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SPACE VEHICLE
DESIGN CRITERIA
(CHEMICAL PROPULSION)

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LIQUID ROCKET ACTUATORS AND OPERATORS

MAY 1973

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

FOREWORD

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

Environment
Structures
Guidance and Control
Chemical Propulsion

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

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

This monograph, "Liquid Rocket Actuators and Operators," 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 James G. Absalom of Rocketdyne Division, Rockwell International Corporation, and was edited by Russell B. Keller, Jr. of Lewis. To assure technical accuracy of this document, scientists and engineers throughout the technical community participated in interviews, consultations, and critical review of the text. In particular, Donald D. Laine of Systems Group, TRW, Inc.; D. D. Price of Aerojet Liquid Rocket Company; and Eugene J. Cieslewicz and Ross G. Willoh of the Lewis Research Center individually and collectively reviewed the text 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.

May 1973

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GUIDE TO THE USE OF THIS MONOGRAPH

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

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

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

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

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

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

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LIQUID ROCKET ACTUATORS AND OPERATORS

1. INTRODUCTION

The most numerous devices in rocket engine and spacecraft control systems are actuators and their operators. An actuator is a device that converts hydraulic, pneumatic, electrical, or potential energy into mechanical motion. An operator is a device that causes an actuator to function. Actuators such as those used for thrust chamber gimballing are designed as separate assemblies; most frequently, however, actuators are incorporated in assemblies of shutoff valves, flow-metering valves, vent and relief valves, pressure regulators, and operators for other actuators. Operators are designed as separate assemblies or subassemblies and as integral components of larger assemblies.

This monograph treats all the types of actuators and associated operators used in booster, upper stage, and spacecraft propulsion and reaction-control systems except for chemical-explosive actuators and turbine actuators. Discussion of static and dynamic seals, mechanical transmission of motion, and instrumentation is included to the extent that actuator or operator design is affected.

Selection of the optimum actuator configuration for a specific application requires a tradeoff study that considers all the relevant factors: available energy sources, load capacity, stroke, speed of response, leakage limitations, environmental conditions, chemical compatibility, storage life and conditions, size, weight, and cost. These factors are interrelated with overall control-system design evaluations that are beyond the scope of this monograph; however, literature references are cited for a detailed review of the general considerations. The monograph is limited to flight hardware, but pertinent advanced-state-of-the-art design concepts are surveyed briefly.

Technical literature on the subjects of actuators, operators, and their constituent elements provides extensive coverage of design techniques and design details for general service. General design practices, however, frequently have been inadequate in rocket engine and spacecraft applications. The severe environmental and operational conditions associated with space-flight service and the requirements for high reliability have necessitated the evolution of a highly specialized technology directed toward the successful solution of problems in the design of space-flight hardware.

The basic problems have been those involved in maintaining structural and functional integrity of a design while achieving required dynamic performance. Dynamic seals in piston-cylinder actuators have been damaged, with consequent loss of sealing capability and shifts in actuator response, by metal particle contaminants generated by moving or vibrating piston springs, by abrasion by contacting sliding surfaces, and by the effects of wide temperature ranges on seal material properties. Diaphragms and bellows have failed as a result of installation mismatch, welding heat damage, overstressing during assembly, interply or interface contamination, interply pressurization, fluid-flow-impingement damage, temperature-related shifts in reference spring forces, and vibration-induced fatigue. Solenoid-actuator structural and performance inadequacies arising from mechanical vibration, exposure to wide ranges of temperature, current-drain limitations, requirements for fast and repeatable response, and exposure to reactive fluids have necessitated development of advanced techniques in construction and in the use of materials. The extensive use of pneumatic actuators and operators has required development of compensation techniques for accommodating fluid-compressibility effects and fluid-temperature effects on dynamic response.

Because solutions to the primary problems in space-flight actuators and operators involve the detail hardware design, the actuator and operator categories in the monograph are established on the basis of salient features of design configuration. The monograph emphasizes experience with particular detail problems and with demonstrated successful solutions. It is concerned with hardware rather than with procedures and concentrates on the specific configuration features that are critical in successful design. It recognizes that many problems (e.g., seal damage by flushing solvents and test fluids) are not strictly component design problems but that their severity often can be lessened by a design that anticipates potential hazards. The monograph considers complete and detailed specification of processing, installation, test, and servicing procedures as an integral part of design activity for control of design-associated problems.

2. STATE OF THE ART

The technology for space-flight actuation systems has evolved from an extensive background of general machine design experience. More specifically, the most modern hardware technology, as demonstrated in the Apollo flights, was derived from experience with hardware for earlier space-flight and ballistic-missile programs that in turn were heavily influenced by design practices for aircraft engines and vehicles.

The most commonly used actuator configurations for space-flight service are the linear-motion piston, diaphragm, bellows, and solenoid. These configurations minimize problems associated with fluid leakage, fluid chemical compatibility, large temperature excursions, severe vibration environments, and control of dynamic response. The extensive use of these kinds of actuators has resulted in a highly developed and sophisticated state of the art in the design of the critical features.

A variety of electric rotary-motor types and configurations, designed especially for rocket engine and spacecraft systems, has been developed successfully for reliable operation. Leakage-control problems and inefficient use of actuation fluid have restricted the use of hydraulic and pneumatic rotary actuators.

The use of nutating actuators and actuators with digital or with fluidic-logic inputs has been limited to advanced-state-of-the-art experimental applications or study programs.

The most commonly used operators for actuators are the various configurations of pilot valves, servovalves, and pressure-divider circuits. As operators also require actuators, the state of the art in operator design is an extension or specialized application of the current knowledge applicable to all designs requiring actuation concepts.

The wide range of environmental and operational conditions for space-flight actuators and operators has resulted in development of an extensive array of concepts and devices for static and dynamic seals. Techniques for welded joints have been developed to eliminate static seals at hardware joints and at interfaces where leakage is critical or where chemical compatibility or severe vibration presents design problems. Considerable effort has been expended in research and development programs investigating metal-to-metal sealing surfaces and fundamental seal leakage phenomena.

Design concepts for the mechanical transmission of actuator power are derived from general machine design experience, with refinements in the use of materials and fasteners to accommodate high vibration levels, exposure to reactive fluids and, in many applications, nonlubricated sliding contact.

Instrumentation for actuators and operators includes electrical position-indicating switches, potentiometers, and transducers and pressure-indicating switches and transducers. Designs for these devices reflect the needs arising from the requirement for exposure to severe environmental and operational conditions.

Progress in the design of the most modern space-flight actuators and operators has resulted from intensive development and refinement of both materials applications and the design details of fundamental actuation concepts. This flight-hardware experience, supplemented with information resulting from advanced-state-of-the-art programs, provides a sound basis for the design of new space-flight hardware.

2.1 ACTUATOR CONFIGURATIONS

Actuators are classified as on-off types for two-position control, or as modulating types for variable-position control. Also, they are classified as direct-acting when they respond directly to system inputs, or as operator-controlled when their response is controlled by an intermediate device, an operator, that responds to system inputs. Additionally, actuators are classified according to salient physical features of the design configuration.

The factors involved in selecting the optimum actuator configuration for a specific application are interrelated with overall control system design considerations beyond the scope of this monograph. However, references 1 through 8 and other technical literature referenced throughout the monograph review the general considerations leading to selection of specific actuator and operator configurations.

In selecting a specific actuator configuration, the first consideration is the energy source, which is determined by overall system considerations. In control systems for rocket engines using kerosene-type fuels, hydraulic actuation subsystems that use engine fuel as the working fluid usually are selected. In storable-propellant systems, propellants have been used for actuation. In control systems for rocket engines using both fuel and oxidizer at cryogenic temperatures, pneumatic-actuation subsystems sometimes have been selected; these systems use a stored supply of helium gas as a working fluid suitable for operation in an extreme range of environmental temperatures. The high force levels required for large-engine gimbal actuators (e.g., those for the S-IC, S-II, and S-IVB boosters) have resulted in the choice of high-pressure hydraulic subsystems with independent electric-motor pump actuators and fluid-temperature control, as described in references 9 through 11. Electrically powered solenoid actuators, torque motors, and rotating motors are in common use for applications requiring response to electrical signals from guidance or logic circuits. The relatively low force levels required for valve actuation in small reaction-control rocket engine subsystems permit direct use of solenoids as valve actuators. High-force-level devices that respond to electrical signals have been obtained with electrically powered operators controlling hydraulic or pneumatic actuators. (By definition, the term "hydraulic" applies to any liquid, and the term "pneumatic" applies to any gas).

