainless steel wrapped spirally with a soft filler of either asbestos or Teflon between the turns (fig. 25). This type of seal was used extensively in early engine designs for both cryogenic and high-temperature (up to 1000°F) applications. Its chief drawbacks were the need for heavy, rigid flanges and high bolt loads (on the order of 3000 to 4000 lbf/circumferential inch). The high unit loads of the edges of the steel ribbon caused marring of the mating flanges and reduced the possibility of achieving a good seal when the joint was reassembled. This kind of gasket is still used today where leakage is tolerable, usually in isolated engine locations and in large diameters where they have a cost advantage over more sophisticated machined seals.

The other metallic gaskets shown in table I require extremely heavy flanges and high axial loading to effect a seal. As a result, they have found only limited use in rocket engines.

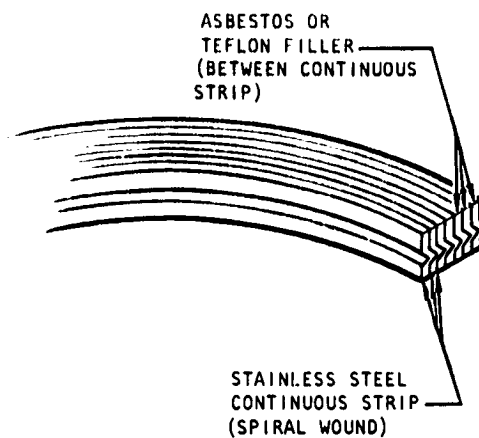


Figure 25. — Spiral-wound gasket.

### 2.5.1.2 PRESSURE-ASSISTED SEALS

Pressure-assisted seals were developed for cryogenic systems to replace the troublesome spiral-wound gaskets and crush gaskets used on the Thor and Jupiter engines. Later, as a result of success, the design concept was modified to seal hot gases. This type of seal was used extensively on the J-2 and F-1 engines. The media sealed were cryogenic fluids at temperatures as low as  $-423^{\circ}\text{F}$  and pressures as high as 4000 psi, and hot gases up to  $1400^{\circ}\text{F}$  and pressures up to 1500 psi. These seals can seal effectively under less severe conditions, but they are more troublesome and more expensive than elastomeric seals. Therefore, they are not used when an elastomeric seal can meet the requirements.

The seal cross sections for the different types of pressure-assisted seals range from a "C" shape where the cross section is round like an O-ring to a "U" shape where the seal has two open legs that act as springs to load the seal; the sections are open to the fluid pressure (table 1). All these seals are spring loaded to effect a low-pressure seal; subsequent application of fluid pressure supplies an additional sealing force that compensates for the higher pressure trying to pass the seal. Seal deflections are limited by a built-in spacer plate or by installation of the seal in a controlled groove. Several seal configurations rely on the groove to provide additional radial strength in high-pressure applications; the additional strength permits the use of a small seal envelope. Most of the pressure-assisted seals are metal with soft platings or plastic coatings on the sealing surfaces. Figure 18 shows typical installations of pressure-assisted seals.

The problems with pressure-assisted seals usually are related to (1) obtaining the required seal load at the flange contact surface and (2) providing seal resilience for following flange

deflections. The seal contact load is related directly to the leg thickness and the amount of deflection and generally is selected to provide the required contact pressure without marring the contact surface. A plastic coating or metal plating on the seal that will flow plastically at a relatively low load can protect the contact surface. The resultant low-load seal is more resilient than a seal with a high contact load for a given envelope. Resilience also is increased by the use of a tapered leg, which is more flexible than a straight leg. On metal-plated seals, the plating is machined or buffed to provide a flat, smooth surface to improve conformance of the seal to the contact surface. If the seal configuration does not provide sufficient resilience, then the mating contact surface is made rigid; however, increased rigidity results in a heavier coupling.

Cryogenic seals generally are Teflon coated, but metal-plated seals also are used successfully. In some applications such as pressurization-line couplings, temperature ranges cover both cryogenic and high temperature; for these applications, metal platings are used.

Metallic pressure-assisted seals are used extensively on the SSME. The fluids sealed are cryogenic fluids at temperatures as low as  $-423^{\circ}\text{F}$  and pressures to 9000 psi, and hot gases to  $1800^{\circ}\text{F}$  and pressures to 6200 psi. The seals are constructed of Inconel-718 base material plated with silver or gold to provide a soft sealing material at the interface. Plastic coating could not be used on the sealing surfaces because the high pressures generated seal loads sufficiently high to cut through plastic sealing material. A groove-type rather than a spacer-type seal (fig. 18) is used to save weight and provide a smaller envelope. This SSME metallic pressure-assisted seal (fig. 16) has been tested up to 18 000 psi at  $-300^{\circ}\text{F}$ .

Reference 53 presents a structural analysis of metal pressure-assisted seals, while references 12 and 54 contain nomographs for use in design of pressure-assisted seals.

### 2.5.1.3 PLASTIC SPRING-LOADED SEALS

Plastic spring-loaded seals were developed for liquid-oxygen service at approximately the same time and for the same reason that the metallic pressure-assisted seals were being developed. The basic construction consists of a plastic (usually Teflon) jacket over a metallic-spring core (table I). The spring provides both the force required to effect the initial seal at low pressure and the capability to compensate for dimensional changes resulting from tolerance accumulations, thermal expansions and contractions, and cold flow of the plastic. These seals are pressure assisted, and thus the sealing load increases as the pressure is increased.

The plastic spring-loaded seal has no operational advantages over the metallic pressure-actuated seal and at cryogenic temperatures has several disadvantages that lower

seal reliability. Therefore, usage on rocket propulsion systems has been confined largely to limited-temperature-range applications where elastomers are not compatible with the fluid (e.g., in storable-propellant systems).

Problems with these seals include fracture of the Teflon jacket at cryogenic temperatures, cold flow of the jacket at high temperature, insufficient spring loading, and installation difficulties with radial seals.

Fracture of the jacket occurs at cryogenic temperatures under pressurized conditions. The Teflon is very brittle at the low temperatures and has a rate of contraction higher than that of a metal; the seal contraction increases the clearance between the seal and the housing and, as the pressure increases, the seal fractures before it obtains adequate support from the housing. Fracture can also occur from sudden application of pressure. The combination of these problems and the requirement for high reliability has resulted in limited use of this type of seal in cryogenic applications; however, they are suitable for use in low-pressure, no-shock applications such as tank manhole covers.

Adequate spring loading is achieved by use of a spring that provides the required load in conjunction with a smooth surface finish (e.g., 32  $\mu$ in.).

Teflon cold flow at high temperatures ( $\geq 500^\circ\text{F}$ ) is avoided by restricting the use of this type of seal to applications with an upper temperature limit of  $300^\circ\text{F}$ .

Installation problems with circumferential seals arise chiefly from stretching of the seal during assembly. Stretching can damage both the Teflon and spring. A split housing generally is used to eliminate the need to stretch the seal (ref. 9).

#### 2.5.1.4 RADIAL OR TOGGLE SEALS

The radial seal concept (fig. 26) is based on converting an axial load to a radial interface load through a toggle action within the seal structure. The coupling flanges are designed to confine the outer diameter of the seal such that radial interference occurs and plastic flow of the seal takes place when an axial load is applied during installation. Because of the plastic flow, the seal is not reusable; however, the flanges are reusable with new seals. The axial load required to install this type of seal falls within the range of 500 to 600 lbf/circ. in. Various configurations of radial or toggle seals are shown in table I.

One type of radial seal, the Conoseal, has been used on a number of engines and vehicles (e.g., the M-1 engine, Titan engines, and SIV and SIVB vehicles) to seal cryogenic propellants, storable propellants, and hot-gas fluids. Coupling sizes have ranged from 1.8-in. tube connectors to an 87-in. coupling that connected a nuclear engine (NERVA) exhaust duct to an engine test stand. The other seal types (Nuco, Bobbin, Gamah) shown in table I

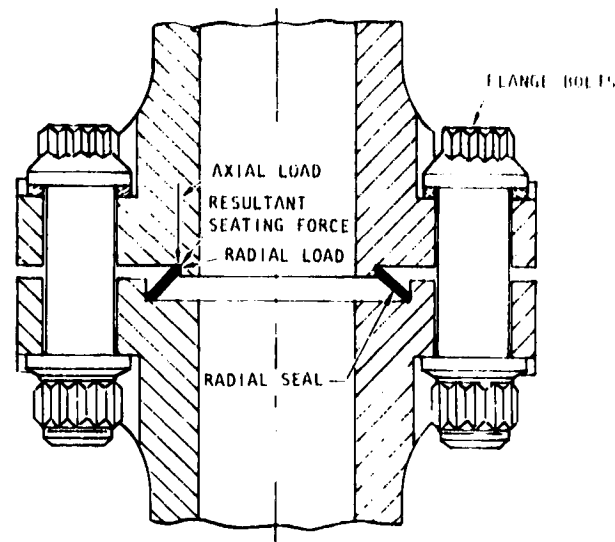


Figure 26. — Radial or toggle seal.

have had limited usage up to now. The Bobbin and Gamah seals were developed for tube couplings; however, because these seals have been developed recently, they have had few applications on rocket engines. Most of the radial seals have been developed by seal manufacturers who control the seal and the mating contact area.

Problems with the radial seal involve indenting and marring of mating coupling flanges, installation misalignment, and material thermal changes. Leakage has also resulted from improper surface finish or coupling contact configuration (i.e., tool chatter marks, scratches, and improper corner radii). For good sealing, a surface finish of  $32 \mu\text{in.}$  without chatter marks or imperfections usually is specified. Fabrication and inspection controls are used to verify that the surface conditions and required flange dimensions are met. Teflon coating or soft-metal platings provide improved seal-flange contact, which improves sealing. With Conoseals, marring of the flanges is minimized by the use of a seal of the same material or of a softer material; e.g., aluminum seals in aluminum flanges, or 300-series-steel seals with 321 CRES or nickel-base alloy flanges.

Excessive seal coupling misalignment occurred on large diameter ( $> 9 \text{ in.}$ ) Conoseal couplings on the SIV, the result being that the seal was forced out of the normal position during coupling assembly. This problem was resolved by utilizing a press-fit ring to retain the seal in the proper position and by using alignment tools during the coupling assembly.

operation. If flanges with different coefficients of expansion are used in a coupling, the seal installation is designed so that the seal load is increased under the extreme thermal condition. For example, on an aluminum-steel coupling combination used in cryogenic service, the aluminum flange is placed on the outer surface of the seal so that it will shrink down tight on the seal at cryogenic temperatures.

The RL10 engine program has accumulated substantial time on experimental engines that incorporated Teflon-coated-aluminum angle gasket in a 347-steel mating flange between the injector and combustion chamber, sealing 395 psi on a 1.2-in. diameter. Similar seals are used in hard plumbing lines sealing higher pressure (up to 1000 psi) but on smaller diameter connections.

### **2.5.1.5 METALLIC BOSS SEALS**

Metallic seals for bosses are used on threaded ports when elastomers are not compatible with the fluid or are not capable of sealing because of extreme temperatures. The boss seals used on early rocket engines at extreme environments were elastomeric O-rings on cryogenic applications and either metal-jacketed asbestos gaskets or copper crush gaskets on hot-gas applications. Leakage at both environmental extremes led seal suppliers to develop cryogenic and hot-gas boss seals that could be used with existing bosses and fittings. Two of the boss-seal types, the K-seal and Natorq seal (table D), are used extensively and successfully to seal both cryogenic and hot-gas fluids on the F-1 and J-2S engines. These seals have a soft metal plating or Teflon coating to achieve seal flange conformance. Figure 17 shows typical instrumentation bosses with seals and fittings installed. The configuration that is used on both the J-2 and J-2S engines—a spot-faced, tapped-hole boss with a pressure-assisted seal—is shown in figure 17(c). This configuration was used to provide leakage-monitoring capability.

Problems with boss assemblies include seal misalignment and boss damage. Leakage has been caused by misaligned seal loading resulting from inadequate thread engagement or bosses or fittings having threads and sealing surfaces out-of-square. These conditions are avoided by designing fittings with adequate thread engagement and by imposing drawing controls on thread-sealing-surface squareness on all coupling parts.

Boss seal configurations that require high bearing pressure can result in damage to the boss. The damage is more severe if the boss is made of soft material such as aluminum or copper. This damage is minimized by the use of a seal configuration that requires low bearing pressure.



## 2.5.2 Static-Seal Design Integration

The various design requirements that are applicable to the operation of all static seals include loads, contact surfaces, environments, and seal resilience. Additional considerations include requirements for seal barriers, leakage control, and leakage monitoring.

### 2.5.2.1 LOADS

Loads on static seals consist of mechanical loads (installation and operational), fluid pressure loads, and thermally-induced loads resulting from environmental temperatures (fluid and component). An effective static seal is capable of withstanding these loads and of transmitting the loads to the contacting structure.

The seal contact load is maintained at a level that will limit leakage under all operating conditions. The initial load is dictated by the installation requirements; however, this load is supplemented on some designs (e.g., elastomeric seals and pressure-assisted seals) by the system pressure and on other seals (e.g., radial seals) by thermal changes. The load requirement on some metal seal configurations is minimized by the use of plastic coatings or soft metal platings that flow plastically to conform to the mating-surface topography.

Low installation loads have resulted in low-pressure leakage, while high installation or operational loads due to pressure or to thermal changes have resulted in marring of the seal contact surfaces. Inadequate loading during operation has been attributed to coupling deflections, thermal changes within the coupling assembly, and yielding of the seal materials. Low installation loads have been resolved by increasing the squeeze on elastomeric seals, by increasing the spring load on pressure-assisted seals, and by increasing the radial force on radial seals. Thermal problems are resolved by using high-temperature materials or thermal barriers, or by changing the structure to withstand high temperatures without excessive deflections or yielding. Surface marring is prevented by reducing the contact load or by using a seal contact material with a higher bearing strength.

Table I shows installation loads for a number of different seals and the allowable coupling separation. The allowable separation defines the upper limit where an adequate load is still present to effect a seal.

Studies on the principles of sealing loads and leakage are included in references 26, 54, and 55.

### 2.5.2.2 CONTACT SURFACES

The seal contact surfaces on couplings provide the special features (flat surfaces, grooves, controlled diameters) and the surface topography required by the static seal.

Contact-surface information is contained in the literature of the various seal suppliers and is discussed in more detail in section 2.2.2.1. Metal seals generally have surface finishes comparable to the required flange surface finish or smoother. Smooth finishes on the seals minimize the load required to achieve complete contact with the mating surface.

Problems encountered with contacting surfaces include the inability of a metal seal to follow out-of-flat and nonrigid flanges under various operating conditions, and marring of contacting flanges by high seal-flange loading. Solutions to these problems are included in sections 2.2.2.1, 2.2.2.2, and 2.5.2.1.

### **2.5.2.3 ENVIRONMENT**

Two environmental conditions, temperature and the nature of the fluid medium, impose special considerations on static seals. Static seals in rocket engines are required to seal from  $-423^{\circ}\text{F}$  to  $+1450^{\circ}\text{F}$ , and the fluids sealed include cryogenic propellants, storable propellants, combustion products, and solvents used during manufacturing and engine checkout.

Temperature influences the operation of seals by causing changes to the physical properties of seal materials. Elastomers and plastics are affected by both high and low temperatures with respect to changes in strength, flexibility, and hardness. The change in elastomer hardness at low temperatures results in the elastomer losing its ability to seal effectively when pressure is applied. For high temperatures, metal seals with soft interfaces are used; the metal used as interface material is selected for its resilience at operating temperature, and the metal for the base material is chosen to provide the proper spring rate to load the seal adequately at operating temperature.

Problems of chemical compatibility of the seal material with the system fluid can occur. Adverse effects of fluid on elastomers include surface destruction, loss of strength, excessive volume swell or shrinkage, change of hardness, and change in elongation. Plastic interface materials may burn when exposed to fluorine or cause an explosion when subjected to an impact load in a liquid-oxygen system. With a metal seal, the metal itself may be chemically attacked by propellants.

The effects of temperature and fluid compatibility are discussed in greater detail in section 2.6.1.

### **2.5.2.4 SEAL RESILIENCE**

Seal resilience is a measure of the capability of the seal to accommodate coupling deflection and separation without seal load being reduced to a level at which the allowable leakage rate

is exceeded. Elastomeric materials have a natural resilience that is used to effect the initial seal and then the pressure acting on the seal utilizes the elastic properties to maintain the seal. Metals have little resilience; therefore, resilience in metallic seals is provided in the seal configuration.

Resilience in elastomeric seals does not present a sealing problem as long as the seal is used within the temperature range over which it remains elastic. Problems with metallic seals occur when a seal with low resilience is used to seal a lightweight coupling. Since aerospace couplings must be lightweight, it is virtually impossible to design couplings without some deflection under operational conditions. This problem has been minimized by using metallic seals with high resilience. Table I includes a number of pressure-assisted and radial seals with good resilience as reflected by the allowable coupling separation.

### 2.5.2.5 SEAL PROTECTOR

Seal barriers are structures that protect the seal and coupling. Seal applications in hot-gas systems are the most troublesome because of loss of material properties and the thermal effects on the coupling components. At the annealing temperature of the metal, the seal loses most of its initial preload and will leak. Also, at these temperatures the flanges are less rigid and warp. A large number of hot-gas couplings are in exhaust systems for solid propellants, and solid combustion products may cause additional problems such as erosion of the seal tip. Problems of annealing and erosion have been solved by installing a barrier on the inner side of the seal to protect the seal from both the high-temperature gas and the abrasive material (fig. 27).

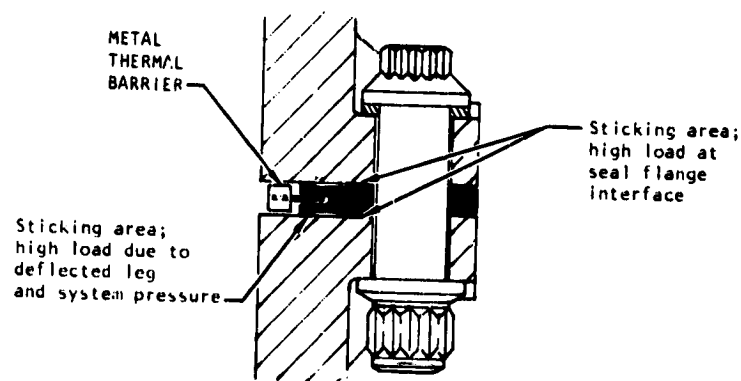


Figure 27. — Coupling design providing protective barrier for seal.

### 2.5.2.6 LEAKAGE CONTROL

Leakage control is utilized to isolate leakage to prevent incompatible fluids from coming into contact with each other and to ensure that the "blow-torch" type of exterior leakage from the thrust chamber does not occur.

Separation of incompatible fluids such as liquid oxygen and fuel or lube oil has been a problem with both static and dynamic seals when a single seal was used to separate two fluids or a single vent port was used to vent leakage. This kind of problem has been resolved by separating the seals and by using separate vent areas to port leakages to different areas (fig. 28).

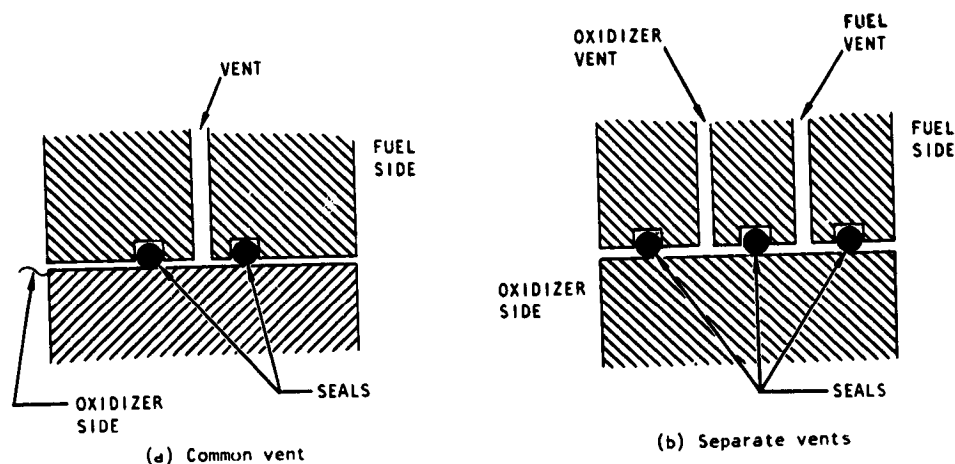


Figure 28. — Use of seals and vents to separate incompatible fluids.

On ground test of the RL10 engine, the oxidizer-pump seal dam is pressurized with helium to prevent mixing of any oxidizer and fuel leakage. In actual in-space operation of the engine, the vents are opened to environmental pressure, the philosophy being that regardless of the magnitude of the leak that might develop, pressure great enough for initiation of combustion does not exist around the ports.

Leakage of hot gas from the thrust chamber is minimized by using a combination coupling. In both the F-1 and J-2 engines, the coupling configuration is shown in figure 29. If the hot-gas seal leaks, fuel, which is at a higher pressure than the combustion products, leaks into the combustion chamber and not to the exterior of the engine.