2.1.1 Piston Actuators

Linear-motion piston-cylinder actuators are categorized as hydraulic or pneumatic, and as single-acting or double-acting. Piston-cylinder actuators are designed as separately mounted assemblies for thrust-chamber gimballing. Frequently, these actuators are included in valve assemblies for direct positioning of poppet valves, or with rack-and-pinion or bellcrank linear-to-rotary motion transmissions for angular positioning of butterfly and ball valves. Basic configurations for piston-cylinder actuators are illustrated in figure 1. Symbols used therein and elsewhere in the monograph are defined in the Glossary.

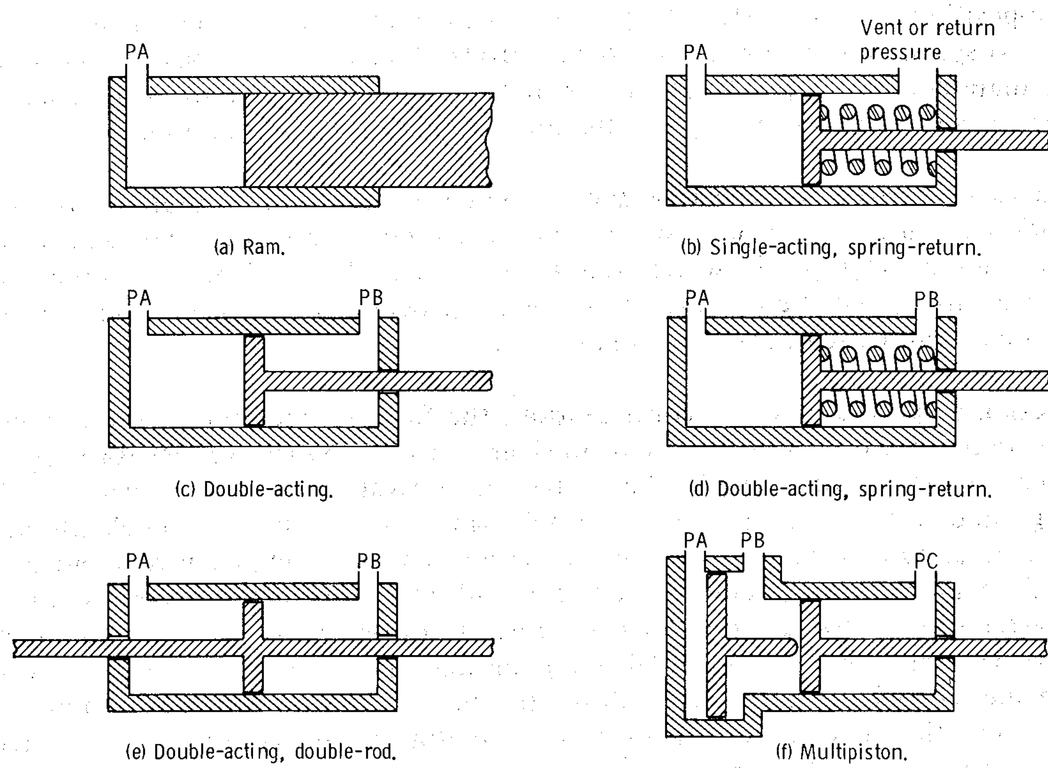

Figure 1. - Schematics of basic configurations for piston actuators.

The ram actuator (fig. 1(a)) appears most often as one end of a pressure-actuated spool valve with return forces applied at the opposing end. The single-acting spring-return piston actuator (fig. 1(b)) frequently is selected as a valve actuator when spring forces are adequate for deactuation. Double-acting piston actuators (fig. 1(c)) are used when forces required for deactuation result in excessive size for an equivalent single-acting spring-return configuration. Helical springs are included in some double-acting piston actuators (fig. 1(d)) for maintaining a preferred position prior to application of pressures. Double-acting double-rod configurations (fig. 1(e)) are used when dynamic response considerations require equal piston areas and equal volumetric displacements on both sides of a piston. Multipiston actuators (fig. 1(f)) are used in applications wherein there is a time delay between partial actuation and full-stroke actuation. Such a delay is required, for example, for the J-2 engine main oxidizer valve during an engine-start sequence of operations.

The most commonly used operator for a single-acting piston actuator is a three-way valve that can apply system supply pressure to the actuator or vent the actuator to system return pressure. The double-acting configuration requires a four-way valve for controlling pressures on both sides of the piston. The choice between a single-acting spring-return actuator and a double-acting actuator requires evaluation of the required force levels and attendant actuator sizes, operator complexity, and system plumbing. Selection of a spring-return actuator may be based on requirements for fail-safe closing or fail-safe opening of a valve.

Piston actuators with dynamic seals to limit piston and piston-rod leakage are characterized by long working strokes, high force levels, and simple design. In applications requiring complete isolation of actuator working fluids, as in the case of piston-actuated valves for reactive fluids, bellows elements sometimes are used as piston-rod dynamic seals. When a bellows seal is impractical because of space or mechanical limitations, a double sliding-contact seal with an intermediate vented isolating cavity sometimes is used.

In applications wherein dynamic-seal friction forces are undesirable, as in actuators for pressure-sensing operators that must actuate at a precise sensed-pressure level, diaphragm or bellows actuators are used in place of piston actuators, as discussed in sections 2.1.2 and 2.1.3.

2.1.1.1 GENERAL CONSIDERATIONS

2.1.1.1.1 Pneumatic Modulating Control Dynamics

Most space-flight actuators and actuation systems have been designed for nonmodulating on-off control of shutoff-valve position and velocity in response to discrete system inputs. Actuators for variable-position modulating control in response to continuous system inputs, as in engine-gimbal actuation systems or in throttle-valve systems, usually are operated with hydraulic fluids to minimize fluid-compressibility effects. Pneumatic modulating actuators are used most frequently for positioning valves in pneumatic pressure-regulating systems that reduce supply pressure or relieve controlled pressure.

As a result of compressibility of the actuation fluid, modulating servovalve-controlled pneumatic actuators frequently exhibit an inherent tendency toward hunting, or oscillation, when they respond to changes in servovalve position or to load changes. This tendency becomes more pronounced when fast response is required and servovalve flow capacity is increased or when sudden load changes must be accommodated and servovalve pressure-change capacity is increased. To achieve a successful design with minimum laboratory effort, an effective dynamic analysis of servovalve and actuator operating characteristics is essential.

Section 2.2.2 and references 12 and 13 discuss the design of servovalves with nonlinear flow-gain and pressure-gain characteristics contoured to contribute to actuator dynamic stability and load-stiffness while satisfying speed-of-response requirements. Static and dynamic friction forces, variable and reversible load forces, subsonic pneumatic flow, servovalve saturation, and actuator position limits introduce nonlinearities and discontinuities that must be evaluated. The use of linearized analytic techniques contributes to an understanding of the inherent problems but generally is ineffective in establishing quantitative design criteria for successful design of dynamic nonlinear valve and actuator systems.

Reference 14 presents a detailed discussion of the characteristics of pneumatic servovalve and actuator systems and a description of analog computer circuits for dynamic analysis. Also described is a technique for the use of plenum volumes, separated from piston-cylinder actuator volumes by flow restrictors, to add compensation to a pneumatic actuation system. This is an effective way to achieve dynamic stability and smooth steady-state operation. A controlled leakage through a double-acting piston can be used for adding dynamic damping to a pneumatic actuation system. In the mechanical design of piston actuators, a minimum piston skirt length usually is dictated by overall actuator design considerations. With little addition to the moving mass of the piston, and with little or no increase in piston skirt length, compensating plenums can be included in a pneumatic actuator design, as shown in figure 2.