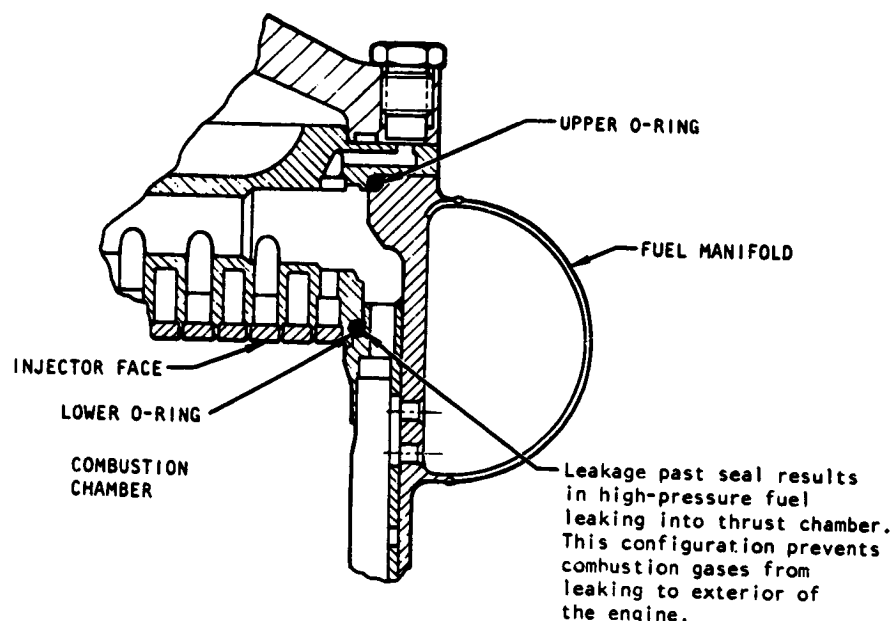


Figure 29. — Configuration for hot-gas seal for thrust chamber.

### 2.5.3 Dynamic-Seal Configuration

Dynamic seals are mechanical devices used to minimize or reduce to an acceptable level leakage from one region to another when there is relative motion at the sealing interface. Relative motion, either sliding or rotary, generally occurs between a static or stationary interface and a moving interface. In sliding motion, the direction of motion is perpendicular to the seal interface, with the contact surface continuously changing location. In rotary motion, the direction of motion is in line with the seal contact surface. The motion may be continuous or intermittent, reciprocating in sliding applications and oscillating in rotary applications. The types of dynamic seals discussed herein are limited to applications in which fluid pressure must be contained and leakage around the moving members of components (e.g., valve stems, actuator shafts, and pistons) is minimized. As noted, not included herein are dynamic seals for high-speed rotating shafts, valve closures that must move off and onto a seat, or hermetic-type seals such as diaphragms and bellows; these seals are treated in detail in references 1, 6, and 19. One of the best studies on dynamic sealing as discussed herein is reported in reference 54.

Dynamic seals are designed with a balance between fluid leakage, friction, and wear. No leakage usually is a design goal; however, in order to obtain acceptable friction and wear life,

leakage is allowed. In some designs, the leakage acts as a lubricant and lowers friction and improves wear life. If leakage cannot be tolerated, bellows or diaphragms are used in place of dynamic seals.

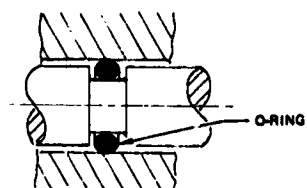
Figure 30 illustrates the principal types of rotary and sliding dynamic seals; their characteristics are summarized in table III. In liquid-rocket propulsion systems, elastomeric O-rings (figs. 30(a) and (b)), as in static seals, have a wide range of application. They have no equal when they can be used with chemically compatible fluids and operate within their temperature range. Among the fluids with which O-rings are used are hydrocarbon fuels such as JP-4 and RP-1, storable propellants, hydraulic oils, and helium and nitrogen gas in pneumatic systems. The many different compounds available make possible a matching of elastomer and operational fluid properties to the best functional advantage. The all-around reliability and low cost of elastomeric O-rings make them the logical seal choice for any application in which they can be used.

Lip seals had a long and successful history of usage in commercial and military hardware before their adaptation to aerospace designs. Plastics such as Kel-F, Mylar, and Teflon replaced the elastomeric and leather materials for cryogenic applications. Lip seals are used for both rotating and sliding types of motion. The chief advantage of lip seals over O-rings lies in the lip seal configuration (figs. 30(c), (d), (e), and (f)). The basic configuration exploits material resilience and flexibility to achieve an effective seal even at cryogenic temperatures. In addition, the lip seal has greater ability to absorb radial motion between male and female members that results from thermal expansion and deflections.

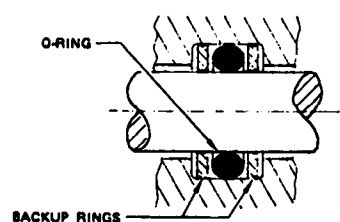
Piston rings and mechanical seals generally are used for sealing hot gases where elastomeric seals would deteriorate. Piston rings have been used in commercial applications as seals in internal combustion engines and compressors, and in industrial hydraulic and pneumatic cylinders where the ruggedness of piston rings is advantageous and where leakage can be tolerated. Various ring-seal configurations and special mechanical-seal configurations illustrated in table III are used in both rotating and sliding applications in rocket propulsion systems. These metal seals require close machine tolerance for both seals and cavities, smooth surface finishes, and wear-resistant surfaces for obtaining the required leakage level and cycle life.

Other rotary and sliding seals with limited flight usage include the various compression-packing designs. The compression packings are limited to slow surface speeds and offer no advantage over elastomeric O-rings in flight applications.

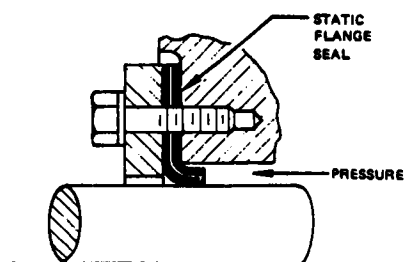
References 9 and 10 discuss the state of the art for the various seal configurations, provide considerable design information, and include a large bibliography useful in seal design.



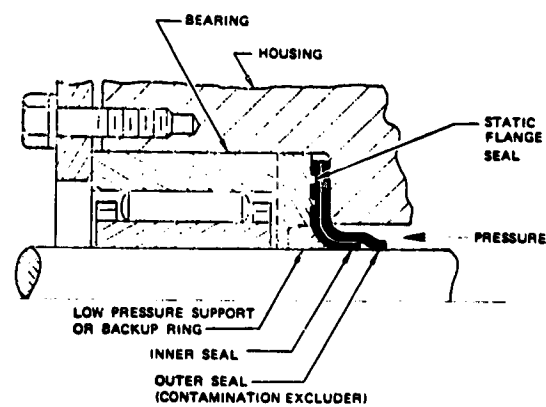
(a) O-ring



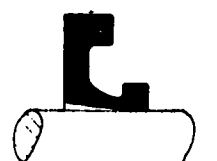
(b) O-ring with backup rings



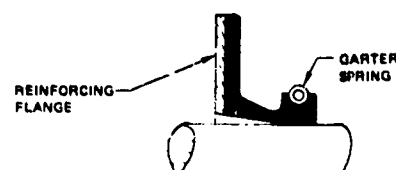
(c) Lip seal



(d) Lip seal with backup ring and contamination excluder



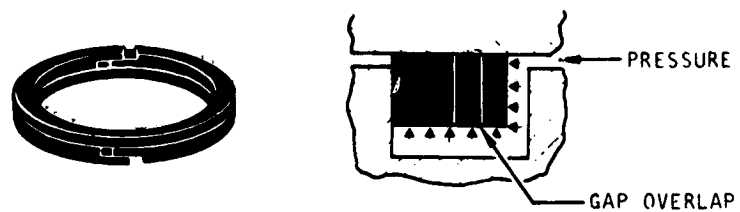
(e) Formed lip seal



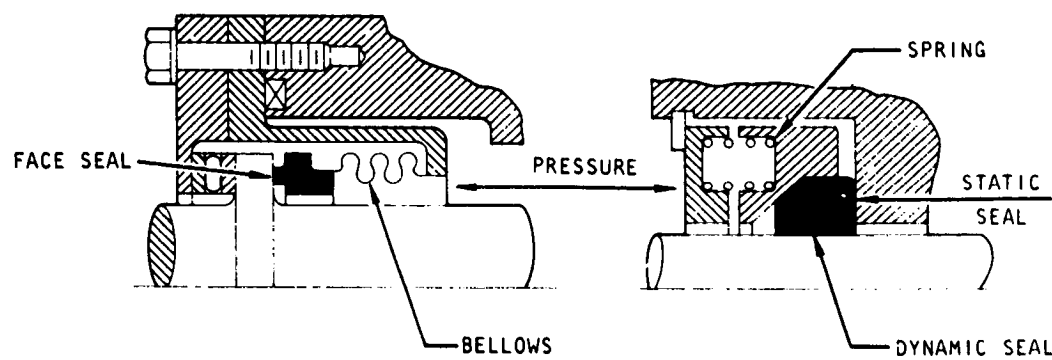
(f) Formed lip seal with reinforced flange and garter spring

Figure 30. — Principal types of rotary and sliding seals.





(g) Two-piece piston ring

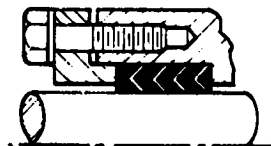


(h) Bellows-loaded face seal

(i) Spring-loaded combination seal



(j) Compression packing



(k) Chevron seal

Figure 30. — Concluded.

Table III. Chief Characteristics of Principal Types of Rotating and Sliding Seals

Configuration	Application		Method of sealing	Method of loading	Material	Operating pressure range, psia	Operating temperature range, °F	Interface requirements		Fluid sealed
	Rotating	Sliding						Surface finish $\mu$ in. rms	Hardness	
O-ring	Yes	Yes	Circumferential	Pressure energized	Fluoromer	0 to 3000	80 to +600	5 to 6	$R_c \geq 50$	Hydrocarbon fuel, storable propellants, helium and nitrogen gas, hydraulic oil
O-ring w. Teflon, hard rubber, or leather backup rings	Yes	Yes	Circumferential	Pressure energized	Fluoromer	0 to 5000				
Lip seal (pressure energized or assisted)	Yes	No	Circumferential	Interference fit, pressure assisted	Kel-F, Mylar, Teflon	0 to 3000	452 to +140	5 to 16	$R_c \geq 30$	Hydraulic oil, storable propellants, liquefied gases
Lip seal	Yes	Yes	Circumferential	Interference fit, pressure assisted	Fluoromer	15 to 150	80 to +250	10 to 20	$R_c \geq 30$	Hydraulic oil, storable propellants
Dead element lip seal	Yes	Yes	Circumferential	Interference fit, pressure assisted	Nonmetallic metallic combinations	15 to 150	80 to +600 Fluorocarbon polymers 120 to +400	15 to 20	$R_c \geq 50$	Oils and hydraulic fluids, water
Crossgroove lip seal	Yes	Yes	Circumferential	Interference fit, pressure assisted	Kel-F, Mylar, Teflon	15 to 3000	452 to +140	10 to 20	$R_c \geq 30$	Liquefied gases
Compression packing	Yes	Yes	Circumferential	Mechanically preloaded	Leather, synthetic leather, rubber, and Teflon	15 to 10 000	65 to +180 for leather 65 to +250 for rubber 65 to +300 for Teflon	16 to 32	Minimum Brinell of 500	Ash, hydrocarbons, water, steam
Piston ring	Yes	Yes	Combination circumferential and face	Mechanically preloaded, pressure assisted	Steel, cast iron, bronze, fluorocarbon polymer	15 to 4000	452 to +2000	16 to 20	$R_c \geq 50$	Cryogenics, hot gases, hydraulic fluids, hydrocarbons
Bellevue actuated rotary face seal	Yes	No	Face	Mechanically preloaded, pressure assisted	Steel, nickel base alloys, carbon	0 to 3000	452 to +1200	16 to 20	$R_c \geq 30$	Hot gases, liquefied gases, hydraulic fluids, hydrocarbons
Spring actuated mechanical seal	Yes	Yes	Combination circumferential and face	Mechanically preloaded	Steel, nickel base alloys, fluorocarbon polymer					

### 2.5.3.1 O-RINGS

Because of their simplicity of design, extensive standardization, low cost, and excellent reliability record in the aerospace industry, elastomeric O-rings (figs. 30(a) and (b)) are, within their temperature and compatibility limitations, preferred for dynamic seals for both rotating and sliding dynamic applications. The wide choice of elastomeric compounds available permits the use of the O-ring throughout the pressure range from near vacuum to over 3000 psi and at temperatures from about -80°F to approximately +450°F. Installations above 800 psi may require a special gland design. Compounds are available for use with pneumatic, hydraulic, and hydrocarbon fluids and many high-performance propellants. Elastomeric materials are discussed in detail in section 2.6.1.2.

O-rings are used in various mechanical components such as valves and actuators to seal the fluid while still permitting relative motion. The O-ring installations generally are controlled by the requirements of the O-ring gland design specifications (ref. 48). The principles involved in use of the O-ring are the same for both static and dynamic applications and are discussed in section 2.5.1.1.1.

Problems associated with dynamic O-rings include damage during installation, improper squeeze, extrusion, and damage due to surface speeds; leakage and contamination can result from each condition. The O-ring installation-damage problem is the same as that for static seals (sec. 2.5.1.1.1).

Squeeze, too little or too great, can cause problems. A low squeeze can result in leakage at low pressure or extrusion of the O-ring into the clearance between the shaft and housing; the latter condition under high pressure can result in O-ring damage and subsequent leakage. Low squeeze may result from a groove that is too deep, deflections that are excessive, or, in low-temperature applications, a reduction of O-ring cross section due to the contraction of the elastomer. A low-squeeze condition is corrected by adjusting the groove depth, reducing the deflections, or changing the elastomer compound. A high squeeze can result in installation damage and is corrected by increasing the groove depth. O-ring squeeze is discussed further in reference 48.

Extrusion occurs because of excessive clearance between the two moving members and is due to improper dimensions or excessive loads on deflections or an O-ring that is too soft. Extrusion may be eliminated by one or more of the following changes: use of a harder elastomer, reduction of clearances, or the installation of backup rings. Damage to the O-ring in the form of a spiral failure (i.e. the O-ring looks as if it has been cut about half way through its cross section in a spiral or corkscrew pattern) occurs in reciprocating applications with speeds less than one foot per minute and low pressures. This damage may be eliminated by increasing the speed or fluid pressure. In rotary applications, if speeds are too great, the O-ring is destroyed. This possibility is minimized by using a relatively hard elastomer, installing the O-ring in the cylinder with very light tension loads, and limiting surface speeds to less than 1500 ft/min.

### 2.5.3.2 LIP SEALS

A lip seal consists of a relatively flexible sealing element, generally nonmetallic, that has an interference fit with the moving element and extends into the fluid cavity so that fluid pressure acts to increase the seal load (figs. 30(c), (d), (e), and (f)). The sealing element can be made of almost any material but in aerospace application is generally an elastomer or fluorocarbon polymer. Lip seals offer a wide range of design parameters (material, thickness, length of lip, flange retention) to meet particular requirements, and perform well for either rotating or sliding applications in such fluids as hydraulic oil, RP-1 fuel, storable propellants, and cryogenic propellants.

Lip seals are molded from elastomeric materials for applications within the useful temperature range of the elastomer and are machined or formed from plastic materials for cryogenic applications. A large selection of configurations and sizes for elastomeric lip seals are available "off-the-shelf" commercially. Plastic seals, on the other hand, generally are designed for the particular application. Several butterfly valves for use with  $\text{LH}_2$  on the Saturn vehicle have Mylar lip seals on the butterfly shaft. These seals are formed at the time of installation from blanks pre-cut from sheet stock. For liquid-oxygen service on the Saturn vehicle, an advanced butterfly valve incorporates a Kel-F shaft seal of similar design. Metal lip seals have been developed for reciprocating shafts in the hydraulic system for aircraft service. The operating conditions include pressures up to 6000 psi and temperatures to  $+500^\circ\text{F}$ . The metal lip seal is operated against a lapped, flame-plated shaft.

Problems with lip seals usually are related to (1) insufficient pressure of the seal lip on the moving element and (2) roughness of the dynamic sealing surfaces, which influences cycle life. Design features intended to solve these problems often are in conflict, since high bearing pressure of lip on shaft is conducive to rapid wear of the sealing surfaces. A successful design is the result of the balance between several factors such as the seal's tensile strength, thickness, and wear resistance; shaft surface texture and wear resistance; and the effect of the operating environment on these properties. Garter springs on elastomeric seals and multi-ply plastic seals often are used to withstand high operating fluid pressures while maintaining lip flexibility. Wedge-shaped backup rings between seal and housing and shaft also are used on the low-pressure side of the seal to support the seal against pressure. Overlapping the outer lip of a multi-ply seal, as shown in figure 30(d), provides a contamination excluder and aids in maintaining contact between the inner seal and the moving element.

The flange of a molded elastomeric lip seal that provides a static seal between the lip seal and the stationary housing usually is reinforced with a metal ring or encased in metal. This reinforcement is required to provide rigidity for installing and effecting a seal at the static mounting face and on some designs to provide additional loading at the dynamic seal face. Design of the flange of a plastic lip seal for cryogenic service is critical to satisfactory performance. Two methods that are often used to maintain a flange load are temperature

compensation and serrations or raised lands (fig. 30(c)). Temperature compensation is accomplished through the use of material combinations that will maintain (or preferably increase) the flange load at low temperatures. For example, the Kel-F flange seal on the lip seal used in the liquid-oxygen butterfly valve for the Saturn vehicle is designed to sustain 2000-psi flange bearing pressure at assembly. The contraction of the 0.020-in. thick seal is compensated by a 440-CRES bearing race in an aluminum alloy housing to raise the bearing pressure to approximately 4000 psi at  $-300^{\circ}\text{F}$ . A 0.015-in. Mylar flange seal for a liquid-hydrogen butterfly valve on the same system has similar compensation: 3000-psi flange bearing pressure increases to approximately 8000 psi at  $-420^{\circ}\text{F}$ . Both seals are further loaded by narrow projections on the seal retainer—the Kel-F by two 0.004-in.-high lands, and the Mylar by a 0.004-in.-high serration.

### 2.5.3.3 PISTON RINGS

Piston rings (fig. 30(g)) as used herein are split rings that either expand (piston rings) or contract (rod seals) to effect sealing and are used in high-temperature seal applications. They can be used as either sliding or rotating seals. The rings are installed with little or no sealing load and are forced into position and loaded by the system pressure. Sealing occurs on two surfaces, a circumferential surface and a groove side wall. The rings are precision fabricated to obtain nearly perfect surfaces. The split is made in either a straight cut or step cut. The straight cut has few applications because of excessive leakage; the step-cut ring, singular or in clusters of two or three, provides improved sealing. On contracting rings, springs are generally required to ensure the best conformity of the seal ring to the rod. Because the sealing loads are low, the friction, wear, and life characteristics are good. Piston-ring usage on rocket propulsion systems has been limited to applications where controlled or fairly large leakages are acceptable.

Leakage problems associated with piston rings are related to surface damage and seal load; use of step rings or multi-rings lessens leakage. Leakage caused by surface damage can be minimized by the proper selection of materials with good wear and friction properties. If leakage requirements are reduced or are found to be in excess of that which a piston ring can consistently seal, then a mechanical seal may be used. Loads are varied by pressure balancing, which consists of adding a groove in the wear surface and venting it to the high pressure by a passageway in the seal ring. In this manner the seal pressure-assist load can be varied by the amount of grooving and venting added. Auxiliary springs are also used to improve the seal load in low-pressure applications. Figure 31 shows both the pressure-balance groove and springs for increasing the sealing load.

### 2.5.3.4 MECHANICAL SEALS

Mechanical seals function with a mechanically loaded dynamic seal interface on a face normal to the shaft (fig. 30(h)) or on the shaft (fig. 30(i)). Pressure forces increase the seal

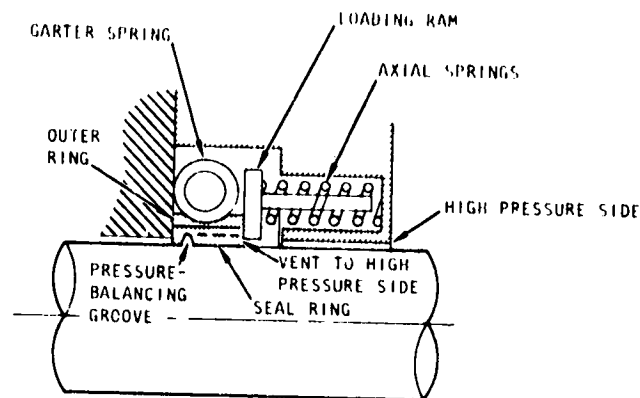


Figure 31. — Two-piece rod seal with pressure-balancing groove and spring load.

load in high-pressure applications. Mechanical seals generally are used where plastics or elastomers are unsuitable because of propellant incompatibility or operating temperature range; however, plastics and elastomers occasionally are used in mechanical seals.

Problems with mechanical seals include low seal loads and cycle wear life. The loads required for sealing are obtained initially by a spring or bellows load and generally are supplemented by pressure to increase the sealing efficiency at the higher pressures. Acceptable cycle wear life is obtained by selecting material combinations that result in low wear rates; in addition, fine, lapped surface finishes are used to minimize wear. The combination of controlled leakage and minimum wear is the foundation of a successful design.

The seal concepts shown in figures 30(h) and (i) are current configurations in rocket propulsion systems. The seal shown in figure 30(h) is employed in rotational applications where all-metal seals are required (e.g., high-temperature applications). This configuration, for example, is used to seal the shaft in the hot-gas tapoff valve for an upper stage of the Saturn vehicle, where the service temperature ranges from cryogenic to +1200°F. Figure 30(i) is an illustration of a mechanical seal that is used in both rotational and sliding applications.

## 2.5.4 Dynamic-Seal Design Integration

The various requirements applicable to the design integration of all dynamic seals include loads; contact surface; seal deflection and resilience; wear; lubrication; and contamination.

### 2.5.4.1 LOADS

The primary loads on dynamic seals are fluid-pressure loads, mechanical loads, (operational and installation), and thermal loads (fluid and component temperature). An effective seal is capable of handling these loads as they are related to structural stresses (tensile, compression, bending, and shear), contact (bearing) stresses, friction, and wear. Elastomeric and plastic seals as shown in figure 30 accommodate these loads and stresses with a unique combination of configuration and material properties providing both the required strength and resilience. Metallic seals such as those shown in figures 30(h) and (i) rely on rigid, finely finished sealing contact surfaces, with deflections absorbed through gross seal displacement.

An effective seal maintains a bearing contact load at the seal interface adequate to control leakage, the bearing load being related to the operating fluid pressure and properties and to the topography of the sealing surfaces. Wide variations in these loads account for the majority of dynamic seal failures and become the most difficult of the design problems in providing a satisfactory seal. Typical problems include low-pressure leakage when adequate installed load is not provided, high-pressure leakage when the pressure-activated area is not large enough or distortion creates a leak path, and leakage when the moving member is displaced perpendicular to its normal motion, a deflection that cannot be followed by the seal. Other problems include deflection due to mechanical loads generated by external forces or thermal loads, high friction, and excessive wear related to load. A successful seal design controls leakage with the lowest possible bearing load and limits the maximum load to one that is tolerable with respect to friction and wear. Also, a successful seal functions within the leakage limits when the bearing load is reduced by wear or creep; or when the surfaces are degraded by contamination generated externally or by the seal itself as it wears.

The frictional force acting on a dynamic seal at the dynamic interface is a function of the seal load and the coefficient of friction at the interface. Both dynamic and static friction are important in the seal design. Prediction of the coefficient of friction to be expected in actual operation is complicated because the value of the coefficient depends on load, material structure, surface topography (roughness), contact area, temperature, lubrication, and related factors. In addition, dynamic friction is dependent on surface velocity, and static friction is dependent on the time at rest prior to movement. Complicating these factors are such things as wear, which causes changes in many factors, and the tendency of an O-ring to roll before sliding in a sliding application. Reference 9 discusses the state of the art, gives a considerable amount of design data, and includes a large bibliography on the prediction of frictional forces on dynamic seals. The design approach used to minimize friction problems includes selecting proper materials; minimizing pressure, mechanical, and thermal loading; specifying well-finished surfaces; and providing lubrication where possible. Friction in some seals is high enough to make a significant contribution to mechanical loads and is included in the analysis of structural stresses.



Pressure Loads. Most dynamic seals including O-rings and lip seals are pressure assisted to achieve higher sealing interface loads with the objective of minimizing leakage at higher pressures. The static-seal contact surface may or may not also have this feature. At the dynamic seal interface, the pressure-assist feature results in two outstanding disadvantages: (1) the higher contact loads produce high friction and wear, and (2) except for O-rings and some compression packings, the seal will restrict leakage with pressure applied from one side only. This latter condition becomes a significant consideration in the design of a double-acting piston seal when it is not practical to use elastomer O-rings. To solve this problem on applications utilizing lip or cup seals, two seals are generally installed back-to-back. The methods used to reduce high-pressure contact stresses vary with seal configuration. O-rings, for instance, often give better results with an adjustment in groove depth, cross section, or elastomer hardness that alters the contact area and bearing stress. Lip and cup-type seals may be altered to change the relationship between lip length and contact area, whereas face-type seals may be altered by a change in the relationship between bellows effective diameter and seal seating diameter. In the design of high-performance seals for flight-type hardware capable of operating throughout extended limits of pressure and temperature, loading problems are resolved by extensive design analysis and development tests.

Mechanical loads. Mechanical loads are applied to a seal in preloading during installation, in deflection of the seal with relative radial motion between components during operation, and by deflection of mating components by external loads during operation. In some cases (e.g., the compression-type seals used in some valve stems), the seal also acts as a bearing that carries radial shaft load.

Mechanical loads are additive in some seal designs, and sealing integrity is not lost. In other cases, notably at the dynamic seal interface, deflection reduces the load, and the installed preload must be high enough to prevent excessive leakage. These problems are minimized by performing thorough analyses to evaluate the loads and deflections and by dimensional control of the seal contact surfaces (e.g., flatness, roundness, and normality).

Thermal Loads. Thermal loads are generated when temperature changes in the component cause expansion or contraction of the various members at different rates; these differentials arise because of temperature gradients or differences in the thermal expansion coefficients of the various members. These loads will either increase or decrease the loads at the seal interfaces and structural stresses within the seal. Temperature changes also create distortions that affect interface loads and structural stresses. Temperature compensation designed into many seal installations reduces the variation in load with temperature change. The (cryogenic) seal shown in figure 30(c) sometimes incorporates an Invar spacer under the head of the bolt to compensate for shrinkage of the plastic seal at cryogenic temperatures and maintain the flange load. Thermal conditions that lead to distortions are minimized with combined heat-transfer/structure studies.

### 2.5.4.2 CONTACT SURFACES

The basic function of a dynamic seal is to provide contact between the seal and the moving component sufficient to limit the leakage path and prevent leakage from exceeding allowable levels. The geometry (size) of the leakage path is a function of the seal configuration, load (pressure and mechanical), and surface finish. The general subject of leakage as a function of interface topography and fluid conditions is covered in detail in reference 9. A successful seal design is one wherein the load at the contact surface is sufficient to prevent excessive leakage and low enough to prevent excessive friction and wear. Design requirements for O-ring seals used in most aerospace hydraulic-oil and hydrocarbon-fuel applications are given in reference 48. Dynamic O-ring seals designed to these requirements will have the proper contact surface configuration for most applications.

Successful operation at extreme temperatures requires seal design more sophisticated than the O-ring. The seal shown in figure 30(c) has been used in several rotating and sliding applications sealing liquid hydrogen and helium at cryogenic temperatures on Saturn upper-stage valves. The seal design has been developed by empirical methods based on a Mylar seal working on a fine-finish hardened shaft. The proper seal configuration is provided by cold forming the seal lip from a blank at installation and mating it with an 8- $\mu$ in. shaft finish. A seal designed for high temperatures is shown in figure 30(h). This seal has been in use in hot-gas valves at 1200°F. The dynamic seal interface material is flame-plated tungsten carbide that is lapped to a 4- $\mu$ in.-rms finish and loaded by bellows compression and fluid pressure on the bellows. Carbon has also been used for the interface material in this design.

### 2.5.4.3 SEAL DEFLECTION AND RESILIENCE

Most rotating and sliding dynamic seals are designed to deflect in one way or another so that they can comply with relative dimensional changes between the moving and stationary members. Resilience of a seal is its ability to return to its original shape after release of an applied load that does not deform the seal beyond the elastic range. Seal resilience is a function of geometry and material properties. Resilience determines the capability of the seal to maintain contact under conditions of deflection of the mating surfaces. High-performance seals for flight-type hardware require materials that have high strength and adequate resilience throughout the operating range.

All-metal seals such as the hot-gas face seal shown in figure 30(h) are exceptions. Intimate contact at the dynamic contact surface is achieved with rigid seal elements finely lapped for finish and flatness. Compliance with relative shaft-to-housing radial and axial displacement is achieved by deflection of the bellows that is incorporated between seal and housing.

#### 2.5.4.4 WEAR

Wear is a characteristic of all dynamic seals and is not considered detrimental until it causes excessive leakage, friction, or contamination. Wear is affected by seal rubbing velocity, because (1) a high rubbing velocity increases wear rate directly, and (2) high rubbing velocity can increase real operating temperature, which can increase wear rate and also shorten seal life. The normal operating life of a seal may be dictated by the wear rate and be independent of the life of the component. Thus wear rate dictates seal replacement on a component servicing schedule.

Controlling wear is largely a matter of selecting appropriate material, specifying proper surface finish, and controlling the bearing pressure at the dynamic contact surface. Lubricants are used to advantage in most O-ring pneumatic installations. The bearing pressure of an O-ring is largely a function of squeeze and is a significant factor in the design of the O-ring groove. Material properties and configuration both play a large part in the bearing pressure of a lip seal. In one application of the lip seal shown in figure 30(c), an excessive wear problem was resolved by reducing bearing pressure through a reduction in material thickness, with no degradation in performance. Finer surfaces in general reduce wear, but the nature of the surface must be considered. A surface with a smooth finish with sharp asperities may cut and abrade a seal faster than one that is rougher but with rounded asperities. Conversely, too fine a finish can be a detriment in some cases. A surface finish of less than 8- $\mu$ in. on the rod of a reciprocating piston in hydraulic service may be wiped dry of lubricant, the result being O-ring failure.

External sources of contamination can also result in seal wear; therefore in some applications wipers are added to exclude contamination. Operating with contaminated fluid may be a design requirement. Corrosive wear occurs when the base metal exposed through the rubbing wear process intensifies the wear rate due to corrosive action. Compatibility of the basic materials of the dynamic sealing elements with the operating environment eliminates corrosive wear.

#### 2.5.4.5 LUBRICATION

Most dynamic seals depend upon some form of lubrication to control friction and wear. The lubricant may be applied at the time of seal installation, like the grease applied to an O-ring for pneumatic service; it may come from the operating fluid, or it may be a component of the seal material. The most difficult lubrication problems occur at temperatures at which lubricants cannot be used. Plastics such as Teflon or Kel-F have been used successfully either as the basic seal material or as coatings. Hard coatings such as tungsten carbide and platinum such as chromium also have been used to reduce friction. Soft metal films such as silver or indium have been reported as successful in reducing friction, but they are extremely delicate.

#### **2.5.4.6 CONTAMINATION**

The capability of sealing fluids that contain varying degrees of contamination is a design requirement for all dynamic seals. On the other hand, excessive wear of the seal may generate contamination detrimental to the operation of the system of which it is a part. Contamination of varying degrees can be absorbed into the dynamic sealing interface without degrading the seal. Beyond this, greater amounts of contaminants or more abrasive particles will create leakage paths or wear and result in seal failure. The amount of contamination that can be absorbed varies with seal configuration and material. In general, soft materials and large dynamic-seal contact-surface areas will absorb more contamination, whereas harder materials are more resistant to abrasion. This consideration often is a prime factor in the selection of a soft seal to operate on a hard surface. However, the amount of wear that can be tolerated may be dictated by the system of which the seal is a component, rather than by the effect of the contamination on the seal.

### **2.6 GENERAL CONSIDERATIONS FOR DESIGN**

#### **2.6.1 Material Selection**

The materials used for disconnects, couplings, fittings, fixed joints, and seals provide (1) the structural strength necessary to withstand all the various loads, (2) special properties for the application (e.g., softness and pliability for seals), and (3) the properties necessary for the joining of components by welding or brazing. The materials used include metal alloys, metal platings, elastomers, and plastics. Material selection considerations include compatibility with service fluid, physical and mechanical properties, environmental temperatures, welding and brazing characteristics, contact of dissimilar materials, corrosion resistance, and cost.

##### **2.6.1.1 METALS**

The metals currently used for the subject components include most of the common structural metals that are inherently corrosion-resistant; principally, these are the 300-series corrosion-resistant steels, nickel-base alloys, and aluminum alloys. Several of the subject components (fittings, couplings, and fixed joints) are integral with a component or are connected to a duct; therefore, the material selection considers the component or duct requirements. On disconnects, where weight is not a significant factor, 300-series corrosion-resistant steels generally are used for both half-bodies and for the latch release mechanisms. Aluminum alloys are compatible with the majority of the requirements of the flight half, but are avoided when possible because of their vulnerability to damage to critical surfaces and mechanism during handling. Since electrical disconnects frequently are located close to fluid disconnects, it may be necessary to avoid magnetic materials. Springs and the

highly loaded latching-mechanism parts require heat-treatable alloys such as 440C CRES or nickel-base alloy. Materials selection for valve components is discussed in reference 6.

Compatibility of metals with the system fluid and the atmosphere is an important consideration. Various fluids used in rocket engine systems are either corrosive to metals or decompose when in contact with metals. To avoid problems caused by these reactions and guide the selection of suitable materials, the chemical compatibility of the fluid and metal is analyzed during the design phase (refs. 56 and 57). In some cases, it is possible to make the metal surface passive to corrosion; the passivation permits the material to be used with a particular fluid. An example of this procedure is the passive fluoride film on stainless steel. The passivation film operates on the theory that the increase in passive-film thickness reduces the rate of corrosion until an equilibrium between the reaction rate and solubility of the film is reached. At this point, there is a slow, relatively steady corrosion rate that is within design limits.

Some heat-treated materials such as 2014-T4 and -T6, 17-7PH, and AM355 have been found to be susceptible to stress corrosion. This problem is avoided by changing the design to use materials not susceptible to stress corrosion (ref. 58).

Galvanic corrosion can be a problem when dissimilar metals are in contact with each other. This problem is minimized by the selection of materials within compatible limits in the electrochemical series and by use of appropriate finishes and platings (refs. 59 and 60).

Because of the growing use of liquid hydrogen in the space program, the designer must consider that a wide variety of pure metals and alloys has been found to be susceptible to hydrogen-environment embrittlement. The embrittlement is an environmental effect: when a susceptible metal comes in contact with newly formed hydrogen, the surface of the metal behaves in a brittle manner when a certain amount of plastic deformation has occurred. Elastic properties, yield strength, and, in many cases, the ultimate tensile strength are not affected by the hydrogen environment; the most significant effects are on tensile ductility, notch strength, and crack behavior. Most structural alloys are embrittled by hydrogen environments; therefore, those alloys that have not been tested in hydrogen are assumed to be susceptible to embrittlement. Only properties measured in a hydrogen environment are used in designing structures for use in a hydrogen environment. Notches or regions of stress concentration are reduced and, if possible, eliminated; surface finishes are tightly controlled; and all surface material defects are removed. Temperature and pressure are important variables; most affected metals are susceptible to a measurable extent only over the temperature range of  $-100^{\circ}$  to  $+300^{\circ}$  F, and the susceptibility increases with increased pressure. Two methods for prevention of hydrogen-environment embrittlement are the use of protective coatings and the addition of inhibitors to hydrogen. Coatings that have shown the most potential for preventing hydrogen-environment embrittlement are copper, gold, and cadmium. Silver-silicon and tin-aluminum coatings have shown some promise for elevated-temperature service. The degree of hydrogen-environment embrittlement decreases

with the amount of oxygen in the hydrogen. For example, the addition of 1-percent oxygen has been shown to prevent embrittlement of a 0.22-percent carbon steel by high-pressure hydrogen. Reference 61 provides detailed information on hydrogen-environment embrittlement of metals, including properties of many structural metals tested in a hydrogen environment.

Materials with widely different coefficients of expansion or contraction have resulted in bolting problems or improper clearances or fits in applications with large operating-temperature ranges. Improperly preloaded bolts are loosened by differential thermal movement when the coupling contracts faster than the bolts, or the bolts yield because the coupling expands faster than the bolts. Improper clearances or fits can cause an operation problem in disconnects and in inadequately welded or brazed joints. These problems are minimized by using materials with similar coefficients of expansion, by evaluating the effects of expansion or contractions, and by making a detailed structural analysis on couplings to determine the proper bolt preload. Reference 62, 63, and 64 provide coefficient-of-expansion data on aerospace materials.

Physical and mechanical properties are important since most of the components are highly loaded to reduce weight. Materials with high strength-to-density ratios are used extensively. At extremely low and high temperatures, ductility or toughness decreases for most metals; this change can result in a rupture failure. Thus materials must be selected to obtain the ductility required at the operating temperature.

Weldability of materials is a very important consideration since welding is the most widely used method for joining fittings and couplings to other components and for making fixed joints. Most currently used components that require a fixed joint are effectively welded by either the gas-tungsten-arc (GTA) or gas-metal-arc (GMA) method. On hot-gas applications, dissimilar materials are sometimes welded (e.g., Hastelloy-C to 347 CRES).

Bolt-and-nut materials used on rocket engines are inherently corrosion-resistant, the most widely used materials being A286, Inconel 718 and X-750, K-Monel, and Rene 41. Material selection is based on the strength level required for the coupling and the service temperature. All of the above materials except Rene 41 are used in applications ranging from cryogenic temperature to 1200°F. Rene 41 normally is used at temperatures in the 1200°F to 1600°F range because of good strength levels, but high cost restricts it from being used at lower temperatures. In order to prevent thread galling, plating and lubricants are used on bolts and nuts. Silver is the most widely used plating, although others are used when required to solve fluid-compatibility problems. The standard thread lubricant used on the Saturn engines is a phosphoric-acid-bonded dry-film lubricant. One problem area that influenced the selection of this lubricant was its impact-ignition compatibility with liquid oxygen. Useful temperature range for this dry-film lubricant extends from cryogenic temperatures to +300°F. Platings are used at higher temperatures.



Cast aluminum components frequently provide a flange for a coupling. Porosity at the seal interface has caused leakage problems. This problem is minimized by specifying the required density for the sealing area on the casting drawing. This requirement is then used in the mold design, and the density is verified on qualification test. Stringers in 321-CRES wrought material resulted in helium gas leaks on the F-1 and J-2 engines. This problem was eliminated by using higher quality, vacuum-melt 321-CRES material.

### 2.6.1.2 ELASTOMERS

An elastomer is a material that returns to its original shape and dimensions after distortion. Selection of the proper elastomer compound is based on the service fluid, fluid pressure, and service temperature range.

Fluid compatibility is the first thing to be considered in selecting an elastomeric compound. All fluids that will come in contact with the elastomer (i.e., fluid to be sealed, outside air, lubricants, cleaning compounds, and purging agents) must be considered. Compatibility problems can occur if the elastomer selection does not take into consideration the chemical reaction between the fluid and the elastomer. Adverse effects include surface destruction, loss of strength, excessive volume swell or shrinkage, change of hardness, and change in elongation. Changes in these properties are accelerated, in most cases, by increased temperature. An example of compatibility problems occurred on the F-1 engine program when a solvent-flush requirement was added to the RP-1 fuel system. The elastomer utilized until that time, Buna-N, changed physical properties and swelled excessively when exposed to the solvent. To solve this problem, a different elastomer, Viton A, which was compatible with both the propellant and solvent, was substituted.

At times, the designer and materials engineer must go beyond conventional materials for a successful design solution. Sometimes it is possible to modify an elastomer compound to provide improved compatibility. An example of such modification is a new formulation for butyl rubber with improved resistance to nitrogen tetroxide ( $N_2O_4$ ), the oxidizer used on the Titan II system (ref. 65). In addition to a formulation change, the seal configuration was changed from an O-ring to a molded-in-place seal to minimize fluid exposure (fig. 32).

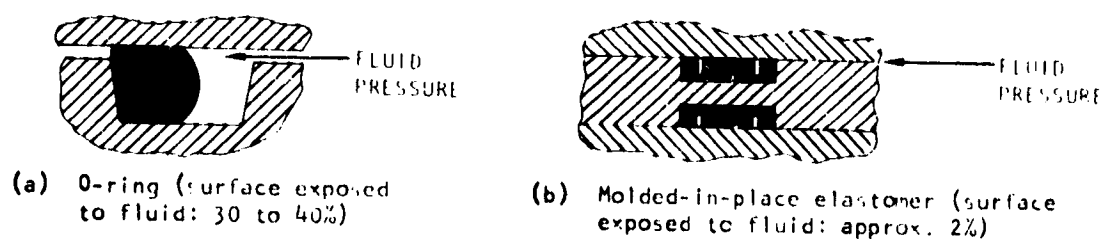


Figure 32. Elastomer exposure to system fluid for two types of seals.



Recent advances in the formulation of new and improved elastomers have resulted in compounds AF-E-411 for use in hydrazine seals, and AF-E-124D for use in hydrazine and nitrogen tetroxide seals. General properties and test data for these and other new elastomers are detailed in references 66 and 67. Compatibility data on conventional elastomers is presented in reference 68.

The system pressure must be considered in selecting an elastomer compound since it tries to force the elastomer out of the coupling. Therefore, the elastomer must have adequate modulus and hardness to resist extrusion or the housing must provide the necessary clearance limitation to prevent extrusion. These relations are shown in figure 22.