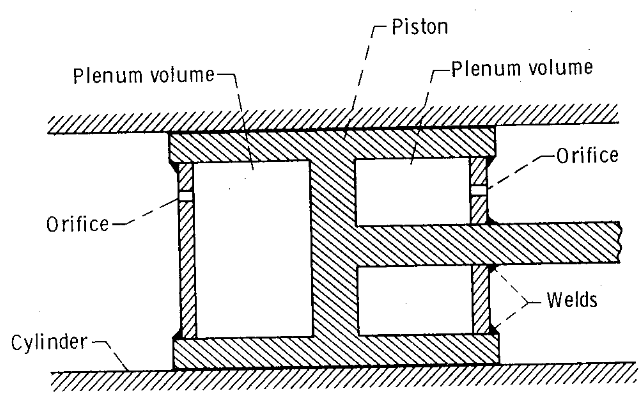

Figure 2. - Schematic of piston actuator with compensating plenum volumes.

Reference 15 presents an introductory description of digital-computer techniques for the design analysis of pneumatic piston actuators. Reference 16 describes a general-purpose digital-computer program used for the design analysis of mechanical-hydraulic-pneumatic control systems and components with nonlinear and discontinuous describing functions. This program originally was developed for dynamic analysis of the complete J-2-engine pneumatic control system and its components. It has been expanded for application to other control systems and components as an effective means for establishing quantitative design requirements.

2.1.1.1.2 Hydraulic Modulating Control Dynamics

The dynamics problems associated with modulating hydraulic actuators designed for servovalve control differ from the dynamics problems with their pneumatic counterparts in that fluid-compressibility effects can be minimized by including self-priming features in the actuators. Actuator seals and guides designed for controlled static and dynamic friction provide Coulomb damping; excessive friction, however, can result in excessive response deadband. The characteristics and critical design features of hydraulic servovalve and actuator systems are discussed in detail in references 17 through 32. To minimize laboratory effort in shaping the characteristics of the closed-loop system and in determining the need for system compensation, a computer program (e.g., ref. 16) is used to analyze the nonlinear and discontinuous characteristics of hydraulic servovalves and actuators.

2.1.1.1.3 On-Off Control Dynamics

In a primed hydraulic system, control of the transient response of an actuator requires accommodation of any difference in the volumetric displacements on either side of the piston as it moves. Figure 3 illustrates a double-acting piston-cylinder actuator in which

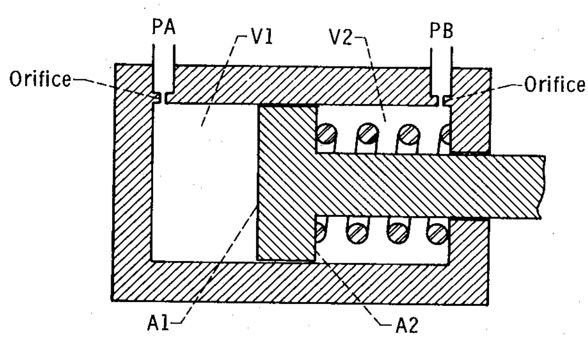

Figure 3. - Schematic of piston actuator with dissimilar active areas and volumes.

there are appreciable differences in the piston effective areas (A1 and A2), in the initial volumes (V1 and V2), and in the swept volumes on each side of the piston face. Actuators of this configuration frequently are controlled by on-off pilot valves; flow restrictors (orifices) of different size for each side of the piston accommodate differences in flowrates. When a servo-operated, symmetrical, four-way valve is required to provide modulating control, significant differences in piston effective areas may result in directional differences in speed of response.

In on-off pneumatic systems, the time required for actuator internal pressures to increase or decay to the levels at which motion commences is additive to the time required for displacement. On each side of the piston, the pressure response time is a function of the flow-restrictor size and the actuator volume. Flow restrictors sized for the desired motion timing may result in an undesired initial pressurization time for breakaway. The time for motion start as well as the time for full-stroke displacement may be different in each direction of motion. Successful design of actuators shown in figure 3 requires sizing of the volume, the piston area, and the flow restrictor on each side of the piston so that the required breakaway time and motion time can be obtained in each direction. Section 2.2.1.3 discusses response problems associated with actuators in which a signal pressure is applied to opposing surfaces to produce a net actuation force that may vary with operating conditions.

2.1.1.1.4 Hydraulic-Actuator Priming

In an hydraulic actuator, any air, other gas, or vapors entrapped in internal cavities or in actuator plumbing may affect repeatable response or actuation system stiffness. The time required for the hydraulic fluid to compress any trapped gas and then fill the volumes initially occupied by the gas is additive to the response time of a fully primed system; uncontrolled variable volumes of residual gas thus result in variations in response time. In modulating hydraulic actuation systems, the presence of compressible fluids has resulted in oscillatory or dynamically unstable operation. Hydraulic actuators frequently are provided with controlled leakage or circulation flow paths to ensure removal of gases or vapors and to maintain a fully primed condition for operation.

2.1.1.1.5 Hydraulic-Fluid Temperature Control

It is essential that heat transfer under operating conditions does not change the actuator fluid temperature beyond its allowable limits. For repeatable speed of response, the actuation-fluid temperature must remain between limits established by the temperature-related effects of fluid viscosity and fluid density on fluid flow and consequently on actuator dynamics. When elastomeric or plastic materials are used in an actuator, the fluid temperature must remain within the allowable temperature limits for these materials.

Figure 4 is a schematic of an hydraulic on-off piston actuator with response timing controlled by flow restrictors at the two control ports. With the piston in the position

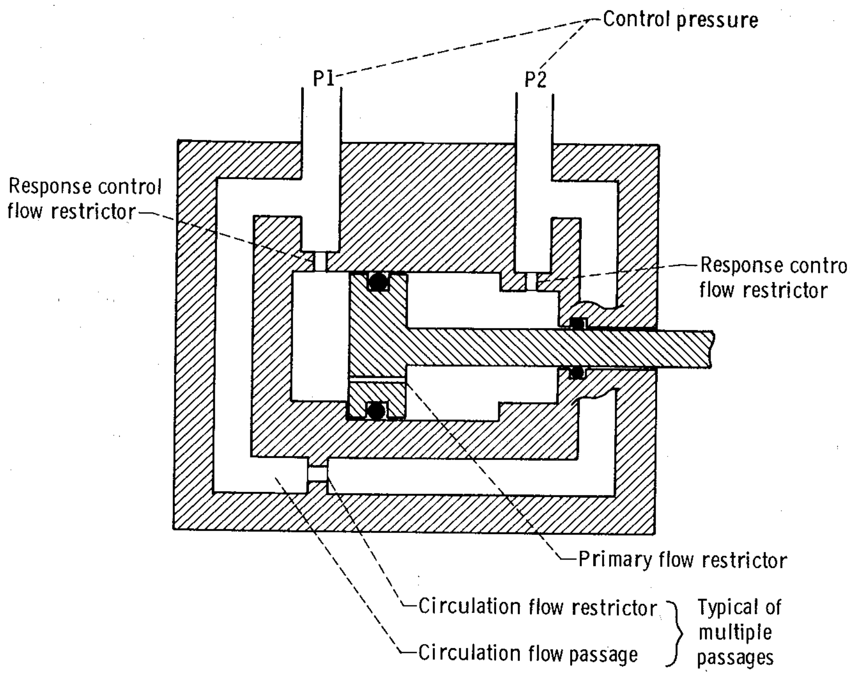

Figure 4. - Flow paths for priming and for fluid-temperature control in a piston actuator.

shown, and with pressure P2 greater than pressure P1, an orifice through the piston controls bleed flow from P2 to P1 for actuator priming. Multiple restricted heat-exchanger passages in the housing permit circulation flow from P2 to P1 for control of actuator and fluid temperatures. In the actuated condition, with pressure P1 greater than pressure P2, bleed flow and circulation flow is obtained with flow from P1 to P2. The illustrated concept requires fluid flow from a temperature-controlled source.

2.1.1.1.6 Safety Requirements for Test and Operation

Proof-pressure tests of each actuator and operator assembly and burst-pressure tests of sample specimens are performed to verify structural integrity and to safeguard personnel during subsequent checkouts, tests, and operation. Proof pressures usually are one and one-half to two times the maximum normal operating pressure, and burst pressures usually exceed two to four times the maximum normal operating pressure, the factor depending on the safety-factor policy established for specific systems. In addition, a safe operating pressure, defined as the maximum allowable pressure without shielding for personnel safety, frequently is established for each system or component.