As noted, temperature influences elastomers by causing changes to the physical properties. Elastomer usage is limited by both high and low temperatures with respect to changes in strength, flexibility, and hardness. Table IV shows the useful temperature ranges for a number of elastomers, and figure 33 shows the effect of low temperature on hardness.

Table IV. Useful Temperature Ranges for Commonly Used Elastomers

Elastomer	Specification	Useful temperature range, °F
Buna N	MIL-P-5315 (ref. 69)	- 65 to 250
Butyl	AMS 3238 (ref. 70)	65 to 300
Fluorosilicone	MIL-R-25988 (ref. 71)	80 to 350
Polychloroprene (neoprene)	MIL-R-1149 C1-1 (ref. 72)	55 to 250
Polyurethane	MIL-R-83397 (ref. 73)	60 to 200
Silicone	AMS 3304 (ref. 74)	80 to 450
Viton and Fluorel	MIL-R-83248 (ref. 75)	10 to 600
Viton-A	MIL-R-25897 (ref. 76)	0 to 400

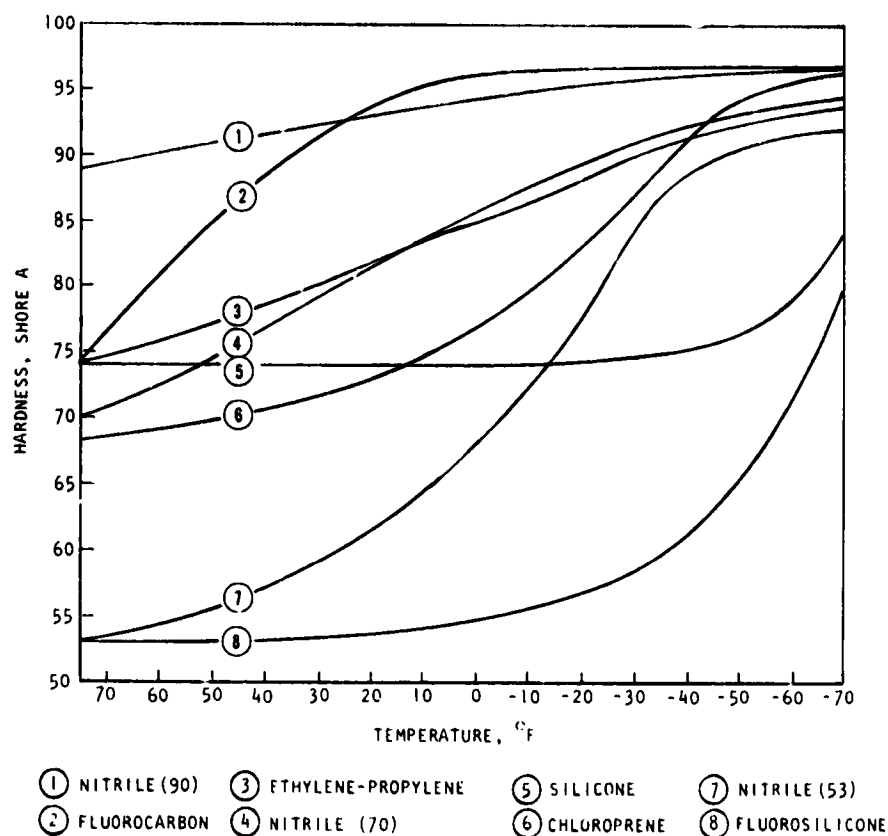


Figure 33. — Effect of temperature on hardness of various elastomers.

Low temperature causes elastomers to lose their elastic properties. This phenomenon limits the ability of elastomeric seals to follow the coupling surfaces, and results in the seal functioning like a gasket rather than being energized by the system pressure. This condition is intensified by the elastomer's coefficient of thermal expansion (or contraction, at low temperature), which is much greater than that of most metals and consequently has the effect of changing the initial installation squeeze on O-rings and molded-in-place seals. The combination of these factors results in elastomeric seals leaking at low temperatures. The best low-temperature elastomers are the silicones, which have demonstrated flexibility as low as  $-120^{\circ}\text{F}$ ; however, they have poor physical properties (e.g., tensile and tear strength) in comparison with other generally used elastomers and are incompatible with hydrocarbon fluids. Fluorosilicone elastomers, which are compatible with hydrocarbon fluids, are the

second-best material for low-temperature service, with good flexibility to  $-80^{\circ}\text{F}$  and improved physical properties over silicones. Figure 34 shows a comparison of low-temperature capability in the form of percent retraction at various temperatures for fluorosilicone, Buna N, and Viton A (ref. 77). The percent retraction is the percent of initial ambient-temperature stretch that the material will recover after release when chilled to low temperatures. The 10-percent retraction level is considered to be the lowest temperature at which an elastomer functions as a resilient material.

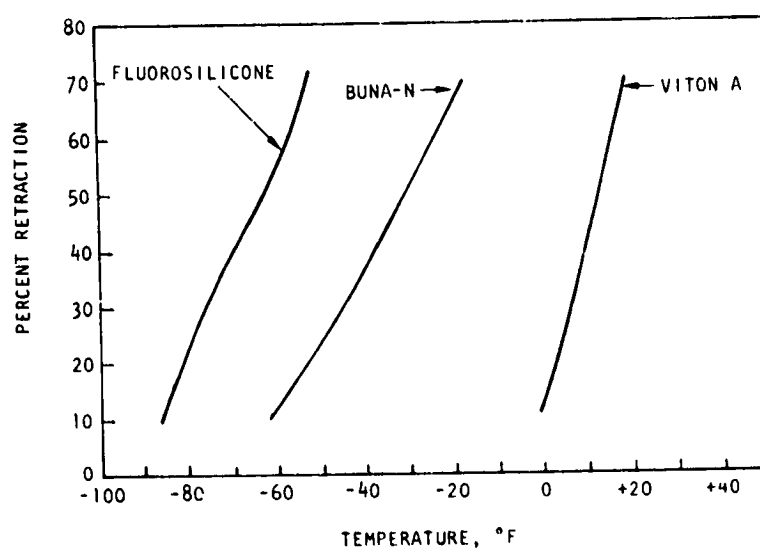


Figure 34. — Elastomer retraction at various temperatures (ref. 77).

At the other temperature extreme, high temperature, the elastomer decomposes and chars. The loss of physical properties and subsequent charring of the elastomer interface causes leakage in seal applications. Hardness of the elastomer is also reduced as the temperature increases, and thus the tendency of the elastomer to extrude increases. Figure 35 shows a seal life-versus-temperature chart for some of the common elastomeric materials.

Most temperature-related problems with seals are solved or precluded by making a thorough thermal analysis and by staying within the temperature limits of the material. Development of new elastomers has been helpful in solving some problems where temperature changes were brought about by extended engine operating conditions. One example of this is the use of fluorosilicone elastomers to overcome a problem due to long vehicle hold periods, wherein the cryogenic propellant chilled the boattail and caused fuel leaks past Viton A static seals. Development of reliable metallic seals with extended temperature capabilities to cope with both temperature extremes has eliminated the problem of using elastomers beyond their temperature limits.

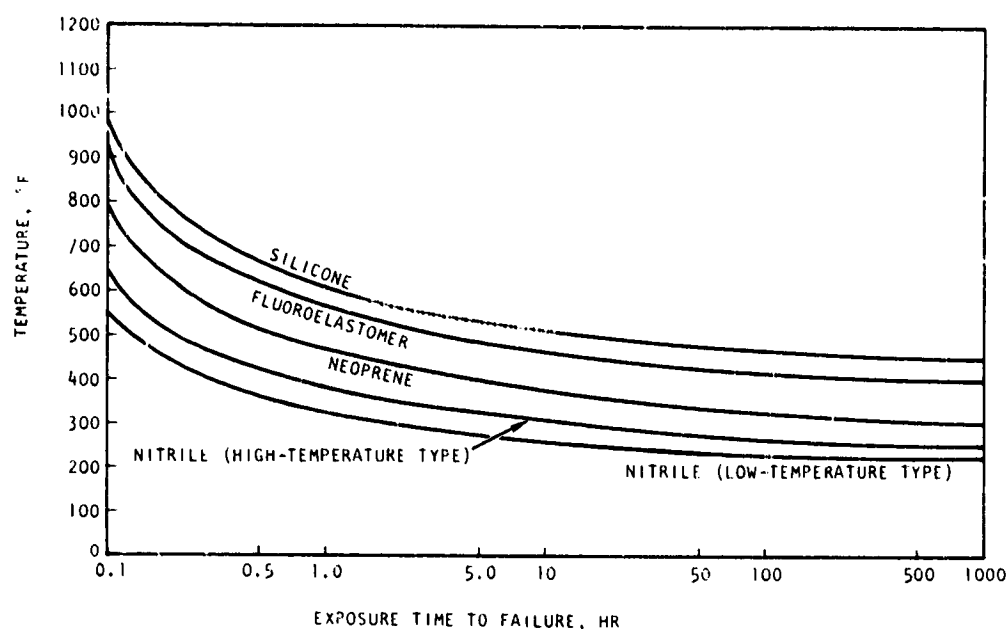


Figure 35. — Seal life as a function of temperature (various elastomers).

### 2.6.1.3 SEAL INTERFACE MATERIALS

Leak-free couplings are extremely difficult to obtain consistently with normal structural metals unless plastic flow occurs. This condition can occur only with high unit loading and subsequent damage to either the seal or flange, which limits the reusability of the coupling. To minimize this problem, various plastic coatings and soft-metal platings are applied to seals to provide a soft interface at the contact area. In operational usage, metal platings are employed from  $-452^{\circ}\text{F}$  to  $+1400^{\circ}\text{F}$ , while plastic materials are used from  $-452^{\circ}\text{F}$  to  $+300^{\circ}\text{F}$ . Table II lists the various coating and plating materials and their usable temperature ranges. The soft interface material on cryogenic seals is usually Teflon, while silver and gold generally are used on hot-gas seals. Additional information on interface materials is included in references 47 and 78 through 82.

Two problems occurred in silver-plating hot-gas seals on the J-2 engine program: blistering, and sticking of the silver to the mating flange. The blistering problem was resolved by a change in the plating procedures that incorporated a gold flash under the silver. Sticking was eliminated by the application of a rhodium over-plate on the silver (ref. 83).

Problems with Teflon generally are related to loss of bonding or excessive cold flow. These problems have been minimized by close processing controls and by a performance test to verify load-carrying ability (ref. 84).

## **2.6.2 Component Handling**

Handling protection is provided on all components to prevent damage and to maintain cleanliness level either for subsequent processing or for preventing contamination during assembly. Handling abuse is critical for the sealing surfaces of disconnects, couplings, and seals, while cleanliness is important to all components for maintaining clean fluid systems. On fittings and joints, cleanliness is required in the joining operations.

Handling abuse of components early in the J-2 and F-1 engine programs resulted in a large number of leaks at couplings during engine leak and functional tests and resulted in fires during engine hot-firing tests. These problems were minimized by using protective containers for seals and protective covers for flanges (fig. 36). Cleanliness is maintained by enclosing parts in heat-sealed plastic or by using protective covers. Even small particles of plastic are incompatible with some propellants (e.g., fluorine). Therefore, material such as scuffless nylon must be used for packaging components. A principal consideration in the design of any protective device is that it should be impossible to accidentally install any portion of the protective material with the component.

## **2.6.3 Evaluation Testing**

Testing on the subject components consists of development testing and fabrication testing. Development testing verifies that the component is capable of meeting all of the design functional and structural requirements. Fabrication testing verifies that each item meets the required performance and has the required structural soundness.

### **2.6.3.1 DISCONNECTS**

Conventional static-test techniques are used for evaluating the valves and the manually operated disconnect mechanisms in disconnects. References 5 and 6 discuss valve testing. Dynamic tests on breakaway and forced-separation disconnects, however, require sophisticated test equipment to simulate operation conditions. These tests consist of separation under simulated flow, pressure, and temperature conditions as they exist in operation (ref. 85). Since failure to separate properly during launch would jeopardize the vehicle, simulated tests during the development program are essential. Test conditions must be controlled to prevent unrealistic transverse loads caused by erroneous test setups that may distort data and may result in leakage past the interface seal or mechanical malfunctions in the latch-release mechanisms.

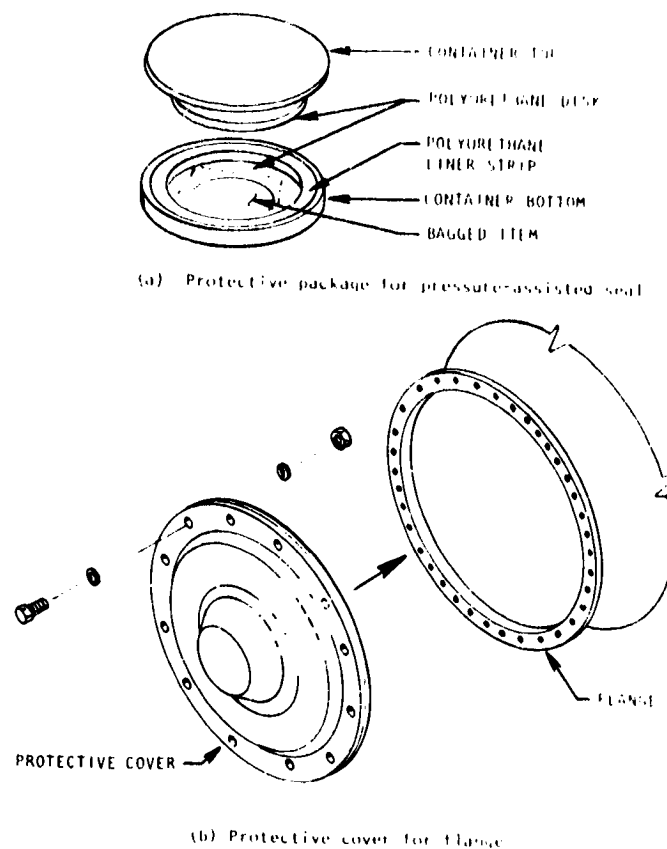


Figure 36. — Typical devices for protecting flanges and seals.

### 2.6.3.2 COUPLINGS AND SEALS

Testing is performed on couplings and seals to discover leakage and structural problems prior to their occurrence during the engine development phase and to eliminate coupling leaks on production engines prior to delivery. A new coupling seal design is evaluated as a component in simulated flight-type test fixtures to determine sealing ability and mating-flange requirements and, on dynamic couplings, friction and cycle-wear life. The evaluation tests include low-pressure (ambient) leak checks; operating pressure at applicable temperature; proof and burst tests; pressure cycling at both operating and surge pressures while at application temperature; and tolerance-extreme seal coupling tests. Laboratory evaluation tests have been successful in simulating most engine conditions; however, there is

a problem in simulating the thermal conditions on hot-pac couplings. This problem has been resolved by evaluating the coupling's thermal environment during engine operation. On both the F-1 and J-2 engine programs, static couplings were successfully evaluated during engine operation. This evaluation was possible because of the design requirement to provide the capability to monitor leakage during ground testing or flight. Figure 17 shows typical elastomer-seal/coupling and pressure-assisted-seal/coupling designs used for this evaluation. References 28, 29, and 30 report the results of the F-1 and J-2 leakage-monitoring programs.

Leakage-measuring techniques include the collection and measurement of fluid over a given time (sight-glass or immiscible-fluid displacement); use of flow meters (glass capillary tubes or heated thermocouple); and mass-spectrometer leak detection. Figure 37 shows typical

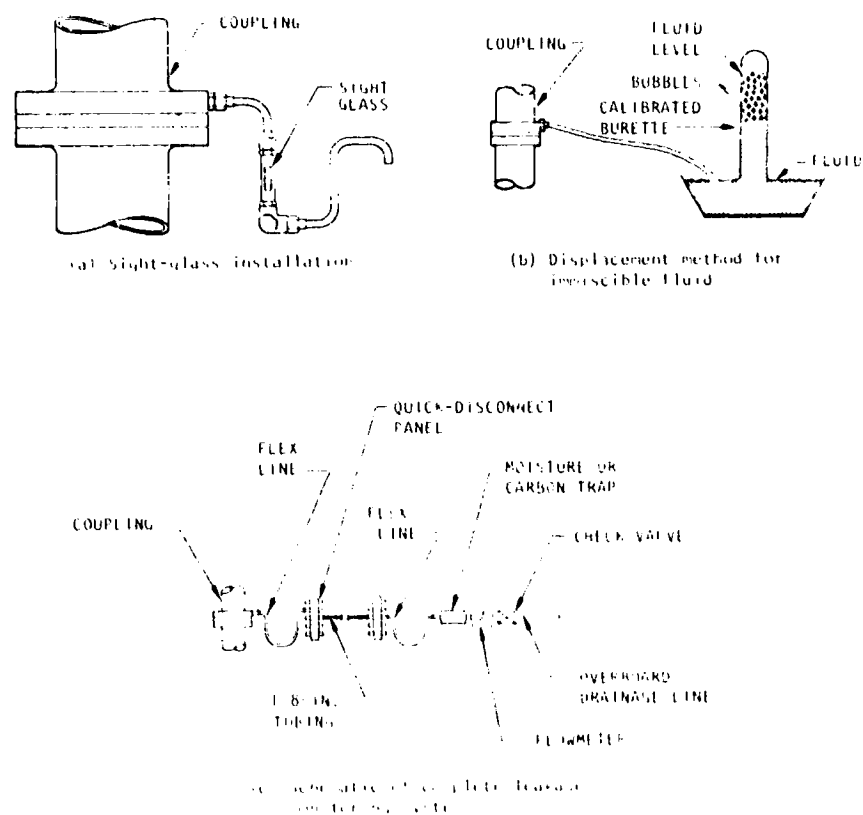


Figure 37. Typical leakage-monitoring systems.



leakage-monitoring systems. References 12, 27, and 86 provide in-depth discussions on leakage monitoring and testing. Thermocouples, strain gages, and accelerometers are utilized for measuring parameters to aid in thermal analysis, strain levels, and vibration-responses on problem couplings during both laboratory and engine firing tests. Data obtained from instrumentation have been used to change the coupling design and eliminate leakage problems.

### **2.6.3.3 FITTINGS**

Pressure-loss measurements are the primary data accumulated in testing of a fitting design. These data may be limited to simple upstream and downstream static pressure measurements taken at various flowrates to establish a loss coefficient for a particular design. The basic pressure-loss measurements may be expanded to include mapping (by means of an array of pitot-tube samplings across the flow passage) the velocity distributions associated with certain fitting designs.

Temperature-distribution measurements have been useful in some instances. For example, temperature-distribution measurements were made in an effort to establish an effective design for a manifold-type fitting carrying a cryogenic fluid through a hot gas duct on the F-1 tank pressurization heat exchanger (refs. 87 and 88). These data were used to alter the design so that thermal gradients were minimized; thus stresses were reduced and fatigue cracking was precluded.

Analog-type model testing has been effective in studying certain flow phenomena in hot-gas tapoff ports, nozzle-extension ports, flexible hoses, and similar fittings. Two-dimensional, large-scale models were built to contain open-channel flow of water pumped across a flat glass surface. Photographs taken through the glass of tracers in the water or on the water's open surface showed flow direction of the water inside the model boundaries. This type of test can be effective in establishing guide-vane configurations for fittings with unusual shapes. References 89 and 90 describe some of the applications and capabilities of this type of testing.

Laboratory tests on fittings usually are conducted to determine pressure loss or flow distribution. The preferred test is one in which the fittings perform their full function in the system. A hot-fire test thus is the most meaningful type of test because almost all elements of the operating environment are present, and accurate measurements can be obtained.

### **2.6.3.4 FIXED JOINTS**

Fixed-joint designs are tested initially to verify the design concept and then to see if the joint meets all requirements for the particular engine program. These tests include destructive examination of sample joints, nondestructive tests such as penetrant and X-ray

inspection; proof and burst tests; environmental exposure tests; vibration; and pressure cycle fatigue tests. Reference 41, a detailed report on the development of brazed and welded joints, defines the various tests.

Nondestructive testing is performed on joints to locate joint defects while production joints are being fabricated. The types of defects that may occur in a fixed joint depends on the method of fabrication. Welded joints are subject to defects such as incomplete or insufficient weld penetration, porosity, and cracking of weld bead or parent metal. Brazed joints are subject to insufficient flow of braze alloy, lack of adhesion (poor wetting), entrapped flux or oxides, and void areas. The process defects usually occurring in diffusion-bonded joints are not known. However, the bond line is subject to cracking from bending or from heating to high temperatures, which may occur with welding and brazing nearby.

Nondestructive test methods used to determine the quality of production joints include the following (refs. 42, 91, and 92): penetrant inspection, X-ray inspection, ultrasonic inspection, mass spectrometer leak detection at system operating or proof pressure, and soap-solution leak testing at system operating or proof pressure. One or more of the first three techniques are used immediately after a production joint is completed. One of the pressure test methods may also be used on completion of the joint. However, where a high confidence level in joint fabrication is achieved, joint pressure testing is incorporated in the final pressure test.

The main problem arising in testing joints has been one of inadequate consideration of the limitations of test procedures in the initial design of the joint. For example, ultrasonic inspection is an excellent method for inspecting sleeve joints. However, allowance is rarely made for effective use of the ultrasonic probe; therefore, this test technique has been misleading. Another example is that of expecting penetrant inspection to detect a defect that would result in bubble formation during a helium-gas leak test.

### **3. DESIGN CRITERIA and Recommended Practices**

#### **3.1 DISCONNECTS**

*The disconnect system design shall be an optimum for the particular space-vehicle umbilical and service system.*

The design of disconnect assemblies must be integrated with other considerations to arrive at the design of an umbilical or service system for a particular space vehicle. The disconnects in such systems must be reusable; provide for minimum refurbishment; provide for easy and rapid engagement and verification of engagement; minimize ground turnaround time for connection, verification, and checkout; and be inherently reliable. In the choice of the proper disconnect configuration for a system, detailed consideration must be given to the special characteristics and unique features of umbilical systems. References 93 and 94 are recommended for review and study of good umbilical-systems design.

##### **3.1.1 Configuration Design**

*Both halves (flight and ground) of a disconnect assembly shall be compatible in all modes of disconnect operation.*

Determine all interface requirements. Recognize all operating conditions in both the coupled and uncoupled modes. Closely coordinate design effort among functional organizations and, when required, among contractors. Submit all proposed changes in performance requirements and in operating procedures for appropriate reviews prior to release.

##### **3.1.1.1 MANUALLY OPERATED DISCONNECTS**

*A manually operated disconnect shall include safety features adequate to protect personnel during coupling or uncoupling.*

When line pressure is high, use a pressure-balanced disconnect to reduce the potential hazard caused by the propulsive effect of the discharging gases. When the application is extremely hazardous because of high pressure or unusual fluid properties, the check valves (or disconnect closure valves) should require a separate manual operation to open and close; and this operation should not be performable unless the latching mechanism is engaged. In moderately hazardous applications, the latching mechanism should require a deliberate

action before it can be disengaged (e.g., push in and twist). When possible, provide visual indication that the latching mechanism is fully engaged.

During the connecting operation the interface seal should be engaged before either check valve starts to open; and during the disconnecting operation the interface seal should remain engaged until after both check valves are closed. This sequence is illustrated in figure 1.

Prepare clear and explicit written procedures describing each step to be followed, in sequence, for connecting and disconnecting operations. These procedures should ensure that connecting and disconnecting operations are performed after the hardware has returned to safe handling temperatures and after internal system pressure has been vented to a safe level.

Quality assurance verification should be required as necessary to ensure proper adherence to procedures.

### 3.1.1.2 BREAKAWAY DISCONNECTS

*A breakaway disconnect shall provide maximum simplicity, safety, and functional reliability.*

Make and secure the ground-to-vehicle interface connections manually under ambient conditions, with no fluid in the system. The ground-to-vehicle interface connection should not be disturbed prior to launch. Panel mount disconnects where feasible to simplify and reduce individual latching requirements. Latching mechanisms should be positive and simple. Redundant latching and separation systems should be provided on panel-mounted disconnects to increase reliability. Any limitations or tolerances for misalignment between the disconnect halves should be spelled out in procedures.

All cryogenic disconnects should be of the slip type with dual self-forming lip seals. A tertiary seal adjacent to the dual seals should be provided to contain a gaseous purge. The gaseous purge prevents cryopumping and ice buildup on the sliding seal surface.

Cryogenic disconnects should not require the application of additional insulation after mating. The volume between the dual lip seals should be vented through a tubing connection on the ground side. This vent tubing should then be monitored for leaks during verification of the connect phase when the disconnect is tested by pressurization internally with gaseous helium. The disconnects should also provide for the mounting of leak-detection devices.

The mounting provisions for the ground half of the disconnect and the attached flexible duct should allow lateral and angular motion with respect to the ground carrier to ensure that the disconnect halves will align during engagement and disengagement. The vehicle half of the disconnect should be rigidly attached to the vehicle.

Breakaway disconnects for spacecraft staging should require low force levels for disconnection, and disconnects on propellant lines should have integral check valves in both halves. These disconnects have unique requirements that require close study of all forces involved in order to avoid altering a carefully planned trajectory by disconnect operation.

### 3.1.1.3 FORCED-SEPARATION DISCONNECTS

*A forced-separation disconnect shall incorporate safety provisions to protect personnel, a redundant mechanical latch-release mechanism, and adequate mounting structure.*

With explosive-actuated disconnects, prescribe appropriate precautions for handling explosive charges. Do not install explosive charges until final preparations for use are being made. Use a device such as a shear pin, which requires a high force to rupture, in the latch mechanism to prevent premature disengagement.

A redundant release mechanism should be used to ensure separation. For example, an explosive-actuated disconnect could utilize controlled gas pressure acting on an actuating piston, and a pressure-actuated disconnect could use either an explosive charge acting on an actuating piston or a mechanical release mechanism operated by a lanyard. Both the primary and the redundant release systems should be considered in the design of all structural members.

Consider the use of pressure-balanced disconnects wherever possible, because they may result in lower weight and reduced hazard to personnel.

## 3.1.2 Design Integration

### 3.1.2.1 VALVES

*Valves in disconnects shall satisfy the leakage, spillage, and pressure-drop requirements of the disconnect application.*

The selection diagram below (fig. 38) shows the primary factors to be considered and the valve configurations recommended for various design requirements.

The following factors also should be considered in selecting the final valve configurations: fluid compatibility; temperature (storage and operating); endurance requirements; actuation (pressure, as in a conventional check valve, or mechanical, as in the connecting operation); safety devices (e.g., interlock between valve and latch); contamination and handling-damage potential; availability; and cost.

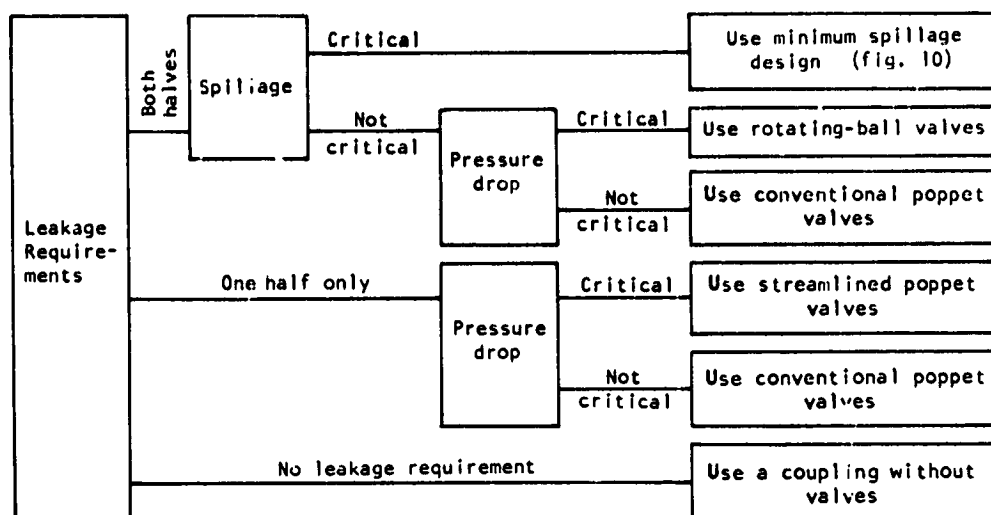


Figure 38. -- Sciection diagram illustrating recommended valve configurations for disconnects.

Pressure-actuated valves should be designed for minimum chatter under all flow conditions to which they may be exposed in service. Techniques for preventing chatter are presented in reference 5.

References 5 and 6 are recommended for in-depth discussions on valve technology.

### 3.1.2.2 INTERFACE SEALS

*Interface seals shall function, as required, to control external leakage within acceptable limits during all modes of disconnect operation.*

Sections 2.5 and 3.5 present information and recommendations on seal configurations and seal design integration.

In general, design features should ensure that the interface seal is engaged during valve actuation when the disconnect is being coupled or uncoupled. A detailed tolerance study to confirm engagement is recommended when valve actuation is accomplished by the coupling/uncoupling action. An alternate recommended design technique is to require a separate operation for valve actuation while the interface seal and latch mechanism are engaged.

On the vehicle half of manually operated disconnects, use protective covers (see 3.6.2) that utilize the same latching mechanisms and interface seals as the ground halves and are capable of withstanding full system pressure (in case a check valve leaks) as well as the vehicle vibration loads. Protective covers should also be provided for ground half of disconnects for protection during periods of non-use and exposure to engine exhaust during launch.

### 3.1.2.3 LATCH/RELEASE MECHANISM

*Latch/release mechanisms shall ensure proper release while preventing inadvertent separation of disconnects.*

Use a latching mechanism such as a spring-loaded push-pull or twist latching collar on manually operated disconnects and provide means for visual verification of proper latch engagement. To prevent inadvertent separation of disconnects (e.g., by vibration), modify the push-pull collar to require an initial twist before separation can occur. The latch mechanism should be designed to withstand any operating pressure loads to which it may be exposed, and a protective cover should be used as a redundant seal to withstand full system pressure on the cover if the valve leaks.

When disengagement occurs at low temperatures, use a flexible shroud to protect the latch/release mechanism against condensation and freezing. It may be necessary to provide a carefully designed purge of the shroud over a cryogenic disconnect to preclude formation of ice or condensation under the shroud.

A ball-type latching device is recommended on breakaway disconnects. Alternate choices are positive-actuated pawls and break-bolts. Clustering of small disconnects and installation of latch/release mechanisms on the umbilical panel are recommended. Individual self-contained latch/release mechanisms should be provided on large disconnects. If the design permits, large disconnects may also be retained on an umbilical panel because the panel release mechanisms are fewer in number and less costly than multiple individual disconnects. Use redundant, lanyard-operated release systems for the latching device.

Not more than one latch/release mechanism per panel should be used.

### 3.1.2.4 MOUNTING

*Disconnect mounting shall provide system compatibility and ease of coupling and uncoupling.*

Locate manually operated disconnects at the point of application. Disconnects on systems that are incompatible or that could affect one another adversely should not be located close



together. When a system requires use of multiple, manually operated disconnects of the same size, prevent erroneous cross-connections by making it physically impossible to interchange ground halves and vehicle halves used in different systems. Use of incompatible sizes or unique arrangements of keys in ground halves and matching keyways in the vehicle halves are recommended techniques.

Breakaway disconnects should be independent of each other, with adequate space between them for mounting simplicity and operational reliability but clustered on umbilical panels when feasible. Temperature and fluid compatibility and the effects of interface seal malfunction should be considered in clustered installations. Mount vehicle halves rigidly and provide individual adjustments for the ground halves of breakaway disconnects to ensure proper alignment of interface seals and freedom from binding during separation. Accumulation of adverse tolerances, both axial and transverse, should be taken into account in designing the adjustments. Mechanical guides should be used to limit angular displacement and to absorb transverse loads during separation. Transverse loads on the cluster panels should be limited to allowable levels.

### 3.1.2.5 SPILLAGE CONTROL

*Spillage shall be a safe factor below the maximum amount tolerable for the application.*

Limit the volume between the valve seats and the interface seal in the coupled position, consistent with other design objectives. When spillage is very critical, employ a design concept as shown in figure 10 and discussed in reference 9.

## 3.2 COUPLINGS

*The number of couplings shall be a minimum consistent with the propulsion system requirements, and the couplings themselves shall introduce a minimum of system problems.*

Make an initial design-phase evaluation of each proposed coupling to justify its existence (e.g., justify a requirement for component removal or for maintenance access). Only after this evaluation should a coupling be added to the propulsion system. Couplings originally provided to allow easy removal and installation of components under the engine development phase should be eliminated after they have served their purpose and the engine enters into a production phase.

Minimize coupling problems by using the fewest number of different types. In selecting couplings for a given system, consider all possible combinations of operating pressures and

temperatures, line sizes, wall thicknesses, line materials, and applied loads. Design the couplings with specific objectives in mind:

- Integrity of the coupling must be verifiable before flight.
- Coupling must contribute minimum weight to the system.
- Coupling must achieve acceptable service life and maintainability.
- Coupling must be reusable.

Some specific guidelines for design are as follows:

- Minimize the number of leak paths.
- Improve reliability by using dual seals in series.
- Minimize the possibility of generating contamination in the line by error in the installation process.
- Minimize the possibility of installing the wrong coupling on a given tube or duct.
- Avoid separate sealing elements where possible, because they can be omitted or improperly installed.
- Minimize dependence on critical finish conditions of sealing surfaces.
- Protect sealing surfaces from handling or installation damage by providing shoulders, recesses, and similar configurations as required.

Additional discussion of the problems encountered in coupling design and of the various tradeoff considerations is included in reference 15.

### **3.2.1 Configuration Design**

#### **3.2.1.1 FLANGED COUPLINGS**

*Flanged couplings shall withstand high vibration, high pressure, and extreme temperature and yet shall be lightweight and reliable.*

Multi-bolted flanged couplings are recommended where high loads are imposed by vibration, pressure, or temperature. Sufficient bolts should be provided to prevent flange bowing and

separation. The coupling should be designed to prevent flange rotation by providing sufficient duct wall thickness, flange rigidity, and flange contact outside the bolt circle. For extremely-high-pressure couplings, flange separation and rotation requirements can be relieved by designing the point of flange rotation as close to the point of sealing as possible. To select the proper seal for flanged couplings, follow the recommendations in section 3.5.2. Complete analysis of flanged coupling design should include effects of temperature, pressure, vibration, and fluid properties. To ensure reliability, couplings should be tested at operating conditions. The endurance cycles and exposure times for qualification testing of couplings should be four times the flight-life requirements.

### **3.2.1.2 THREADED COUPLINGS**

*Threaded couplings shall provide sealing capability without detrimental distortions under operating conditions.*

Threaded couplings are recommended only for small-diameter lines where accessibility and weight do not permit a flanged coupling, or where leakage requirements are not critical. Couplings that utilize either pressure-assisted or radial seals with parallel load paths are recommended. However, seals should not restrict flow or cause contamination of the fluid system.

Use locking features to prevent nut backoff and loss of torque.

Standard commercially available threaded couplings are acceptable in low-pressure, small-diameter lines ( $< 0.5$  in.) where vibration is not of prime concern. They are generally not recommended for cryogenic systems.

### **3.2.1.3 DYNAMIC SWIVEL COUPLINGS**

Swivel couplings are not recommended for rocket fluid systems.

## **3.2.2 Design Integration**

### **3.2.2.1 SEAL CONTACT SURFACE**

*The seal contact-surface condition and tolerance extremes shall be suitable for the seal material and seal loading.*

It is recommended that the groove and surface-finish requirements in the literature of the seal supplier be specified for the seal interface. In addition to the supplier's requirements, the

sealing surfaces should be clearly identified on the drawing. Coded to the sealing surfaces should be a note that specifies that the sealing surface shall be free of nicks, scratches, tool marks, and similar imperfections. Drawings that define sealing surfaces should specify the use of protective covers to prevent handling damage. A tolerance study similar to that presented in reference 22 is recommended for all couplings to ensure proper seal interface and assembly requirements in tolerance-extreme conditions. This study should include consideration of seal/flange shift, coupling restraining mechanism, and seal alignment.

### **3.2.2.2 RIGIDITY**

*The coupling shall be sufficiently rigid to provide adequate seal loading at all operating conditions.*

Coupling deflection at the seal contact area should be calculated for the most severe thermal condition to ensure that the deflections are within the limits of the sealing element. Consult reference 24 for procedures. Deflections are usually related to flange mass; therefore, for lightweight ducting systems, seals capable of compensating for large flange deflections are recommended. Flange-separation values for various seals are shown in table 1.

A large number of bolts is recommended to reduce flange bowing. The seal diameter and bolt circle should be as small as possible to minimize coupling separation loads that will subsequently reduce flange rolling. The sealing element should be located as close to the bolt circle or constraining point as possible since flange deflections are the smallest at these points.

There is no simple procedure for overcoming or compensating for the effect of adverse thermal gradients. Each problem must be reviewed to determine the design solution. Possible solutions include the use of a material with high strength at high temperature (Inconels or Hastelloy C) or an insulating material (sprayed-on ceramic) to reduce the temperature gradient.

### **3.2.2.3 RESTRAINING MECHANISM**

*The preloaded restraining mechanism shall prevent coupling separation under all service conditions.*

The preload should be large enough to limit flange separation due to either loads or thermal conditions to the ability of the seal to follow the deflection (sec. 3.2.2.2). A detailed load analysis to determine the loads imposed on the coupling should be performed. This study should include both internal and external loads resulting from pressure, vibration, misalignment, thermal, and other sources. These loads should be used in the design of the

restraining mechanism and the component parts of the coupling to ensure that the stresses are within the yield strength of the material.

Bolts are generally recommended for flanged couplings. Ring clamps, while acceptable for low-pressure ( $< 500$  psi) use, are not recommended for high-pressure couplings.

### 3.2.2.4 LEAKAGE MONITORING

*Leakage-monitoring provisions shall make it possible to collect and measure leakage, and shall do so with a minimum increase in coupling size and complexity.*

Figure 17 shows recommended monitoring provisions. Tolerance studies should be performed to verify that the monitoring provisions do not result in coupling leakage and that the monitoring cavity is not blocked. To reduce hardware complexity and cost, monitoring provisions should be deleted after the joints have been verified.

## 3.3 FITTINGS

### 3.3.1 Configuration Design

#### 3.3.1.1 TUBE FITTINGS

*Tube-fitting configurations shall be as simple as possible and shall not result in unacceptable pressure loss in the tube.*

The basic recommendation is to use a one-piece fitting machined from a bar or a forging. The tubes should be welded to the fitting with a radiographically inspectable butt weld. The butt-joint intersections should be concentrically aligned to prevent discontinuity in flow passages and attendant energy loss. Tube welds should be specified as 100-percent penetration for structural reasons; but only minimum droptrough should be allowed in order to keep discontinuities to a minimum at the inner surface of the flow passage.

Where pressure loss is critical, elbows should be optimized by techniques such as those given in references 19, 31, and 95. In applications where the flow paths are necessarily more complex, departure from the one-piece design is frequently warranted. For example, figure 39 shows an "H" fitting recommended for consideration where two inlet tubes of different diameters intersect with two outflow tubes. In this case the flow from either tube to the two outflow tubes is made with minimum pressure loss. Use of diffusers aids in minimizing pressure loss.

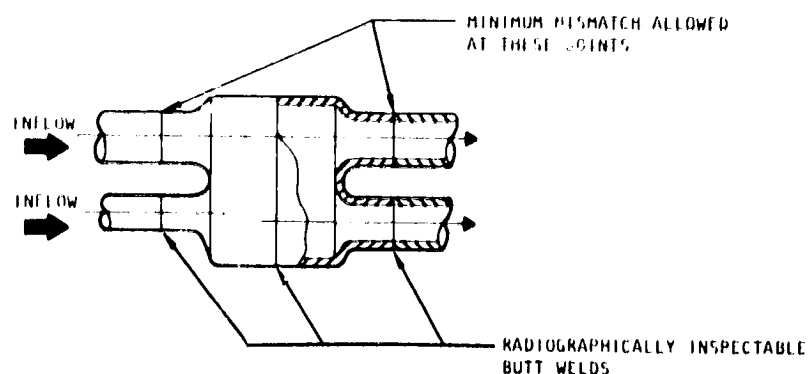


Figure 39. — Recommended design for low-pressure-loss fitting for joining tubes of different sizes.

When tubes are required to tap flow from high-velocity flow in ducts and do so at low pressure loss, the tapoff (fitting) should be designed to take flow at (or near) the main-flow total pressure. Recommended designs for accomplishing this are tapoffs that protrude upstream along the centerline of the duct flow or take off at a shallow angle to the centerline of the main flow. Figure 40 shows examples of these configurations. References

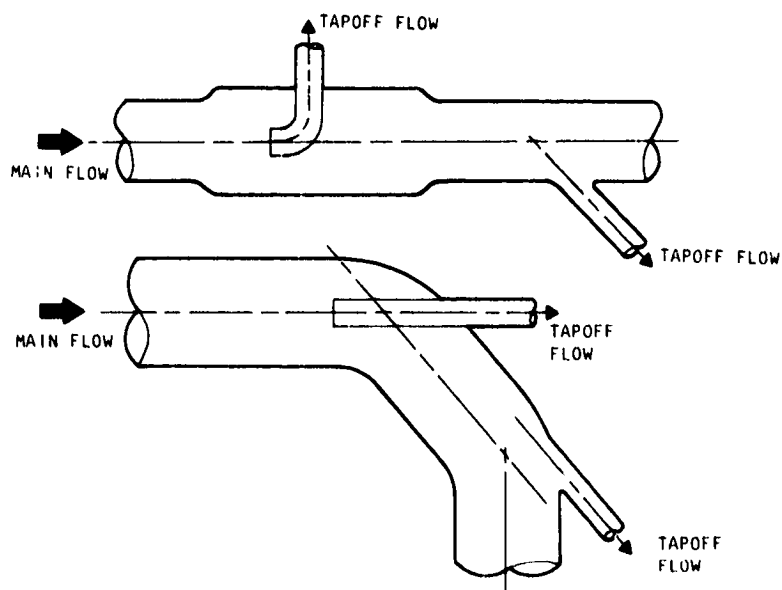


Figure 40. — Recommended designs for tapoff fittings.