Adequate leakage tests sometimes have been precluded when components were designed to comply with specified proof-pressure and burst-pressure requirements but were not designed for safe operating pressures that permitted nonshielded testing under normal system operating conditions. System leakage tests requiring the close presence of personnel cannot be conducted at normal operating pressure if a component in that system has a lower safe operating pressure.

Components have leaked or failed structurally when marginally defective components were not detected in tests with a single application of proof pressure. Typically, a seal-weld crack may require repeated pressurizing cycles before it propagates sufficiently for a detectable leak, and a solid-particle contaminant in contact with a diaphragm or bellows may not cause a puncture the first few times that pressure is applied. When a component has been set up for proof-pressure testing, repeating the test a limited number of times does not impose testing problems and does provide greater confidence in the test results. Fracture-mechanics criteria, as established for each design program, are considered in hardware design and in the specification of proof pressures and the number of test cycles.

Mechanical springs in actuators often present personnel safety hazards during assembly and disassembly procedures. When such hazards exist, a metal warning tag is fastened to an actuator in a prominent location to identify the hazard. Ordinarily, these hazards are not present when an actuator is designed so that all springs with significant forces return to their free lengths with the restraining fasteners still engaged.

2.1.1.2 FLOW RESTRICTORS

2.1.1.2.1 Flow Control

Response of hydraulic and pneumatic on-off actuators usually is controlled by an adjustable or replaceable flow restrictor. This restrictor may be a small needle valve or, more commonly, an orifice. Replacement of orifices in a "cut-and-try" approach is an established method of adjusting actuator response times in control systems, but it is a time-consuming and expensive process. The use of orifices often introduces difficulties in obtaining repeatable response among nominally identical control systems. This lack of repeatability has been associated with variations in leakage flow paths in parallel with an orifice and with variations in orifice effective flow area as a result of manufacturing tolerances.

The pneumatic control system for the J-2 engine used precision orifices to control actuator response; this practice achieved the design objective that 9 out of 10 production engines satisfy all timing requirements during the first control-system checkout without orifice replacements. Each orifice is designed for installation with essentially no leakage in parallel with its size-controlled flow path. The effective flow areas of fixed orifices are controlled by the use of precise dimensions, tolerances, and surface finishes. Component-level flow testing of these precision orifices is not required.

Manufacturing costs incurred in obtaining precise critical dimensions and surface finishes are justified by elimination of flow-test costs and by minimizing the costs of checkouts of the engine control system. References 33 through 42 present basic information for sizing flow restrictors.

2.1.1.2.2 Temperature-Effect Compensation

Changes in the rate of flow through a flow restrictor as a result of changes in fluid temperature have produced excessive variations in actuator response. This kind of response problem is particularly acute in pneumatic open-loop actuation systems designed for checkout at standard sea-level atmospheric temperature (288 K) and for subsequent operation with high-temperature or low-temperature gases. The reason is that volumetric flow, and consequently actuator response, varies with the square root of the gas absolute temperature when flow is controlled by a fixed flow restrictor.

Figure 5 illustrates a flow restrictor with temperature-effect compensation that was

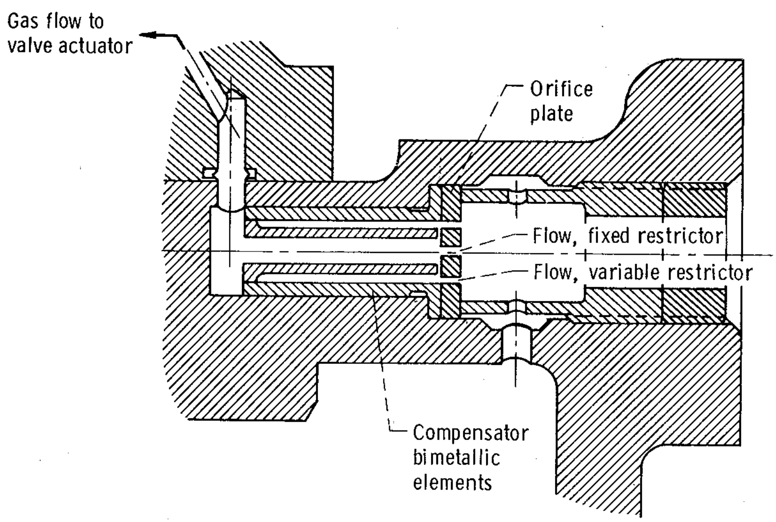

Figure 5. - Schematic of flow restrictor with temperature-effect compensation.

developed for the pneumatic actuators for the J-2 engine main propellant valve. The restrictor is attached to an actuator housing and assumes a temperature approximately equal to the temperature of the actuator. Under system operating conditions, the helium gas actuation-fluid temperature as a result of heat transfer is nearly equal to the actuator temperature. The restrictor effective flow area is thereby automatically preset by actuator temperature when actuation occurs. The restrictor presents parallel flow paths, one through a fixed orifice and one through a clearance controlled by bimetallic elements so that it varies with temperature. The restrictor is designed so that the total effective flow area at 210°R

(117 K)[1] is $(530/210)^{1/2}$ times the area at 530°R (294 K). With a regulated supply pressure, gas volumetric flowrate and actuator response time at 210°R (117 K) is nominally equal to gas volumetric flowrate and actuator response time at 530°R (294 K). The combination of a fixed flow resistance and a flow resistance that varies linearly with temperature provides approximate compensation at intermediate temperatures.

2.1.1.2.3 Contamination

Partial blocking of a response-control flow restrictor by contaminants can alter its flow resistance and thereby change actuator response. The sensitivity of a small orifice to contamination occasionally has been reduced by the use of a number of larger orifices in series having the same overall flow resistance as a single smaller orifice.

If a flow restrictor has its own filter element, accumulation of contaminants at the filter element can alter the flow resistance of the filter/restrictor combination and produce a change in actuator response. A minute amount of moisture in a small filter on occasion has spread through the filter element and, when frozen, has changed the filter flow resistance. Freezing temperatures can occur in a pneumatic system in a sea-level room-temperature environment as a result of gas expansion from a high-pressure source to the pressure level at a system flow restrictor. Despite both precautions intended to prevent entry of moisture into a pneumatic system and procedures for removing moisture, small filters tend to accumulate and retain moisture, and their use in some instances has detracted from, rather than enhanced, the reliability of a flow restrictor.

Hydraulic systems occasionally have been contaminated by particles from filter elements when filters located near flow restrictors were damaged by impingement of high-velocity fluid jets issuing from the flow restrictors. Filters themselves have been a major source of contamination. Filter elements are difficult to clean, and cleaning an element after it has been welded to a casing is even more difficult. Contaminants lodged in filters and fragments of filter elements have migrated during system operation, and created problems they were intended to prevent. The design of filter elements and assemblies is covered in reference 43.

Protection of flow restrictors from contamination requires that overall system cleanliness be maintained and that large-capacity filters be used at strategic locations in the system. Protection against inadvertent introduction of contaminants when components are installed or replaced in a system is a primary concern, as are contaminants that are dislodged or generated in system components during operation. Contamination control during component test usually is achieved when formal specification of requirements for component and test-setup cleanliness and filtration are supplemented by quality control practices for clean test environments.

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[1]Parenthetical units are in the International System of Units (SI units). See Mechtly, E.A.: The International System of Units. Physical Constants and Conversion Factors, Revised. NASA SP-7012, 1969.

Current general specifications for space-flight hydraulic systems (refs. 44 through 46) require the use of a strainer with grid openings of 0.008 to 0.012 in. (0.203 to 0.305 mm) for each hydraulic-system orifice with a diameter of 0.070 in. (1.78 mm) or smaller. In practice, filters frequently are used in place of the specified strainers to obtain finer filtration. These general specifications are interpreted as being applicable only to critical orifices, such as response-control orifices, and not to orifices used for bleed flow or circulation flow.