96, 97, and 98 are recommended sources of data on pressure-loss characteristics for specific configurations.

Tubes protruding into a flow stream are very susceptible to flow-induced vibration. One possible fix for this is to use a tapered probe. Vortex shedding thus is suppressed since the shedding does not occur along the entire length of a tapered tube at any given flowrate.

Fittings should be free of fluid traps that could collect cleaning fluids, impurities, or propellants and subsequently result in corrosion, violent reactions, or production of toxic fumes. To the greatest extent possible, tube assemblies and fitting designs should provide for self draining in any attitude.

### 3.3.1.2 DUCT FITTINGS

*Duct-fitting configurations shall exhibit the minimum pressure drop practicable in the envelope allowed, and the fitting structural strength shall be adequate for all flow-induced and mechanically induced loads of the application.*

It is recommended that a thorough analysis be made of the operational flowrate range of the fitting. From this, a range of operating Reynolds numbers should be determined. Loss coefficients for fittings are characteristically presented as functions of Reynolds number and geometry. When the operating flow regime is known, the most efficient geometry that fits the envelope can then be selected. When pressure drop is critical, perform pressure-loss flow tests on the duct assembly; most fittings will behave differently from the performance indicated in published data, especially if other fittings are close to them, either upstream or downstream.

Fitting configurations should have smooth internal flow surfaces and changes in area and direction of flow that are as gradual as possible. Flow-control devices (ref. 19) should be considered for use when it is necessary to reduce losses on tight turns and large changes in area.

Welded joints should be located in areas with smooth transition of wall thickness to prevent thermal structural problems. Boss and mounting-lug attachment configurations that provide maximum structural capability without fluid traps are described in reference 19. Consideration should be given to the fabrication tooling that will be required to effect proper alignment for welding.



### **3.3.1.3 VACUUM-JACKETED FITTINGS**

(Design criteria and recommended practices for vacuum-jacketed fittings are presented in reference 19.)

### **3.3.1.4 FLOW-CONTROL DEVICES**

(Design criteria and recommended practices for flow-control devices are presented in reference 19.)

## **3.4 FIXED JOINTS**

### **3.4.1 Welded Joints**

*A welded-joint configuration shall be consistent with the system requirements for strength, assembly sequence, and cost.*

Use bench-fabricated welded joints wherever possible to reduce cost and ensure quality. Butt welds are most desirable because of superior fatigue strength, resistance to corrosion, and ease of radiographic inspection.

Where closeout joints (i.e., final welds that seal the system) are required on installation, an in-place sleeve weld is recommended. This type of weld eliminates tube trimming to close tolerances. Provide clearance about the exterior of the joint for the welding equipment.

Details for joint design are provided in references 40 and 41.

#### **3.4.1.1 WELDING METHODS**

*The welding method shall be consistent with the joint configuration and the system fabrication requirements.*

It is recommended that joints be designed to tolerate the greatest possible variation in welding conditions without sacrifice of joint-to-joint reproducibility. Whenever possible, design for the use of GTA or GMA welds, the choice depending on the wall thickness of the tubular components. If extreme accuracy of the joint is necessary, a small weld bead is required, or the heat input at the weld must be minimized, consider EB welding for achieving the desired control of welding conditions.

Resistance welding must be ruled out as an acceptable joining method for rocket engine systems because of the large number of fatigue failures that have resulted from poor fitup of parts and crevice corrosion.

### **3.4.1.2 PROBLEM AREAS**

#### **3.4.1.2.1 Vibration**

*Welded joints shall withstand the vibration environment of the application without loss of joint integrity.*

Welded joints should be designed for adequate fatigue life on the basis of an analysis of the alternating stresses developed by the predicted operational vibration spectrum (ref. 99). Vibration damping clamps, strategically located along tube assembly length, should be used to attenuate vibration input to welded joints.

#### **3.4.1.2.2 Weld Oxidation**

*Internal oxidation due to welding shall be at a minimum.*

Prevent oxidation on the inside of tubular joints during welding by making through-holes in both components being welded to provide adequate circulation of inert gas on the backside of the joint. The final welds in a system may require the use of small ( $\approx 0.065$ -in. diameter) vent holes in the joint; these holes are welded shut after the joint weld is made.

#### **3.4.1.2.3 Contamination in Crevices**

*Welded joints shall be free of crevices on both the inside and outside of the joint or the effects of such crevices shall be minimal.*

The design of butt-welded joints and the welding process (GTA, GMA, or EB) should ensure a full-penetration weld. Thick wall sections should be EB welded, or the designer should specify (or show) a joint preparation (such as a "V" or "U" groove) suitable for GTA or GMA welding. When sleeve welds are used, fillet welds should be made on the outer surface between the end of the sleeve and the tube (fig. 19(f)). This procedure eliminates the external crevices; the effects of the internal crevices must then be minimized through careful cleaning before and after welding.

#### **3.4.1.2.4 Tolerances**

*The component and joint tolerance stackup shall ensure reproducible quality in welded joints*

Automatic welds should be used in making propellant-system joints. Ensure the location of an automatic weld by controlling component part and total joint tolerance within definite limits. The critical dimensions requiring control are as follows: (1) butt-welded joints: diametrical mismatch between components, and normality of the ends of the tubular components; (2) sleeve-welded joints: the gap between the sleeve ID and the OD of the inserted component, and the normality of the tube ends; and (3) fillet-welded joints: normality of the tube end. The values for critical dimensions should be developed by testing sample welds made on the equipment that will be used for making production welds.

#### **3.4.1.2.5 Weld Preparation**

*The surfaces and edges to be joined shall accept the weld material and shall not cause cracks or weld porosity.*

Foreign matter such as cutting oils and material from forming dies should be removed from the surfaces of component parts prior to welding. The type of cleaning necessary should be determined from the fabrication history of the components and the metal alloy used for the components. Cleaning procedures should be detailed on the drawing or should be included in a specification referenced on the drawing. For components that are to be EB welded or that use a sleeve weld, specify a deburring procedure either on the drawing or in a specification referenced on the drawing. Some welding processes require surface preparation (including passivation) that should be specified on the drawing or in a specification referenced on the drawing.

#### **3.4.1.2.6 Wall Thickness and Material Composition**

*The wall thickness and material composition of the components to be joined shall be suitable for the welding process.*

At the joint, the wall thicknesses of tubular components should be equal within normal machining tolerance limits. Tube sections should not be joined to a bulky fitting with a fillet weld. Rather, the fitting should be designed to include a tube-like projection having a wall thickness approximately the same as that of the tube to be attached. Except for GTA or GMA welding of CRES alloys to nickel-base alloys, avoid joining dissimilar metals by welding. When CRES alloys are to be welded to nickel-base alloys, a filler wire compatible with the two materials must be specified on the drawing or in a specification referenced on the drawing.

### 3.4.1.3 WELD REPAIR

*Weld repair procedures shall be appropriate for the original weld and consistent with design objectives for integrity of the joint.*

Cracks and seam-like defects occurring in or adjacent to a weld should be either ground out and rewelded, or the components should be refurbished and then rewelded in accordance with the original weld procedure. Porosity can be eliminated by rewelding if it is not severe. Severe porosity should be reworked in the same manner as crack and seam-like defects. Oxides and other inclusions should be evaluated both for loss in joint strength and for reaction with propellant; then, if a repair is required, the defect should be removed by grinding and the joint rewelded. In any repair welding, the increased size of the heat-affected zone in the parent metal must be evaluated for loss in overall joint strength.

## 3.4.2 Brazed Joints

### 3.4.2.1 BRAZING METHODS

*The method of brazing shall be suitable for the material and configuration of the joint.*

Dip brazing should be used for joining aluminum parts. The equipment size must be considered when designing an aluminum brazed joint. Joints designed for dip brazing should not contain cavities, gaps, or blind holes that will trap salt from the dip braze bath. When subsequent on-site or field installation of an aluminum brazed joint is required, a joining method other than brazing should be considered.

Furnace or induction brazing should be used for metal alloys other than aluminum. Equipment size should also be considered here when designing the joint.

In all cases, the braze alloy should be carefully pre-placed to ensure good flow.

### 3.4.2.2 JOINT PREPARATION

*The surfaces to be joined shall accept the braze material and shall not cause joint structural weakness; the braze material shall not introduce constituents that will cause joint voids or weakness.*

All foreign matter such as cutting oils and material from forming dies must be removed from the component-part surfaces prior to brazing. A chemical cleaning method is recommended

for braze-joint components; the type of cleaning depends on the material used for the components. Cleaning should include the braze alloy when it is used in the form of pre-placed strips or rings. When braze alloy wettability or flow characteristics are marginal on CRES or nickel-base alloys, surface treatments such as shot peening or nickel plating of the surface to be brazed should be considered. Pre-braze surface treatments and cleaning procedures should be detailed on the drawing or should be included in a specification on the drawing.

#### **3.4.2.3 BRAZE-ALLOY CHARACTERISTICS**

*The braze alloy shall provide wettability and flow characteristics adequate to ensure high-quality joints.*

The selection of braze alloy should be based on a thorough study of the ability of the braze alloy to provide proper wettability and flow for the joint components. Consideration should also be given to the requirement or possibility that a rebraze or braze repair may be required; if required, the selected braze alloy should not dissipate or deteriorate during the initial braze cycle. References 9 and 43 are recommended for use in selecting a braze alloy.

#### **3.4.2.4 DIMENSIONAL CONTROL**

*Dimensions and locations of braze-joint components shall ensure post-brazing joint integrity.*

Dimensions and tolerances on braze-joint components should be controlled to the extent that the available braze alloy will fill the gap provided. The joint should also be designed so the braze alloy is primarily in shear when system loads are imposed. In applications subjected to high vibration or bending loads, the ends of the joint should be tapered or stepped, and the braze alloy should be stopped short of the edge of the joint so that stress concentrations are relieved. References 9 and 43 are recommended for detailed information on clearances between joint members and the relation between the joint and braze area. On critical joints, provisions should be made for nondestructive testing.

### **3.4.3 Diffusion-Bonded Joints**

*Diffusion-bonded joints shall be accomplished with proven techniques that provide for control of bond line temperatures and pressures within specified limits.*

At the present time, the diffusion-bonding technique is recommended only for titanium-to-stainless steel joints.

It is recommended that the bond line of a diffusion-bonded joint be located and suitably identified before subsequent fabrication steps. Subsequent system joining operations such as welding or brazing should be dimensioned from the bond line, and should be located sufficiently far from the bond line so that the heat from the joining operation will not heat the bond line above the 1200°F temperature limit. When space limits the length of the diffusion-bonded joint to less than optimum distance from welding or brazing operations, a bond-line cooling method (e.g., a chill bar) should be specified on the assembly drawing. Joint suppliers should be contacted for necessary information pertaining to the tube length requirements.

### **3.4.4 Soldered Joints**

*Solder-joint design shall prevent solder and flux from entering the tube; the joint shall not be subject to preload; and joint surfaces shall provide proper wettability for soldering.*

Use an interference-fit Teflon sleeve inside the line or tube to prevent solder or flux from entering.

If possible, use soldered joints only where joint preloads can be avoided. In general, preclude preloads by tightening all B-nuts before the joint is soldered.

Clean and plate all surfaces to be joined according to known methods for effective surface preparation.

Soldered joints are not recommended for use in high-pressure lines or in locations subject to vibration.

### **3.4.5 Interference-Fit Joints**

Interference-fit joints are not recommended for use in a rocket fluid system.

## **3.5 SEALS**

### **3.5.1 Static-Seal Configuration**

*The static-seal configuration shall be consistent with the design requirements for limiting leakage from one region to another across a static interface under operating conditions of the particular application.*

Because of its sealing efficiency, simplicity, and low cost, the elastomeric O-ring seal should be the first choice where fluid compatibility, temperature, pressure, and joint separation permit. The general range of usefulness, dependent on material, is from 0 to 3000 psi pressure and -80° to 600°F temperature. Figure 22 should be consulted regarding limits for joint separation for elastomeric O-rings.

The molded-in-place seal should be used in place of the elastomeric O-ring when a positive indication of seal installation is required, seal groove tolerances are difficult to obtain, or elastomer exposure to the fluid must be kept to a minimum. The general operating range for this seal is the same as that for O-rings.

Metallic gaskets are recommended for applications where weight and leakage are not of prime concern. The advantages of metallic gaskets are low cost and ability to operate at extreme temperature and pressure. The general operating range is from -452° to +1800°F. The pressure limit is dependent on the particular design, with some developmental designs going as high as 40 000 psi.

Metallic pressure-assisted seals are recommended for applications involving extreme temperature where leakage or weight are of prime concern and where the plastic spring-loaded seal cannot be used because of temperature or pressure limitations. The useful temperature range is -452° to 1200°F, and developmental systems have employed these seals to 10 000 psi pressure.

The plastic spring-loaded seal should be used in applications where the temperature is low (below the useful range of elastomeric O-rings), the pressure is moderate, and weight is of prime concern. Minor leakage does occur at low temperatures (<-200°F) with most plastic spring-loaded seals, generally because of shrinkage. The pressure operating range extends up to 10 000 psi; however, this limit is greatly dependent on temperature and overall seal diameter. These seals have been used in the temperature range of -452° to 600°F.

The radial or toggle metallic seal should be used in circumferential applications (piston-bore type of seal) where the pressure or temperature range is beyond that of elastomeric O-rings. These seals have a temperature range of -452° to 1200°F, and developmental systems have used these seals to 10 000 psi pressure.

The metallic boss seal should be used for small ( $\leq 1$  in.) boss and coupling applications where the pressure or temperature range is beyond that of elastomeric O-rings. These seals have a temperature range of -452° to 1200°F, and systems using these seals to 10 000 psi have been designed.

Specific guidelines for installation of the various seals are given in sections 3.5.1.1 through 3.5.1.5.



### 3.5.1.1 GASKETS

#### 3.5.1.1.1 Elastomeric O-Rings

*The elastomeric O-ring properties and installation controls shall limit leakage as required but shall not allow extrusion, pinching, or cutting of the O-ring.*

As noted, the elastomeric O-ring should be the first consideration for a seal where this design permits. Elastomeric O-rings, however, are somewhat limited by fluid compatibility, temperature, pressure, and joint separation. For proper design, these parameters plus O-ring squeeze and installation dimensional controls should be considered when designing an elastomeric-O-ring coupling.

The recommended installation practice is to use circular-grooves. The O-ring squeeze for various static seal applications should be in accordance with the recommendations in table V. Reference 51 provides the dimensions for circular groove installations to achieve the proper squeeze. The structural rigidity and elastomer hardness are important factors in obtaining a seal at the maximum pressure. The no-extrusion area of figure 22 defines the limits of elastomers without the use of backup rings. On circumferential O-ring installations, backup may prevent extrusion; however, a backup ring should be placed on each side of the O-ring to prevent improper installation. In conjunction with figure 27, use the elastomer hardness at the operating temperature (fig. 33) and joint separation at the maximum pressures. Joint separation may be reduced by increasing the rigidity of the coupling or by use of a clearance-compensating device such as that discussed in reference 52. Provide lead-in chamfers and undercuts as illustrated in figure 21 to prevent O-ring pinching or cutting during installation.

#### 3.5.1.1.2 Molded-In-Place Seals

*The molded-in-place seal flange and restraint devices shall have sufficient rigidity and clamping force to control leakage and prevent seal extrusion.*

Under all service conditions of temperature and pressure, the total separation between the two mating flanges should not exceed 0.0003 in. This value includes separation due to out-of-flat conditions on both the flanges and seal retainer plate. Separation is reduced by using flanges and plates with tighter flatness controls and by increasing the rigidity and restraining force (i.e., bolting) of the coupling. The retaining plate must provide adequate strength to contain the elastomer during operation and during the elastomer molding operation.

Do not use branching or overlapping configurations when pressures are above 150 psi. For higher pressures, elastomer inserts with separable grooves are recommended.

Table V. Recommended Installation Conditions for Elastomeric O-Rings in Static-Seal Applications

O-ring cross-section diameter, in.		Squeeze, %*		
Nominal	Actual size	Flat-face installation	Radial installation without backup rings	Radial installation with backup rings
1/16	0.070 ± .003	19 to 32	16.5 to 26	13.4 to 28
3/32	0.103 ± .003	20 to 30	16 to 22.5	9 to 16
1/8	0.139 ± .004	20 to 30	16.3 to 22.4	8.5 to 15
3/16	0.210 ± .005	21 to 30	16 to 21	8 to 13.5
1/4	0.275 ± .006	21 to 29	15.6 to 20	10 to 15
		Pressure limitation		
		None. Limited only by structural deflections	1500 psi	Verified to 3000 psi. Application higher than 3000 psi limited by deflections and elastomer hardness. Above 3000 psi, verification testing required.
		Complete requirements for installation dimensions		
		Reference 46	Reference 51	References 48 and 51

\*% squeeze  $\frac{D-d}{D} \times 100$ , where D = diameter of O-ring cross section in free state, in  
d = groove depth, in

### 3.5.1.1.3 Metallic Gaskets

*The metallic gasket installation shall provide loading and rigidity adequate to meet the leakage requirements.*

Use metallic gaskets when operating conditions exceed the limits of elastomeric materials and when extra weight and leakage will not impair the function of the system. Metallic gaskets require high seal loads; therefore, rigid, heavy coupling flanges are required. Predicted seal leakage should be obtained from the seal suppliers and should be considered for its system impact. Required loading and flange-separation limits should also be obtained from the seal suppliers and this information used to design the mating coupling.

### 3.5.1.2 PRESSURE-ASSISTED SEALS

*The pressure-assisted seal material and configuration shall limit leakage as required while withstanding the forces of extreme temperature and pressure, and shall not contribute to excessive weight.*

A deflection analysis of the flanges under operating pressure (ref. 24) is recommended before selecting a static seal. Section 3.2.2.2 provides recommended practices for determining deflections and increasing flange rigidity. The selected seal must have the resilience to maintain the flange contact load required for sealing under the maximum flange deflection (sec. 3.5.2.1), and the seal surface material should fill or conform to the asperities of the mating surface. Metallic pressure-assisted seals are recommended for both cryogenic and hot-gas applications with the lightweight, high-deflection flanges that are common to lightweight aerospace couplings.

Employ soft interface materials to keep the seal contact load to a minimum value and still obtain an effective seal. Teflon is recommended for cryogenic applications where the seal load is sufficiently low to keep from cutting through the Teflon. Soft metallic platings such as gold and silver should be used for both cryogenic and hot-gas applications (table II). For applications above 1200°F, nickel plating should be considered. Section 3.6.1.3 provides additional information on interface materials.

### 3.5.1.3 PLASTIC SPRING-LOADED SEALS

*The plastic spring-loaded seal materials and configuration shall be suitable for the temperature and pressure environment of the application.*

The unit-bearing-pressure/creep relationship of the plastic seal material should be evaluated prior to seal selection for applications above room temperature. Both service temperature and storage temperatures should be considered. Plastic seals are not recommended for

applications over 300°F or for liquid-hydrogen temperatures, although some applications have been successful at liquid-hydrogen temperatures.

It is recommended that the groove-and-seal combination be designed so that the groove provides hoop-stress-carrying support for the seal outer surface under pressurized operation.

#### **3.5.1.4 RADIAL OR TOGGLE SEALS**

*The radial or toggle seal installation shall prevent seal damage during assembly and shall maintain the seal load under the differential thermal conditions of the application.*

For cryogenic couplings, the material with the greatest shrinkage should be located at the O.D. of the seal; for high-temperature usage, the materials with the greatest expansion coefficient should be on the I.D. Perform an analysis to verify that seal load is achieved and maintained. Use sufficient coupling restraining material to prevent radial deflections that unload the seal.

#### **3.5.1.5 METALLIC BOSS SEALS**

*Metallic boss seals shall be suitable for use with the boss and the environment of the application; the unit bearing pressure shall be adequate to prevent leakage without damage to the boss face.*

Metallic spring-action seals should be used for threaded-boss fittings in applications beyond the temperature limits of elastomers. They should also be used when elastomers are not compatible with the fluid.

Drawing controls should provide for the required thread engagement, flatness, finish, and normality for a leak-tight boss coupling. Comply with the seal supplier's recommended requirements. A structural analysis of the thread engagement should be performed to verify the load capability. When reusability is a consideration, metallic boss seals that damage the mating fitting or boss should not be used.

The boss seals shown in table I should be used to prevent marring of aluminum, CRES, and heat-treated steels. Care should be exercised when using these and other seals on soft boss materials such as copper. Other seals not included in table I should be evaluated to determine whether they will mar the boss surfaces. Proper seal selection is recommended to permit maximum reusability of the boss without rework.

## **3.5.2 Static-Seal Design Integration**

### **3.5.2.1 LOADS**

*The static seal shall be capable of performing its sealing function under the installation and operational loads of the application.*

A thorough load analysis should be performed in the design phase for each static seal and coupling to ensure proper sealing under all operational conditions. This study should include deflections of couplings and thermal loading and unloading. Required seal load and allowable coupling separation may be obtained from seal suppliers. Table I provides loading and deflection information for some seal types. The effects of these loads on the structural integrity of the mechanical members involved should also be considered. References 54 and 55 are recommended for information on the mechanism and the contact loads required to effect a seal.