2.1.1.3 SPRINGS

2.1.1.3.1 General Requirements

Hydraulic and pneumatic piston actuator assemblies are designed both with and without mechanical springs that provide an internal or external force bias. The most common spring form in actuators is the round-wire, helical-coil compression spring. Despite their mechanical simplicity, these springs and their installation present major design problems because of the temperature ranges and vibration environments to which space-flight hardware is exposed.

In many piston-actuator applications, spring problems are avoided when an objective and critical analysis of system and component requirements reveals that a force-bias spring is not necessary. This is often the case when the actuator is double-acting or when dynamic-seal friction forces are large. In some cases, assemblies can be designed so that actuator load forces can be used to eliminate the need for mechanical springs.

When mechanical springs are required, the general design practices for springs and their installation (refs. 47 through 63) must be supplemented as discussed in the following sections. The extent and specialized nature of the published literature on the subject of mechanical springs are evidence of the emphasis that this element requires in successful design.

2.1.1.3.2 Generation of Metal Particles

In the acceleration and vibration environments associated with space-flight hardware, mechanical-spring deflection and vibration have generated metal particle contaminants that have damaged dynamic seals and sealing surfaces, partially blocked response-control flow restrictors, or damaged other system components to which the particles migrated. Cyclic relative motion between helical-spring end coils and their retainers, within the constraints of the guide clearances provided for assembly, is an inherent producer of fine metal particles. Thin sections and feather edges resulting from flat grinding of helical-spring end coils provide sources for the breakout of relatively large particles. Metal particles also have been generated by scrubbing contact between helical-spring coils and adjacent parts as a result of vibration-induced lateral deflections of the springs. This problem is compounded when helical springs are installed concentrically, one within another, with envelope limitations on radial clearances between coils. The design of spring-loaded assemblies that will not generate metal particles requires specific emphasis on clearances, end-coil design, and end-coil retention features.

2.1.1.3.3 Helical-Spring Rotation

In the design of linear-motion piston actuators with helical springs, it is necessary to consider the effects of the coil rotation that accompanies spring axial compression (ref. 48, pp. 247-254). If a helical spring is compressed with end-coil rotation restrained, the coil rotation about the spring axis must result in angular displacement of one or both end coils or in distortion of the spring. When rotational forces are stored in a distorted spring, actuator operation or exposure to shock or vibration may result in end-coil rotational slippage. This slippage changes the spring installed load. This design consideration is especially significant in devices that are preset to actuate at a specified pressure level. In addition, end-coil rotation and sliding contact with end-coil retainers, during installation or during operation, can generate metal particle contaminants. Low-friction ball, roller, or needle thrust bearings frequently are installed in compression at spring end coils so that the coil can rotate with negligible angular restraint and with rolling rather than sliding relative motion.

2.1.1.4 MATERIALS

2.1.1.4.1 General

The most common problems with materials in space-flight actuators have been related to corrosion, chemical compatibility, hydrogen embrittlement, welding effects, fatigue of flexing elements, cracking failures in castings, porosity, impact sensitivity, and abrasion or wear at bearing and dynamic-seal contact surfaces.

In most applications, adequate corrosion resistance and chemical compatibility are obtained by the use of aluminum alloys, 300-series CRES alloys, nickel alloys, nonreactive plastics, and corrosion-resistant platings on vulnerable metals. The most severe corrosion and compatibility problems occur in actuators with ferromagnetic materials or materials exposed to reactive fluids. Materials problems and practices associated with specific types of actuator and operator elements are discussed in separate sections throughout this monograph.

Aluminum alloys for general use are chromic-acid anodized to enhance corrosion resistance. Surfaces exposed to dynamic contact sometimes are hard anodized. Complete parts, such as housings, frequently are sulphuric-acid anodized when wear resistance at a dynamic contact surface is required.

The corrosion resistance of metal alloys is affected significantly by the smoothness of surface finishes. A 125 μ in. (3.18 μm) AA finish, carried over from industrial practice, has not been adequate in obtaining good corrosion resistance in the environmental conditions to which space-flight hardware is exposed. Salt-spray tests, sand and dust tests, reactive fluids, and moisture in operating fluids have resulted in a requirement for a 63 μ in. (1.60 μm) AA finish, or smoother, for all machined surfaces.

Material cracking failures attributed to long-term stress corrosion have occurred in certain alloys subjected to sustained stress. As a result, these alloys are now in disfavor for use in space-flight hardware. Table I lists alloys with known susceptibility to stress-corrosion cracking.

Hydrogen embrittlement of flexing elements is discussed in sections 2.1.2.4 and 2.1.3.4. These sections also review nonmetallic materials for diaphragms, welding problems associated with bellows, and fatigue of flexing elements.

To minimize machining costs, actuator housings often are made from castings. The complex configurations obtainable with castings introduce potential cracking problems associated with changes in cross section, casting defects, external loads, vibration, and thermal stresses. Solutions to cracking problems have included changes in casting design, changes in the design of mounting and support brackets, and changes in the plumbing system to reduce applied loads or vibration loads.

Leakage attributable to porous or defective materials has not been limited to castings. Leakage through titanium-carbide inclusions in 321 CRES bar stock has occurred in helium systems. To minimize the presence of inclusions in thin-wall actuators, consumable-electrode-melted 321 CRES, designated as a vacuum-melted alloy, often is specified.

Material deformation or fracture resulting from impact loads and material abrasion or wear at bearing and dynamic-seal contact surfaces are discussed in sections 2.3 and 2.4.

2.1.1.4.2 Helical-Spring Material

Helical-spring materials for space-flight service generally are limited to the corrosion-resistant metal alloys. The most often used are 321 CRES, Inconel X, Inconel 718, and Elgiloy. Rene 41, A286, and Refractaloy 26 have been used in components exposed to elevated temperatures. Precipitation-hardening stainless steels 17-4PH and 17-7PH have been used extensively, but are subject to long-term stress corrosion in the conditions noted in table I.

In assemblies wherein spring compression is adjusted to provide a reference force having narrow limits, thermal effects can be critical. The design of helical springs for these assemblies frequently requires use of a metal alloy such as Ni-Span-C so that there is little temperature-related variation in mechanical spring rate and consequently in installed force.

TABLE I. – Susceptibility of Various Metals to Stress-Corrosion Cracking

A. Aluminum alloys with low resistance to stress-corrosion cracking[1]

Wrought Alloy	Wrought Temper	Cast Alloy	Cast Temper
2014	All tempers	195,B195	T6
2024	T3,T351,T36,T4	220	T4
2219	T31,T351,T37	Ternalloy 7	T6
7001	T6,T651	40E	As cast
7075	T6,T651		
7079	T6,T651		
7178	T6,T651		

B. Steels with moderate resistance to stress-corrosion cracking[2]

Alloy	Temper
Low-alloy	150 to 180 ksi[3] yield strength
Maraging	<200 ksi[4] yield strength
400 series	All tempers
PH14-8 Mo	SRH950 to 1050
PH13-8 Mo	H900 to 1000
15-5PH	H900 to 1000
17-4PH	H900 to 1000
AM 355	>SCT900

C. Steels with very low resistance to stress-corrosion cracking

Alloy	Temper
17-7PH	All except CH900
PH15-7 Mo	All except CH900
AM 355	<SCT900
H-11	All tempers
Vascojet 1000	All tempers
Low-alloy	>180 ksi yield strength
18 Ni maraging	>200 ksi yield strength

[1] Use of material listed is permissible if sheet material, unmachined extrusions, or unmachined plate material is required.

[2] Alloys and heat treatments listed may be used if it can be shown that there are no sustained tensile stresses (residual or imposed).

[3] 1034 to 1241 MN/m^2

[4] $<1379\ MN/m^2$

This kind of alloy can be heat treated to obtain a thermoelastic coefficient that is essentially zero (i.e., spring rate does not change with temperature). The thermoelastic coefficient includes the effects of temperature-related changes in spring geometry and in the torsional modulus of elasticity. Because temperature-dependent variations in torsional modulus of elasticity are the predominant effects, alloys like Ni-Span-C often are referred to as constant-modulus materials. The design of actuators for temperature-effect compensation is reviewed in section 2.1.2.3.2.