### **3.5.2.2 CONTACT SURFACES**

Design criteria and recommended practices for contact surfaces are the same as those described for couplings in section 3.2.2.1.

### **3.5.2.3 ENVIRONMENT**

*The seal shall be capable of functioning under all temperature and fluid conditions of the application.*

An accurate thermal analysis of the seal and coupling should be conducted during the design phase in order to furnish sufficient data for proper seal and coupling design. On the basis of this study and the properties of the fluid, select seal materials that are capable of functioning for the design life without excessive deterioration. Growth potential for extended engine operating limits, soakback after engine cutoff, and radiation effect of the exhaust plume in addition to the normal operational temperature effects should be considered.

### **3.5.2.4 SEAL RESILIENCE**

*The seal resilience shall enable the seal to follow the deflection of the mating coupling.*

A deflection analysis of the coupling as described in reference 24 is recommended. On completion of the analysis, the seal configuration with the required resilience can be selected. If the application requirements are beyond the capability of an elastomeric O-ring, metallic pressure-assisted seals with good resilience should be considered for lightweight applications. Table I provides information on the allowable coupling separation for various types of static seals.

### **3.5.2.5 SEAL PROTECTOR**

*A seal protector shall prevent annealing, sticking, or erosion of the seal and coupling surfaces.*

Perform a thermal study to determine if a barrier could be used to reduce the temperature of the seal to a level that would not anneal the seal material. Barriers should be considered on hot-gas applications above 1000°F to permit a slow, even temperature change to reduce thermal stress. In systems with abrasive particles, use a barrier of the type shown in figure 27 to prevent erosion of the seal and coupling surfaces.

### **3.5.2.6 LEAKAGE CONTROL**

*Leakage control provisions shall maintain positive separation of incompatible fluids within the system and provide high reliability against external leaks of combustion gases.*

Incompatible fluids in close proximity at a coupling within the propulsion system should be sealed with separate seals and separate vents. A recommended configuration is shown in figure 28.

It is recommended that proven configurations be utilized to prevent external "blow-torch" leaks of thrust-chamber gases. Figure 29 shows a recommended configuration.

## **3.5.3 Dynamic-Seal Configuration**

*The dynamic-seal configuration shall be consistent with design requirements to control (minimize) leakage across a dynamic interface under the operating conditions of the particular application.*

Refer to figure 30 and table III for initial guidance in seal selection for particular applications.

Because of their sealing efficiency, simplicity, and low cost, elastomeric seals are a first choice where fluid compatibility, temperature, and pressure permit. Elastomeric O-rings should be considered first and then lip or cup seals if the O-ring configuration is unsuitable. For cryogenic applications, plastic lip seals are recommended. Metal seals of the mechanical type or piston rings should be used for high-temperature applications where elastomers and plastics deteriorate. Metal seals are also recommended for applications that must span both cryogenic and high temperatures. Fluid compatibility should be considered when selecting the seal configuration. The final seal design must provide an acceptable balance between leakage, friction, and wear.

### 3.5.3.1 O-RINGS

*O-rings shall have material properties suitable for the application and the proper squeeze to limit leakage and to prevent extrusion damage*

Select an O-ring material that is compatible with the operating fluid, temperatures, and pressure. The recommended groove configurations that result in the proper O-ring squeeze are specified in reference 48. Use lead-in chamfers and undercuts as illustrated in figure 20 to prevent pinching and cutting of the O-ring during installation and operation. Perform a structural analysis to determine if the clearance exceeds the no-extrusion area limits shown in figure 22 for the selected elastomer hardness at the design pressure. If it does, consider using backup rings, increasing elastomer hardness, or increasing the rigidity of the installation. If backup rings are used, they should be used on each side of the O-ring to prevent improper installation. Lubrication of the O-ring is recommended to prevent damage during O-ring installation and provide lubrication during operation. A lubricant is necessary for pneumatic service to prevent early seal failure. Recommended elastomers and lubricants for O-rings in various fluid systems are shown in table VI.

O-rings are not recommended in rotary applications for temperatures below  $-40^{\circ}\text{F}$  because loss of flexibility and thermal shrinkage reduce squeeze. Elastomer Shore A hardnesses of 80 to 84 are recommended, with minimum O-ring cross-sectional diameter. Reference 49 recommends maximum cross-sectional diameters for various speeds up to 1500 ft/min. Rotary O-ring grooves should be located in the housing to avoid excessive O-ring wear due to centrifugal action. This type of installation also minimizes the tensional stress, which is detrimental where heat is generated under high rotation speeds. To minimize extrusion in reciprocating applications, the O-ring groove should be placed so that the friction of the moving metal surface across the O-ring is opposite to the direction of the pressure. Finishes between 8 and 16  $\mu\text{in.}$  are recommended for surfaces rubbing against the O-ring. Avoid surface finishes of less than 8  $\mu\text{in.}$  on the rubbing surface, because the lubricant can be wiped off this surface; lubricant removal would result in high friction and wear. Rubbing speeds should be greater than one foot per minute to reduce friction. Lower speeds should be avoided since they cause excessive twisting and spiral-type failures. For minimum wear, avoid short, rapidly oscillating motion.



Table VI. Elastomers and Lubricants Recommended for O-Rings in Use with Typical Propellants and Service Fluids

Fluid	Recommended elastomers	Recommended O-ring lubricant
<b>Fuels</b>		
A-50	Ethylene-propylene rubber; butyl rubber	Fluorinated grease*
Hydrogen	Ethylene-propylene rubber; butyl rubber	Fluorinated grease
$\text{GH}_2$	Do not use elastomer	Fluorinated grease
$\text{HH}_2$		Fluorinated grease
MMH	Butyl rubber	Fluorinated grease
$\text{N}_2\text{H}_4$	Ethylene-propylene rubber; butyl rubber	Fluorinated grease
RP-1	Nitrile rubber; fluorocarbon polymer; neoprene	Fluorinated grease
UDMH	Ethylene-propylene rubber; butyl rubber	Fluorinated grease
<b>Oxidizers</b>		
CTI	Neoprene	Do not use lubricant
Fluorine ( $\text{F}_2$ )	Do not use elastomer	Do not use lubricant
$\text{H}_2\text{O}_2$	Fluorocarbon polymer; chlorosulfonated polyethylene	Fluorinated grease
IRI NA	Fluorocarbon polymer	Fluorinated grease
$\text{N}_2\text{O}_4$	Resin-cured butyl rubber; AFE-124 materials	Fluorinated grease
Oxygen	Fluorocarbon polymer	Fluorinated grease
$\text{GO}_2$	Do not use elastomer	Fluorinated grease
$\text{LO}_2$		
<b>Service Fluids</b>		
Helium	Ethylene-propylene rubber; butyl rubber	Fluorinated grease
Lubricating oil (ref. 100)	Fluorocarbon polymer	None needed
Methylene chloride	Fluorocarbon polymer	Fluorinated grease
Nitrogen	Nitrile rubber; fluorocarbon polymer	Fluorinated grease
$\text{GN}_2$	Do not use elastomer	Fluorinated grease
$\text{LN}_2$		

\*Fluorinated grease is commercially available as Krytox 240 AC or Braycote 640 AC

### 3.5.3.2 LIP SEALS

*The lip seal shall have the required seal load, flexibility, and structural strength to limit leakage.*

The seal material should be compatible with the operating fluid, temperature, and pressures throughout the required life. The interference between the seal and the moving element should result in an adequate load to seal low pressures before the pressure-assist feature is effective at the higher operating pressure. Recommended methods for obtaining higher loads include garter springs at the end of the lip and increased seal thickness.

Flexibility for following shaft deflections is provided in the combination of the selected material, seal length, and seal thickness. Multi-ply thicknesses should be considered when additional flexibility is required, as in plastic cryogenic seals. Consider an additional ply or outer seal vs a contamination-excluder seal (fig. 30(d)).

The lip length should provide the longest leakage path consistent with the required seal contact load (bearing pressure) between the seal and moving member. The selected length on cold-formed plastic shaft seals should be within the capability of the seal material to be installed without cracking or tearing. Additional overlapping seals should be considered if additional loading is required and forming limitations or flexibility needs prevent the selection of a thicker seal.

When using a plastic lip seal as a shaft or piston seal in cryogenic service, install the seal in a housing so that shrinkage at low temperature increases rather than decreases the contact pressure at the dynamic sealing surface.

A support on the low pressure side of the lip seal is recommended for high-pressure applications. Locate the support between the housing and shaft; the support itself may be either molded or machined as an integral part of the seal or installed as a separate part.

The static flange seal between the seal and stationary housing flange should have an adequate loading device to prevent leakage. Care must be taken to prevent excessive cold flow of plastic seals. Variations in the flange seal load due to pressures and temperature changes must be considered.

The surface finish on the dynamic surfaces of the seal and mating shaft should be consistent with the leakage and life requirements. Surface finishes of 32  $\mu$ in. on seals and 8  $\mu$ in. with a circular lay on shafts are recommended. Sealing surfaces should be identified on drawings with instructions to obtain and maintain a damage-free surface.

Test prototype seal assemblies to verify the design. The test conditions should be at the design limits to ensure that leakage requirements will be met under all conditions of shaft

deflection, pressure, temperature, environment, and life. Investigate by test the effect of seal lip length and thickness on leakage and life.

### 3.5.3.3 PISTON RINGS

*Piston-ring configuration and loading shall be adequate to limit leakage as required for the service life.*

Piston rings are recommended for applications where an all-metal seal or exceptionally low friction is required and relatively high leakage can be tolerated. The split gap should be capable of accommodating the diametral variations due to manufacturing tolerances and operating conditions. For applications that require minimum leakage, use an overlapping-step-gap design. Additional rings with staggered gaps also reduce leakage and are recommended; a 15-percent reduction in leakage with the use of a second ring is reported in reference 10. On high-pressure systems, pressure balancing is recommended to reduce contact loads that reduce friction and improve wear life. When pressure balancing is used, minimum seal contact width of 0.03 in. should be left intact to ensure adequate seal width. Consider friction, wear, and fluid compatibility to obtain the required functional and environmental requirements. Reference 10 is recommended for additional design information on piston-ring design and pressure balancing.

### 3.5.3.4 MECHANICAL SEALS

*The mechanical seal configuration shall provide the required combination of sealing load and wear life needed for the application.*

Perform a stress analysis to ensure that the bearing pressure at the sealing interface is adequate to control leakage and low enough to permit the required cycle life. Analyze static and dynamic loads under all operating conditions including installation variables. Ensure that the PV (contact pressure multiplied by sliding velocity) factor for the material combination is adequate for the required operating life. Obtain PV factors from material suppliers. Perform a dimensional analysis of seal alignment to ensure acceptable seal loading; consider installation variables and deflection under structural and pressure loads and temperature change.

A recommended analytical method for determining surface-finish requirements for dynamic-seal contact surfaces is outlined in reference 101. The surface-roughness value resulting from this analysis should be divided by a factor of three for seal design to allow for some degradation during life cycling.

Test prototype seal assemblies to verify the design. The test conditions should be at the design limits, to ensure that leakage requirements and cycle-wear-life requirements can be met.

### 3.5.4 Dynamic-Seal Design Integration

#### 3.5.4.1 LOADS

*The seal shall fulfill the design objectives for leakage control and operational life when exposed to the mechanical, fluid-pressure, and thermal loads of the application.*

Analyze seal loads resulting from installation and all operational factors including structure, pressure, temperature, deflection, and friction. Perform stress analysis to verify structural integrity and conformance with interface bearing pressure requirements. Analyze sustained stresses due to installation and operational loads to verify that material creep characteristics have not been exceeded at the operational temperature extremes.

Perform an analysis of the bearing stresses at the dynamic seal interface to ensure the ability to seal at low loads and the ability to operate with acceptable friction and wear. Consult reference 48 for O-ring requirements. Calculate the PV factor to evaluate seal life. Recommendations on PV factors should be obtained from material suppliers.

Minimize friction by designing for minimum contact loading as related to pressure, mechanical, and thermal loads. Select the material combination having the lowest coefficient of friction consistent with other design requirements. Reference 9 is recommended for friction data. Specify surface finishes consistent with the lowest friction coefficient and cost. Lubricate where possible; consider dry-film lubricants, flame plating, and metallic platings for difficult applications. If dry-film lubricants are used, be aware that some liquid propellants remove the dry-film lubricant from the base material. Consider the use of anti-friction devices such as slip rings for O-rings. Provide, where possible, pressure forces acting on the seal in the direction opposite that of friction forces. A stress analysis based on maximum friction forces should be performed to ensure that the seal has adequate strength.

Analyze the seal design from the standpoint of loads developed by fluid pressure acting on unbalanced seal areas. Evaluate the effect of manufacturing tolerances on effective sealing areas and effective seal contact areas. The resultant loads must be combined with mechanical and thermal loads in evaluating structural stresses and interface contact pressures.

Perform a stress analysis of the bearing stresses at the seal interfaces to ensure that material yield and bearing pressures are not excessive and that sufficient contact pressures are maintained to control leakage. Evaluate the effect of manufacturing tolerances and installation on deflection and distortion of the seal. Establish manufacturing, processing, installation, and inspection requirements that ensure a satisfactory installation.

Analyze the stresses induced by expansion and contraction of the component members due to operational and environmental temperatures. Evaluate the resulting interface bearing stresses and seal structural stresses to ensure that sufficient contact pressure is maintained and that maximum allowable stresses are not exceeded. Provide thermal compensation if required to maintain adequate interface bearing stresses.

### **3.5.4.2 CONTACT SURFACES**

*The sealing contact surfaces shall provide the degree and extent of contact required to control leakage, minimize friction, and minimize wear over the range of operating conditions and life of the seal.*

Consult reference 48 for requirements for O-ring dynamic seals, data on dynamic interface leakage as a function of surface topography, and bearing stress. Perform an analysis to ensure that the sealing interface bearing pressure at both the static and dynamic interface is high enough to control leakage. Ensure that the material yield properties are not exceeded at the static interface and that bearing pressures at the dynamic interface are within the acceptable limits of friction and wear. Section 3.5.4.4 treats surface requirements for minimizing friction and wear.

### **3.5.4.3 SEAL DEFLECTION AND RESILIENCE**

*The seal shall possess deflection and resilience capability adequate to provide the dynamic seal contact required to control leakage throughout the range of operating conditions.*

New seal configurations, unless supported by reliable empirical design data, should be subjected to development testing to ensure conformance to design requirements. Engineering judgement and good design practices must ultimately be used in controlling deflection and finishes within the resilience limits of the seal.

Design data useful in evaluating elastomeric and plastic seal resilience have been derived primarily by empirical methods for special configurations. An example is the O-ring, the extensive testing of which has made its use particularly attractive. Development of configurations such as the lip seal shown in figure 30(c)-(f), cup seals, and similar designs requires extensive design analysis and test. Analysis of a face seal such as that shown in figure 30(h) requires an analysis of bellows deflection under all operating conditions.

### **3.5.4.4 WEAR**

*Seal wear throughout the operating life shall not result in excessive leakage, friction, or contamination.*

To control wear, use materials with the lowest coefficients of friction; lubricate where possible; use low loads at the dynamic-seal contact surface; specify the required surface finish; and provide a wiper if required to exclude contamination. Actual wear life as related to coefficient of friction, load, environment, and operating time must be verified by test. Specify surface finishes consistent with the lowest cost that will minimize abrading wear. Do not specify less than 8- $\mu$ in. finish for linear hydraulic O-ring applications, because such a fine finish could wipe the dynamic sealing surface dry of lubricant. Specify component service-life requirements on the basis of minimum anticipated seal life.

#### **3.5.4.5 LUBRICATION**

*Seal lubrication or its equivalent shall control friction and wear as necessary to maintain reliable operation over the range of operating conditions.*

Specify a grease-type lubricant where possible if the operating fluid does not provide adequate lubrication. If lubricants are not permitted because of environmental or operating fluid requirements, use materials that minimize wear. Reference 9 is recommended as a source for material and friction data. Product quality of commercial lubricants varies from batch to batch, and occasionally the composition is changed. In addition, "tradename" lubricants generally are proprietary products, and it is difficult to obtain specific information on ingredients. For these reasons, a military specification should be called out or a control specification should be utilized to control the product.

#### **3.5.4.6 CONTAMINATION**

*The seal shall be capable of performing satisfactorily with specified contamination in the operating fluid and shall not generate contamination detrimental to the system.*

To absorb contamination, use a soft seal material with contact bearing area for the seal as large as possible. Use the maximum possible hardness on the sliding element. For exceptionally dirty fluids, use an auxiliary seal for protection of the primary seal, or specify filtration.

### **3.6 GENERAL CONSIDERATIONS FOR DESIGN**

#### **3.6.1 Material Selection**

##### **3.6.1.1 METALS**

### 3.6.1.1.1 Compatibility

*The metal alloys used in the system shall be compatible with the system fluid and resistant to all anticipated forms of corrosion.*

The following material-compatibility considerations should be evaluated in selecting materials:

- Chemical reaction between materials and system fluid no reaction, slow reaction, or impact ignition
- Propellant decomposition
- Corrosion -- atmospheric, galvanic, stress.

A list of suitable materials should be prepared on the basis of the above considerations. Data on material/fluid compatibility are presented in references 56 and 57. In a liquid-oxygen system, do not use materials such as titanium or organic lubricants that ignite on impact or when subjected to rubbing; in any system, do not use materials that cause propellant decomposition.

The electrochemical series for metals should be consulted when selecting materials for parts that touch each other. Reference 59 provides the recommended usage information on the electrochemical series for dissimilar metals; reference 60 provides information on acceptable platings and finishes.

Materials susceptible to stress corrosion should not be used. Such materials include the following alloys in the standard heat-treated conditions specified: 2014-T4, -T6; 2024-T3, -T4, -T351; PH15-7Mo RH950, TH1050; 17-7PH annealed, H900; 17-4PH RH950, TH1050; AM350 SCT850; AM 355 SCT850 (ref. 102).

### 3.6.1.1.2 Physical and Mechanical Properties

*The physical and mechanical properties of the metal alloys shall be consistent with functional requirements for the components.*

References 62, 63, and 64 are recommended for obtaining physical and mechanical properties of aerospace-type materials. The selected material should result in the lowest cost component that does not penalize weight or envelope.

To provide maximum resistance to disconnections to corrosion, handling, and blast damage, use stainless steel or nickel-base alloys for all exposed parts of the ground half, and of the flight



half also if weight consideration permits. Use heat-treatable, corrosion-resistant steel or nickel-base alloys for springs and load-bearing components of the latch/release mechanism. With cryogenic propellants, do not use materials that are brittle at cryogenic temperature. The effects of different shrinkage rates of materials should be considered in selecting materials for sealing, latching, and releasing mechanisms in cryogenic disconnects. The unit loading on plastic sealing members, for example, may reduce significantly at low temperatures because of their greater coefficient of thermal expansion.

#### **3.6.1.1.3 Weldability**

*Components to be welded to another component shall be weld compatible.*

The component should be of the same material as the adjacent component when possible. If dissimilar materials are selected, they should provide weld integrity and the required strength. Reference 92 contains information pertaining to weldability of commonly used dissimilar alloys and recommended filler material.

#### **3.6.1.1.4 Material Imperfections**

*The seal contact area of the components shall be free of material imperfections that would result in leakage greater than allowable.*

Seal interface surfaces on casting should specify the density requirement to effect a leak-free joint. This requirement should be verified on the first part fabricated to check out the mold design. Reference 103 contains recommended limits on pore size and the number of pores per square inch for various systems that do not require mass-spectrometer leak measurements, i.e., less than  $1 \times 10^{-4}$  cm<sup>3</sup>/sec of helium. Castings are not recommended for components that must meet mass-spectrometer-type leakage rates.

Vacuum-melt 321 CRES is recommended where stringers cause mass-spectrometer-type leaks. However, the additional material cost requires that the mass-spectrometer-type leak requirement be justified before considering vacuum-melt material.

#### **3.6.1.2 ELASTOMERS**

##### **3.6.1.2.1 Fluid Compatibility**

*The elastomer shall be chemically and physically compatible with the system fluid.*

The known elastomers should be reviewed for compatibility with the system fluid(s) to determine which ones have the best compatibility properties for the applications. Table VI lists both the elastomers recommended for the common fluids used in liquid-rocket engines and lubricants that are compatible with both the elastomer and fluid. Reference 56 is recommended for additional compatibility data on elastomers.

#### **3.6.1.2.2 Pressure Capability**

*The elastomer material properties shall limit extrusion and wear under all functional and test pressures.*

Elastomer hardness should be selected so the pressure condition and clearance is within the no-extrusion area of figure 22. If it is impractical to use harder elastomer materials to achieve this condition, then other means should be used to avoid extrusion. These means are discussed in section 2.5.1.1.1.

The ability to seal and resistance to extrusion are related directly to the elastomer's hardness; the recommended hardness for most applications is Shore A hardness of 70. For low fluid pressures or low temperatures or both, a softer material is recommended (for improved sealing). For service where wear is a consideration, a harder material should be used.