As a general practice, safety factors are applied to the nominal values of material yield strength to allow for variations in material properties, for the effects of forming, and for the effects of dynamic loading. Stress relaxation, resulting in reduced installed force, has occurred in Ni-Span-C springs under vibration conditions; this behavior indicates a need for more conservative factors of safety for this material than for other spring alloys. Satisfactory performance of Ni-Span-C springs has been obtained when the solid-height maximum fiber stress was limited to the range of 60,000 to 80,000 psi (414 to 552 MN/m^2), with 60,000 psi (414 MN/m^2) as the limit in applications wherein spring vibration is anticipated. Temperature-effect compensation with Ni-Span-C springs has not been achieved when heat treatment failed to produce the desired constant-spring-rate characteristic. Spring-rate tests of samples from each production lot of springs, performed at several specified temperatures, sometimes are specified to supplement conventional room-temperature inspection procedures.

2.1.2 Diaphragm Actuators

Linear-displacement diaphragm actuators are piston actuators with flexing rather than sliding dynamic seals and are used primarily as pressure-sensing elements with small working displacements (usually less than 0.06 in. (1.52 mm)). Diaphragms may be multiple-ply or single-ply, metal or nonmetal, preformed or formed in place. Flat diaphragms and diaphragms with a single annular convolution are the favored configurations because they are the simplest to fabricate and are the most predictable in performance. Diaphragm actuators offer the advantages of leak-tight frictionless sealing. When these advantages are desired in an actuator with working stroke requirements that exceed the displacement limitations of diaphragm elements, then bellows actuators are selected (sec. 2.1.3).

2.1.2.1 GENERAL CONSIDERATIONS

2.1.2.1.1 Fluid Inflow Effects

Reliable operation of diaphragm actuators is achieved only when the design provides for protection of the thin diaphragm elements. High-velocity fluid jets impinging on diaphragm elements have caused impact and erosion damage. In pneumatic systems, actuator inflow during initial pressurization or during a displacement transient may reach sonic velocity. Under this condition, diaphragms have been ruptured by the impact of solid-particle contaminants.

In actuators with small clearances between a movable diaphragm and fixed surfaces, fluid inflow has deposited solid-particle contaminants in the clearances. Subsequent clearance changes accompanying diaphragm flexing have resulted in diaphragm damage or rupture when contaminants were imbedded in thin diaphragms. Inadvertent formation of ice in small-clearance diaphragm actuators has had the same damaging effects as an inflow of solid-particle contaminants.

Diaphragm actuators frequently are designed with flow passages large enough to ensure low-velocity inflow under all initial-pressurizing and operating conditions. The flow through restrictors often is directed against actuator walls. Close clearance or contact between movable diaphragm surfaces and fixed surfaces is avoided whenever feasible.

2.1.2.1.2 Vibration

Diaphragm structural failures attributed to an excessive number of flexure cycles have occurred most frequently in modulating diaphragm actuators. In these actuators, unimpaired fatigue life is essential, but the number of deflection cycles or partial cycles that must be endured may be unpredictable. Diaphragms in modulating actuators are especially sensitive to dimensional control of the corner radii on the clamping parts and to contamination. Resonance of reference springs under operating conditions occasionally has required a change in spring rate or in the masses of moving elements or both in order to alter the resonant frequencies. Diaphragm pressurizing cavities often are designed with orificed porting for snubbing of pressure transients and for dashpot damping of diaphragm oscillations. Extensive vibration testing during development of actuators is necessary to ensure adequate endurance capabilities.

2.1.2.1.3 Installation Mismatch

When a diaphragm is installed in an actuator assembly, correct positioning of the diaphragm with respect to contacting parts is essential for reliable operation. The assembly problem is compounded when the diaphragm has a center hole and clamping of both inner and outer edges is required.

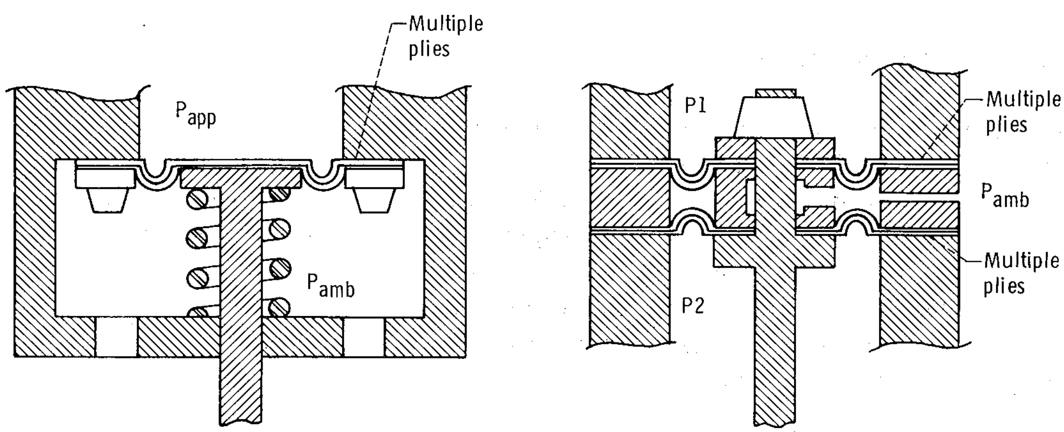
The primary problem resulting from diaphragm mismatch is a reduction in cycle-life capability. Diaphragm convolutions have been damaged mechanically by improper fit with retainers. Diaphragms have been damaged by applied pressures that partially reformed previously formed convolutions. Eccentric installation has resulted in nonuniform diaphragm stress distributions. Flexing contact between diaphragm convolutions and retainers has resulted in abrasion failures.

When preformed metal diaphragms are used, the rigidity of the material assists in aligning inner and outer clamping surfaces. When preformed pliable nonmetal diaphragms are used, greater care in assembly is required to prevent mismatch. Assembly mismatch problems with these diaphragms frequently are circumvented by the use of diaphragms that are flat initially, the convolutions being formed in place by the application of controlled pressures following installation.

Dimension and tolerance stackup drawings, often 10-times actual size or greater, are used to evaluate all conditions that affect matching either a metal or nonmetal diaphragm to its adjacent parts.

2.1.2.1.4 Interply Pressures

When multiple-ply diaphragm elements are used, the interply voids must be protected from exposure to applied pressures, especially when one of the plies is not capable of withstanding the maximum pressure differential that may be encountered. Leakage of actuation pressure into interply voids may result in loss of redundancy or in possible rupture of a diaphragm ply. Such leakage may be trapped between the plies when actuation pressure is removed, the result being distortion of the diaphragm and loss of the intended diaphragm configuration. Figure 6 presents two approaches for avoiding this problem.



(a) Periphery vented.

(b) Peripheries and center holes vented.

Figure 6. - Schematics of multiple-ply diaphragms with exposed edges vented to ambient pressure.

Figure 6(a) illustrates a design concept that utilizes a multiple-ply diaphragm as a static seal, the periphery of the diaphragm being exposed to ambient reference pressure P_{amb} only. Figure 6(b) illustrates a design concept in which multiple-ply diaphragm elements (back-to-back configuration) are exposed to opposing applied pressures P1 and P2. Surfaces near the outer diameters of the diaphragms are used for static sealing, with the outer edges exposed to ambient pressure only. Surfaces near the diaphragm center holes are used for static sealing, with an intermediate spacer designed to vent the inner edges.

2.1.2.1.5 Convolution Reversals

Formed diaphragms, or diaphragms in which some forming may occur during test and operation, may be damaged or reduced in cycle-life capability if differential-pressure reversals result in deformation of diaphragm convolutions. The rigidity of metal diaphragms permits limited pressure reversals without damaging deformation; formed nonmetal diaphragms, however, will be distorted with very little pressure reversal. The back-to-back diaphragm configuration (fig. 6(b)), with a vented intermediate space, frequently is used to preclude convolution reversals.

2.1.2.1.6 Overstroking

If axial displacement of a diaphragm exceeds its maximum allowable displacement, the diaphragm may be distorted or its cycle-life capability may be reduced. Although diaphragm axial displacement may be limited by the actuation mechanism in a completed assembly, inadvertent overstroking frequently occurs in partially completed assemblies. Diaphragm subassemblies, therefore, sometimes are designed with stroke-limiting stops that protect the diaphragm when the subassembly is separated from stroke-limiting features in a complete assembly.

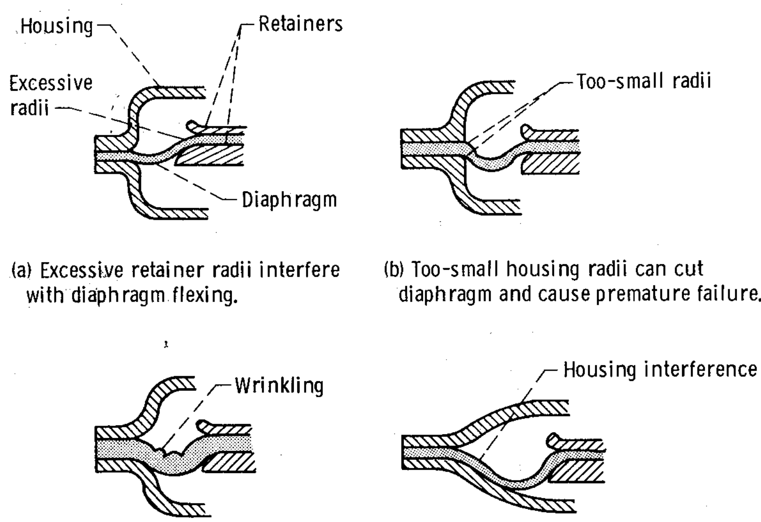
2.1.2.2 DIAPHRAGM ELEMENTS

2.1.2.2.1 General Requirements

The various diaphragm configurations and the general design practices related to their applications are illustrated and discussed in handbooks and design manuals such as reference 3. For pliable nonmetal diaphragms, in comparison with metal, mechanical spring rates are low, flexure stresses resulting from actuator displacement are low, and the most significant stresses are the clamping and applied-pressure stresses. For metal diaphragms, mechanical spring rates and the flexure stresses accompanying displacement are significantly high, and combined stresses resulting from installation displacement, working-stroke displacement, and applied pressures must be considered.

References 64 through 72 describe analysis, design, fabrication, and installation techniques that have been developed for flat and single-convolution metal and nonmetal diaphragms. Localized changes in diaphragm material properties and material deformations resulting from the forming of convolutions that are deep, relative to the material thickness, greatly affect the accuracy of an analysis. For practical design purposes, there is heavy reliance on empirical data, experience with existing similar designs, and testing of alternate configurations.

Figure 7 (ref. 73) illustrates some basic and recurrent problems in design of single-convolution diaphragms and their retainers. In figure 7(a), overly generous retainer



(a) Excessive retainer radii interfere with diaphragm flexing.

(b) Too-small housing radii can cut diaphragm and cause premature failure.

(c) Excessive diaphragm thickness can cause wrinkling.

(d) Housing interference can alter diaphragm loading.

Figure 7. - Sketches of basic problems in design of single-convolution diaphragms (ref. 73).

radii interfere with diaphragm flexing to the extent that diaphragm axial working deflection is accompanied by scuffing contact with retainer radii. As shown in figure 7(b), sharp corners or too-small retainer radii can produce stress concentrations and diaphragm cutting at the sharp edges, and can result in premature fatigue failures. As shown in figure 7(c), excessive diaphragm material thickness can result in wrinkling. Figure 7(d) illustrates housing interference with full-stroke diaphragm deflection. These problems can be avoided through design review of large-scale drawings that show a diaphragm actuator in its extreme displacement positions (allowing for fabrication tolerances).

2.1.2.2.2 Interply and Interface Contamination

When multiple-ply diaphragm elements are used, interply contaminants may produce abrasion or rupture when a diaphragm is exposed to pressure, displacement, or vibration. Nonmetal diaphragm materials frequently are used in multiple-ply configurations for greater strength. They are, however, more susceptible to damage by contamination than are the more abrasion-resistant and puncture-resistant metals. Contamination at the interfaces with retaining parts has occurred with single-ply as well as with multiple-ply diaphragms. Metal diaphragms that are welded to retainers are especially susceptible to contamination because of the potential for trapping weld spatter or other contaminants that are not detectable by visual inspection. Formal procedures for forming, processing, and installing diaphragms are often specified as a design contribution to effective cleanliness control.

2.1.2.2.3 Diaphragm Forming

2.1.2.2.3.1 Die Forming

Control of critical dimensions and of performance characteristics of a preformed diaphragm is dependent on the configuration of the forming dies, the diaphragm material and its processing, and the forming procedure. Changes in tooling, materials, material processing, or forming procedures have reduced diaphragm endurance capabilities or have altered diaphragm effective area and spring rate. A change in die geometry or surface finish, or a change in forming procedure, can affect diaphragm critical dimensions, especially in the vicinity of small radii. Material substitutions, such as the use of ground stock when rolled stock is specified, or the use of cold-worked material when fully annealed material is specified, can affect diaphragm structural and functional characteristics.

References 64 and 65 present an illustrated discussion of metal-diaphragm hydroforming techniques. Testing of representative samples of preformed diaphragms, in prototype hardware as well as in fixtures, is customary in supplementing thorough inspection in proving the adequacy of a particular set of dies.

Formally specified fabrication and quality control procedures are implemented to maintain proven dies and ensure consistent production. Diaphragm dimensional variations can be detected by physical inspection. Variations in localized material properties can be detected only by destructive testing of sample diaphragms from each production lot.

2.1.2.2.3.2 Forming in Place

Convolution damage resulting from mismatch between preformed diaphragms and their clamping surfaces sometimes can be precluded by the use of initially flat diaphragms that are formed subsequent to installation. The actuator assembly is used as the diaphragm forming fixture. Minor misalignment of inner and outer peripheral clamps is accommodated by forming a nonuniform convolution that is matched to the assembly. The diaphragm is formed by applying actuation pressure so that both pressure and actuator stroke contribute to forming the diaphragm. This forming method is especially applicable when plastic diaphragm materials are used. However, when proof or repeated operational pressure is applied, a preformed plastic diaphragm may partially reform to match the assembly, with consequent convolution damage. Diaphragm damage also has occurred when surge pressures exceeded the forming pressure. Consequently, the use of formed-in-place diaphragms is confined to those applications wherein the maximum predicted pressure is well below the forming pressure.

2.1.2.3 SPRINGS

2.1.2.3.1 Spring-Force Control

Springs that apply lateral forces or nonuniformly distributed axial forces to a diaphragm have produced excessive friction and hysteresis in an assembly. They also have distorted diaphragms and reduced cycle-life capability. These problems usually have been associated with springs that were inadequately guided, or with springs that did not retain end-coil squareness when compressed to installed height. Corrective action in such cases has been to fabricate a spring for end-coil squareness with the axis at the installed height, with no specification for free-length squareness.

Reference 74 is a specification that supplements drawings for helical compression springs in establishing detailed design and quality control requirements. Each spring is identified by its drawing as a Class A, Class B, or Class C spring. Class A and Class B spring end coils are ground flat within 0.005 in./in. (0.127 mm/mm) of spring outside diameter; Class C spring end coils are ground flat within 0.010 in./in. (0.254 mm/mm). Spring end coils must be square with the spring axis within 1/2° at installed height for Class A springs, within 1° at installed height for Class B springs, and within 3° at free length for Class C springs. The flat-ground end-coil area must encompass 270° to 300° for Class A and Class B springs, whereas 250° to 300° is allowed for Class C springs. Reference 74 also includes detailed requirements for (1) finish on end-coil tips so that tip structural integrity is maintained, (2) deburring, (3) uniformity of coil pitch, (4) spring straightness and coil concentricity, and (5) general quality control. A single specification document, supplementing all drawings for helical compression springs, permits greater detail in specifying design features and quality control procedures than is feasible on individual drawings and ensures uniformity in spring specifications.

Machined springs or flexures in a variety of configurations have been used for precision reference springs in unique situations, usually in actuators for operators or for instrumentation. Precision springs, machined or formed, may require measurement of geometric and functional parameters such as end-coil squareness and spring rate at specified increments of deflection throughout the operating range.