#### **3.6.1.2.3 Temperature Capability**

*The elastomer shall have physical properties adequate to withstand the service temperature of the application.*

Recommended practice is to select carefully, from the physical data available, an elastomer that will not lose its elastic properties when exposed to the operating temperature. Table IV gives the useful temperature ranges for the basic types of elastomers, and figure 35 shows seal life versus temperature curves. In selecting an elastomer for a high-temperature application, the actual temperature at the seal, the time at temperature, and possible soakback conditions should be considered. With this information, the proper material for the application can be selected. The designer should use a realistic temperature range without a large margin of safety in order not to penalize the design. This kind of penalty could occur if the resultant temperature exceeded the limit for any of the elastomers and forced a new design approach. In selecting an elastomer for low-temperature service, flexibility and the coefficient of expansion (or contraction) are prime considerations. In conventional seal applications, an elastomeric compound should not be used below its 10-percent-retraction level. Figure 34 shows these data for Buna N, Viton, and fluorosilicone compounds. Low-temperature service may also require additional squeeze at room temperature to compensate for the elastomer contraction at operational temperature.

### 3.6.1.3 SEAL INTERFACE MATERIALS

*Seal interface material shall be suitable for the service environment, leakage allowable at the interface, and the storage life of the engine*

The interface material should flow plastically under design loads. Teflon, which is recommended for cryogenic service, will creep or cold flow during installation loading and storage; therefore, the unit bearing pressure should be kept low enough for the material to perform its intended function. Cold-flow data are available for most plastic materials used for seals (Teflon, Kel-F) and should be referred to in the design phase (refs. 78 through 82). Where possible, control cold flow by trapping (or confining) the plastic material mechanically. Processing controls to maintain bonding and bearing strength should be specified. Metal platings recommended include silver and gold and, for temperatures above 1400°F, soft nickel. Silver-plated seals should be processed to resist blistering and flange sticking on hot-gas applications above 600°F.

## 3.6.2 Component Handling

### 3.6.2.1 SURFACE PROTECTION

*Seal contact surfaces shall be protected during all fabrication and handling operations.*

Identify the critical seal contact surfaces of couplings and seals on drawings and manufacturing orders. Manufacturing orders should include instructions for exercise of caution and use of protective devices as soon as possible in the fabrication. Figure 36(b) shows a typical protective device.

### 3.6.2.2 CLEANLINESS

*Foreign matter shall not be introduced into the installation upon assembly*

Care should be exercised in maintaining the individual elements or components in a clean condition prior to further processing, as in welding and brazing, or prior to assembly. Parts should be heat sealed in plastic after cleaning to maintain the clean condition, with subsequent assembly in an aerospace-quality clean area. Care should be taken to prevent the entrance of any contaminant during assembly.

### 3.6.3 Evaluation Testing

#### 3.6.3.1 DISCONNECTS

*Evaluation tests of disconnects shall demonstrate both the separation function, including appropriate signal of disconnect position as required, and sealing function.*

Development test conditions should be controlled for proper loading, simulated fluid flow, pressure, and environmental temperature. Fabrication testing should include a performance check of the latch-release mechanism, connect and disconnect indicator switches if included, operational check of the valves, and a leakage test on the seals.

#### 3.6.3.2 COUPLINGS AND SEALS

*Evaluation tests of couplings and seals shall demonstrate that the assembly seals as required. Testing of dynamic seals shall include demonstration of cycle life.*

Evaluation testing should be performed on static couplings and seals to verify their sealing ability under all operational environments. Dynamic couplings require additional testing to evaluate friction and cycle-wear life. Laboratory testing is recommended for initial evaluation on new coupling designs to determine effects on leakage of (1) various pressure levels (e.g., low-pressure leak checks and proof and burst tests); (2) pressure cycling; (3) tolerance - extreme seal/coupling installation; and (4) dynamic cycling. Selection of instruments for measuring leakage must be made on the basis of leakage level permitted. Reference 86 provides information for matching allowable leakage rates obtained with various methods and instrumentation.

Operational leakage monitoring is recommended to verify typical seal-couplings during engine hot-firing. If leakage is greater than that predicted from laboratory evaluation, additional test data should be obtained to permit design improvements. This additional testing could include (depending on the specific problem) tests to obtain temperature, strain, and vibration data.

The impact of leakage-monitoring provisions on static seal-coupling design should be minimized, so that the provisions may be eliminated after the design approach has been proven. Recommended methods for accomplishing this are shown in figure 18. This approach improves seal-coupling effectiveness by keeping the bolt circle close to the seal and has the additional benefit of lighter coupling weight.

### 3.6.3.3 FITTINGS

*Evaluation tests for fittings shall demonstrate pressure loss and flowrate values with the application.*

Hot firing on engine static test is recommended for verification of design when all desired measurements can be made. Many fitting problems are structural, and the full spectrum of loading is difficult to duplicate on other tests.

A recommended design for a tap that will enable accurate measurement of static pressure is presented in figure 41 (ref. 104).

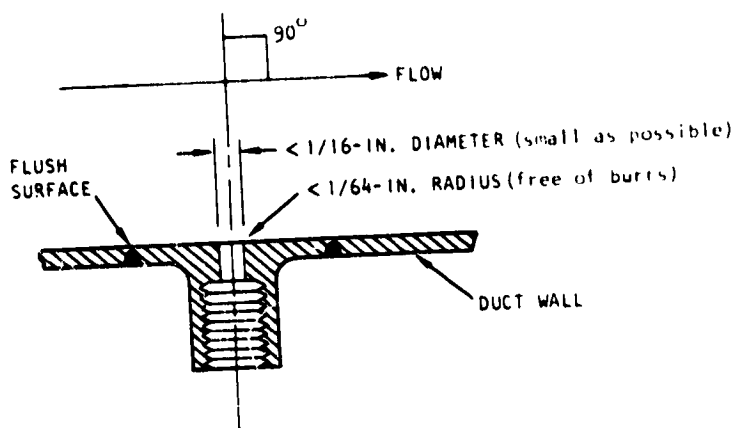


Figure 41. — Recommended design for static-pressure tap (ref. 104).

Accurate orifice and nozzle flow measurements should be made in accordance with reference 105. Presented in this reference are geometric construction details of orifices, nozzles, and tap locations; flow coefficients, fluid expansion factors, and orifice expansion factors; sample flow calculations, and accuracy limitations of the recommended coefficients. References 96 and 104 contain information pertinent to making the necessary pressure and temperature measurements.

### 3.6.3.4 FIXED JOINTS

*Evaluation tests of fixed joints shall demonstrate adequacy of the joining technique with regard to structural strength and leakage.*

Laboratory testing of the different fixed-joint designs is recommended to verify that the joints meet all requirements for the particular engine program. Recommended tests include (1) destructive examination of "as fabricated" joints and of joints that have passed the various tests, (2) proof and burst tests, (3) environmental tests including exposure to the system fluid, and (4) fatigue or vibration life tests. During laboratory evaluation tests, nondestructive test methods should be established for use on production joints. Joint design changes should be made if necessary to assure meaningful interpretation of the selected nondestructive test method.

All production joints should be subjected to appropriate nondestructive examinations as soon as possible in the system fabrication sequence. Penetrant inspection (LOX-compatible type) is recommended to determine surface defects in all joint types. X-ray or ultrasonic examination is recommended to determine internal defects in all joint types. When applicable, X-ray examination is also recommended for evaluating the joint for cracking of the weld-bead root. When suitable standards can be developed, ultrasonic examination is recommended for determining weld-bead width on sleeve welds. All joints should be subjected to a proof-pressure test. Helium gas is recommended because it is compatible with most fluid systems, and standard leakage instrumentation can measure helium leaks. As evidence that porosity or a weld defect does not exist, a leakage rate of less than  $1 \times 10^{-6}$  standard cubic centimeters per second of helium is recommended.

## APPENDIX A

### Conversion of U.S. Customary Units to SI Units

Physical quantity	U.S. customary unit	SI unit	Conversion factor <sup>a</sup>
Angle	degree	radian	$1.745 \times 10^{-2}$
Force	lbf	N	4.448
Length	in.	cm	2.54
	$\mu$ in.	$\mu$ m	$2.54 \times 10^{-2}$
Load	psi (lbf/in. <sup>2</sup> )	N/cm <sup>2</sup>	$6.895 \times 10^{-1}$
	lbf/in.	N/cm	1.75
Pressure	psi (lbf/in. <sup>2</sup> )	N/cm <sup>2</sup>	$6.895 \times 10^{-1}$
Surface finish or roughness	$\mu$ in.	$\mu$ m	$2.54 \times 10^{-2}$
Temperature	°F	K	$K = \frac{5}{9} (^{\circ}\text{F} + 459.67)$

<sup>a</sup> Multiply value given in U.S. customary unit by conversion factor to obtain equivalent value in SI unit. For a complete listing of conversion factors for basic physical quantities, see Mechtly, E. A.: The International System of Units. Physical Constants and Conversion Factors. Second Revision, NASA SP-7012, 1973.



## APPENDIX B

### GLOSSARY

<u>Term or Symbol</u>	<u>Definition</u>
anodize	form a protective oxide coating on a metal by electrical means
B-nut	coupling nut used to connect the two halves of a threaded connector (fig. 12)
boss	thickened protuberance in the wall of a duct for the purpose of allowing attachment of components or connection of other lines
breakaway disconnect (rise-off or staging)	separable connector that is disengaged by the separation force as the vehicle rises from the launch pad or a stage separates from a lower stage
Brinell hardness	indentation hardness determined by pressing a hardened-steel sphere into the test material under a certain load for a specified time; the diameter of the impression produced is an index to the hardness
chatter	rapid uncontrolled seating and unseating of a valving unit
chevron seal	term for a set of "V"-shaped seals (fig. 30(k) and table I)
circular lay	circular direction of the predominant pattern of a machined surface; ordinarily determined by the production method used
circumferential seal	seal whose sealing surface is parallel to centerline of flow passage (also called radial seal)
cold flow	permanent deformation of material caused by a compressive load that is less than the load necessary to yield the material; some time is required to obtain cold flow
compression set	percent of deflection by which an elastomer fails to recover after a fixed time under specified squeeze and temperature
connector	a mechanical device for joining or fastening together two or more lines or tubes in a fluid system
Conoseal	trademark of Aeroquip Corp., Marman Div. (Los Angeles, CA) for a leak-proof joint using an all-metal radial seal

<u>Term or Symbol</u>	<u>Definition</u>
creep	permanent deformation of material caused by a tensile load that is less than the load necessary to yield the material; some time is required to obtain creep
creep strength	the degree to which a given material resists creep
crevice corrosion	corrosion that occurs in a narrow, relatively deep opening where two similar surfaces meet and trap a reactive fluid that acts as an electrolyte; corrosion occurs because of the concentration gradient of the reactive species established within the trapped fluid
cryogenic	fluids or conditions at low temperatures, usually at or below -238°F (123 K)
cryogenic seals	seals that must seal effectively at temperatures below -238°F
cryopumping	induction of vapor into a confined cavity by a local reduction in vapor pressure caused by condensation of vapor on an extremely cold (cryogenic) surface of the cavity
diffusion bonding	method of joining two different alloys wherein temperature and pressure create intermolecular bonds
disconnect	short term for quick-disconnect — a specific type of separable connector characterized by two separable halves, an interface seal and, usually, a latch-release locking mechanism; it can be separated without the use of tools in a very short time.
durometer	instrument that measures hardness in terms of resistance of material surface to an indenter point working under pressure
dynamic seal	a conformable element used to minimize leakage of a fluid from the flow-stream region when there is relative motion between the sealing interfaces
elastic limit	the maximum stress that can be applied to a body without producing permanent deformation
elastomer	polymeric material that at room temperature can be stretched to approximately twice its original length and on release return quickly to its original length
electron-beam welding	process in which a controlled stream of high-velocity electrons produces the heat for fusion by striking the workpiece in a vacuum

<u>Term or Symbol</u>	<u>Definition</u>
face seal	seal whose sealing surface is perpendicular to the centerline of the flow passage
flexible hose	convoluted metal tubing, normally having a braided wire sleeve on the outer surface
fluid interface	common boundary of two or more surfaces exposed to fluid (e.g., mating flanges of a duct)
forced-separation disconnect	a separable connector that is disengaged by explosive, hydraulic, or pneumatic pressure
garter spring	spiral spring formed into a closed elastic ring
gas metal-arc welding	inert-gas welding process using as a heat source an electric arc between a bare consumable filler wire and the workpiece
gas tungsten-arc welding	inert-gas welding process wherein heat is produced by an electric arc between a nonconsumable electrode and the work; a filler metal is optional
gland	cavity in which O-ring is installed; includes the groove and mating surface of second part, which together confine and squeeze the O-ring
GMA	inert-gas metal-arc welding
GTA	inert-gas tungsten-arc welding
HAZ	heat-affected zone (i.e., region of material affected by heat of welding or brazing)
hermetic seal	airtight seal evidencing no detectable leakage
hydrogen embrittlement	loss in ductility of a metal as a result of the exposure of the metal to newly formed gaseous hydrogen
interface	the region of mating (common boundary) between interconnected elements
Knoop hardness	indentation hardness determined by pressing a rhombohedral diamond into a material; similar to the Brinell test
K-seal	seal shaped like a K

<u>Term or Symbol</u>	<u>Definition</u>
land	the actual sealing surface of the part that mates with a seal
lanyard	a strong cord used to actuate a contrivance from a distance
L/d	length-to-diameter ratio
liftoff	term designating the instant of vehicle flight at which vehicle contact is terminated with all holddown and support devices; also called "first motion" of the vehicle
Mach number	ratio of the speed of fluid flow to the speed of sound in the fluid
manually operated disconnect	a separable connector that is engaged or disengaged by manual forces usually with the aid of a lanyard, cam, or similar device
Naflex seal	trade name for a U-shaped seal made by Navan Products. See table I.
Natorq seal	trade name for a modified ring seal made by Navan Products. See table I.
passivation	formation of a compact and continuous corrosion-resistant film on the surface of metal exposed to air (natural passivation) or to a chemical solution (artificial passivation)
pitot tube	tube having a short right-angled bend and immersed vertically in a moving fluid with the mouth of the bent part pointed upstream; used to measure fluid velocity
plastic	high-molecular-weight material that while usually firm and hard (although often flexible) 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
plastic flow (or deformation)	immediate yielding of material caused by a compressive or tensile load that is greater than that necessary to yield the material
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
preflight	occurring before vehicle liftoff
PV	product of bearing pressure P times rubbing or sliding velocity V

<u>Term or Symbol</u>	<u>Definition</u>
quick disconnect	See "disconnect"
$R_B, R_C, R_D, R_J, R_K$	symbols for hardness values on Rockwell hardness scales designated by the subscript
relative motion	rotary or sliding motion, as related to the seal interface and contact surface, in which one surface is stationary while the other moves
Reynolds number	nondimensional parameter (Re) representing the ratio of the momentum forces to the viscous forces in fluid flow
rise-off	term denoting that actuation is caused solely by vehicle vertical motion
rms	root mean square
Rockwell hardness	indentation hardness (of metals and plastics) determined by measuring surface indentation or penetration by a diamond cone or steel ball under a specified load
Shore hardness	hardness of plastics or elastomers measured by use of a blunt indenter point under pressure. (The Shore A method uses a blunt point with a 2.79-mm-diameter flat with a weight of 1 kg; the Shore D method, used for harder materials, uses a 0.1-mm-radius point with a weight of 5 kg)
solenoid valve	a poppet, spool, or piston valve actuated by an integrally mounted solenoid
spillage	amount of fluid that remains trapped within a double-valve disconnect when the disconnect is partially disengaged and both valves have closed
squeeze (O-ring)	compression of an O-ring between the groove bottom and the surface of the mating part in the gland assembly
SSME	Space Shuttle main engine
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
storable propellant	a propellant with a vapor pressure such that the propellant can be stored in a specified environment (earth or space) at moderate ullage pressures without significant loss over the mission duration
stress corrosion	corrosion of a metallic surface enhanced (i.e., increased in rate) by the existence of localized stresses, whether applied or residual

<u>Term or Symbol</u>	<u>Definition</u>
T3,T4,T6,T351	designations for heat-treating and tempering processes for aluminum alloys
umbilical	line or duct connecting the launch vehicle to ground facilities that supply power or fluids
valve-type disconnect	quick-disconnect coupling that includes valve elements for sealing purposes at separation
vapor honing	method of eroding (or cleaning) a metal surface by blasting a fine erosive material against the surface, a vaporized fluid being used as a carrier for the abrasive material
Vickers hardness	an indentation-hardness test in which the indenter is a diamond cone of a specified angle between opposite faces; the load, test duration, and rate of descent of the indenter are specified
weld droptrough	excessive weld-bead projection on the inner walls of a fluid passage
wettability	the ease with which a fluid will flow over and adhere to a surface (e.g., molten solder over a heated metal surface)
<u>Material<sup>1</sup></u>	<u>Identification</u>
AF-E-124	experimental fluorocarbon elastomers (perfluorovinylether)
AF-E-411	experimental filler/ethylene-propylene terpolymer
A-286	austenitic heat-resistant iron-base alloy
AM-350	semi-austenitic heat-resistant precipitation-hardening stainless steels
AM-355	
PH15-7Mo	
17-4PH	
17-7PH	
Braycote 640AC	trade name of Bray Oil Co. for a Teflon-filled fluorinated grease
Buna N	trade designation for copolymer of butadiene and acrylonitrile

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

<u>Material</u>	<u>Identification</u>
butyl rubber	synthetic rubber produced by copolymerization of isobutylene and isoprene
CTF	liquid chlorine trifluoride, propellant grade per MIL-P-81399
CRES	corrosion-resistant steel
ethylene-propylene rubber	an elastomer, a copolymer of ethylene and propylene
fluorine	elemental fluorine ( $F_2$ ) in its liquid form ( $LF_2$ ) used as a cryogenic propellant per MIL-P-27405
Fluorel	trade designation of 3 M Corp. for copolymer of vinylidene fluoride and hexafluoropropylene
$GH_2$	gaseous hydrogen
$GN_2$	gaseous nitrogen per MIL-P-27401
GOX	gaseous oxygen
Hastelloy C	trade name of Stellite Division of Cabot Corporation for austenitic nickel-molybdenum-chromium-iron alloy (AMS 5530C)
He, helium	pressurant helium per MIL-P-27407
$H_2O_2$	hydrogen peroxide per MIL-P-16005
hydrazine	$N_2H_4$ , propellant grade per MIL-P-26536
Inconel 718 X-750	trade names of International Nickel Co. for nickel-base alloys (AMS 5597A and 5598, resp.)
Invar	International Nickel Co. designation for a nickel alloy with very low coefficient of thermal expansion
JP-4	military jet fuel consisting of gasoline and light petroleum distillate per MIL-J-5161
Kel-F	trademark of 3 M Corp. for a high-molecular-weight polymer of chlorotrifluoroethylene
K-Monel	trade designation of International Nickel Co. for a wrought age-hardenable alloy containing Ni, Cu, and Al



<u>Material</u>	<u>Identification</u>
Krytox 240 AC	tradename of E. I. du Pont de Nemours and Co. for a Teflon-filled fluorinated grease
$\text{LF}_2$	liquid fluorine
$\text{LH}_2$	liquid hydrogen ( $\text{H}_2$ ), propellant grade per MIL-P-27201
$\text{LN}_2$	liquid nitrogen
$\text{LO}_2$ or LOX	liquid oxygen ( $\text{O}_2$ ), propellant grade per MIL-P-25508
methylene chloride	halogenated hydrocarbon solvent per MIL-D-6998
MMH	monomethylhydrazine, propellant grade per MIL-P-27404
Mylar	trademark of E. I. du Pont de Nemours and Co. for polyethylene terephthalate film
$\text{N}_2\text{H}_4$	hydrazine, propellant grade per MIL-P-26536
$\text{N}_2\text{O}_4$	nitrogen tetroxide, propellant grade per MIL-P-26539
neoprene	polychloroprene, a synthetic elastomer
nitrile rubber	copolymer of butadiene/acrylonitrile (Buna N)
nylon	thermoplastic polyamide
polyurethane	any of various thermoplastic polymers that contain -NHCOO-linkages; produced as fibers, coatings, flexible and rigid foams, elastomers, and resins
Rene' 41	trade name of General Electric Co. for an austenitic nickel-chromium-cobalt-molybdenum alloy
RP-1	kerosene-base high-energy hydrocarbon fuel, propellant grade per MIL-P-25576
rubber	an elastomer, either a natural compound obtained from the hevea brasiliensis tree or a synthetic material
Teflon (ETP)	trade designation of E. I. du Pont de Nemours and Co. for fluorinated ethylene-propylene polymer
Teflon (TEF)	trade designation of E. I. du Pont de Nemours and Co. for tetrafluorethylene polymer

Material	Identification
UDMH	unsymmetrical dimethylhydrazine, propellant grade per MIL-P-25604
Viton	trade designation of E. I. du Pont de Nemours and Co. for copolymer of vinylidene fluoride and hexafluoropropylene
Viton A	same as Viton but different cure
300 series (e.g., 304L, 321, 347)	series of austenitic stainless steels
440C	martensitic stainless steel
2014 2024	wrought aluminum alloys with Cu as the principal alloying element

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