Belleville springs designed and installed for operation with negative mechanical spring rates are discussed in section 2.1.5.2

2.1.2.3.2 Temperature-Effect Compensation

The effects of actuator temperatures and temperature distributions on mechanical spring forces and spring rates are discussed in section 2.1.1.4.2. Compensation for temperature effects in helical springs is simplified by the availability of spring materials with spring rates that are nearly constant throughout a wide temperature range. Unfortunately, materials available for forming metal diaphragms have significant temperature-related variations in moduli of elasticity. Consequently, metal diaphragms have corresponding significant variations in spring rates. This variability can result in significant changes in diaphragm installed force and deflection force.

References 64 and 65 describe temperature-effect compensation in an assembly in which positive spring rates of a metal diaphragm and a helical spring are offset by the negative spring rate of a Belleville spring. The Belleville spring can be adjusted for operation in the deflection region in which its spring rate is variable, and thus provide an adjustable combined spring rate to compensate for spring-rate tolerances as well as for temperature effects. Section 2.1.5.2 describes and illustrates the use of negative-rate Belleville springs for actuators with adjustable spring rates. Reference 75 reviews a design concept to accommodate temperature transients and to provide compensation adjustability in each assembly in allowing for positive spring-rate tolerances.

Figure 8 presents a simple design concept used to compensate for temperature effects by

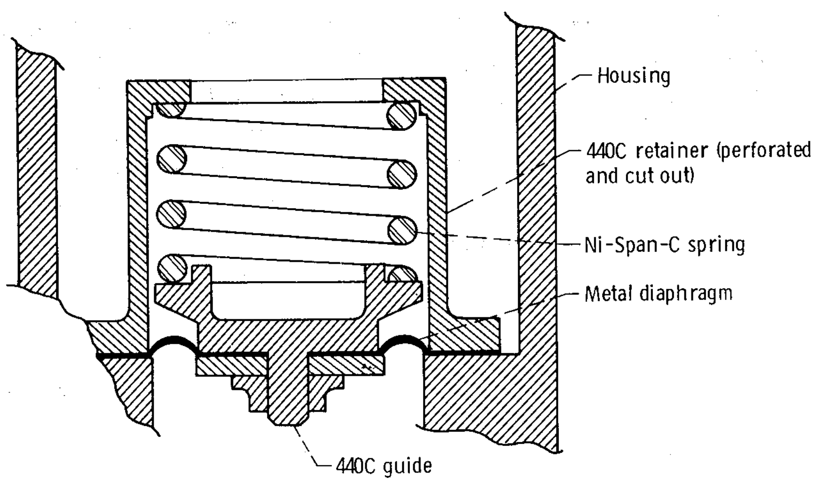


Figure 8. - Schematic of temperature-effect compensation of reference spring.

minimizing variations in diaphragm-actuator mechanical spring forces under transient thermal conditions over a wide temperature range. The figure illustrates the physical arrangement of the parts – the design depends for its effectiveness on the special properties of the materials identified in the figure.

The metal diaphragm is installed in its nominally neutral position so that it contributes little or no mechanical spring force. A helical spring made from Ni-Span-C spring material, heat treated for a nearly constant spring rate throughout the operating temperature range, provides the actuator reference force. The spring guide and spring retainer are made from

440C that is heat treated for hardness and has a thermal coefficient of expansion nearly identical to the coefficient for Ni-Span-C. The spring retainer is designed with perforations and cutout sections so that its transient-temperature response is comparable to that of the helical spring. The spring and its retainer are designed to maintain nearly equal temperatures and to expand and contract together under all service conditions, independently of the housing temperature distribution.

2.1.2.4 MATERIALS

Diaphragm material, either metal or nonmetal, is selected on the bases of (1) compatibility with the fluids to which it may be exposed, (2) strength and fatigue properties throughout the expected temperature range, (3) mechanical spring rate, (4) forming properties, and (5) welding properties when welding is required.

For metal diaphragms, the most commonly used materials are 321 and 347 CRES. Diaphragms have been formed with 17-7PH, but the use of this material is now in disfavor as a result of its susceptibility to long-term stress corrosion. Diaphragms made from 17-7PH and exposed to hydrogen have exhibited drastic reductions in cycle life as a result of hydrogen embrittlement. Inconel 718 diaphragms have been used successfully with hydrogen exposure. The use of Inconel 718 as a diaphragm material is increasing as a result of its good forming and welding properties, its compatibility with reactive fluids, and its relatively high strength and fatigue resistance.

The most commonly used nonmetal diaphragm materials for space-flight service are Mylar, Teflon, and Kel-F. Mylar is especially suitable for cryogenic applications, and it has been used extensively for diaphragms with low spring rates at temperatures as low as -320°F (78 K). Although Mylar has been used successfully in lip seals exposed to liquid hydrogen at temperatures in the vicinity of -420°F (22 K), its reliability for use as a diaphragm material with adequate cycle life at such low temperatures has not been verified. Diaphragms in which plastics or elastomers are bonded to a reinforcing material such as fiberglass have been subject to failures resulting from localized separation into plies and from pinhole leaks that led to internal pressurization of layered material.

2.1.3 Bellows Actuators

2.1.3.1 GENERAL CONSIDERATIONS

Bellows in space-flight actuator assemblies are used primarily as pressure-sensing elements for pressure-actuated valves or instruments and as piston-rod dynamic seals. A linear-displacement bellows actuator is a piston actuator with a flexing rather than a sliding dynamic seal. Bellows are used in place of diaphragms when the required working stroke exceeds the displacement limitations of diaphragms, or when mechanical spring rates lower than those of metal diaphragms are desired. Only metal bellows elements are used in space-flight hardware.

2.1.3.1.1 Fluid Inflow Effects

As discussed in section 2.1.2.1.1, thin-wall elements such as diaphragms and bellows require protection against direct impingement of fluid inflow during initial pressurization and during displacement transients. In bellows applications wherein installation or stroking results in small clearances between convolutions, failures have resulted from an inflow of solid-particle contaminants and, in some cases, from the formation of ice. Protective measures employed are similar to those used for diaphragms.

2.1.3.1.2 Fluid-Flow Dynamic Forces

Bellows have failed from fatigue when bellows elements were exposed to continuing fluid-flow dynamic forces. This condition can occur in a pressure-actuated valve assembly when the bellows separates the actuator cavity from the valve effluent and when the bellows is exposed to turbulent flow through the valve. This condition also can occur when a pressure-sensing bellows in an actuator operator is exposed to turbulent flow or to oscillatory fluid pressures. Fatigue failures under these conditions result from the cyclic deflection of convolutions in response to driving forces. Bellows have failed when surge or water-hammer pressures and accompanying transient stresses have been greater than anticipated and have stressed the material beyond yield. Bellows elements sometimes are enclosed in protective cylindrical sleeves with orifices for snubbing pressure transients.

2.1.3.1.3 Fatigue Life

The design requirements for an actuator usually specify a minimum number of full-stroke cycles and a schedule of vibration inputs for which endurance capability must be demonstrated. A bellows often is designed so that its resonant frequencies are above the range of driving frequencies with significant energy. In cases wherein a bellows is exposed to a liquid, the mass of the liquid in the convolutions is additive to the bellows mass in the computation of axial natural frequencies, and the total liquid mass contained by the bellows is additive to the bellows mass in the computation of transverse natural frequencies.

Bellows fatigue failures resulting from vibration have been eliminated by the addition of damping vanes or articulated bellows guides designed for dashpot damping. Hysteresis and deadband in force-displacement characteristics accompanying bellows guide friction have been minimized by the use of Teflon bushings or film lubrication at guide contact surfaces.

Bellows with corrugations welded to each other exhibit much greater variation in fatigue life than do formed bellows, and formed bellows therefore generally are more reliable.

2.1.3.1.4 Overstroking

If displacement of a bellows exceeds the maximum displacement for which it was designed, the bellows may be distorted, cycle-life capability thereby being reduced. Although bellows axial displacement may be limited by the actuation mechanism in a completed assembly, inadvertent overstroking frequently is possible in a partially completed assembly. Bellows element subassemblies often are designed with stroke-limiting stops in the subassembly to preclude overstroking prior to installation. Otherwise, careful handling and installation procedures are required. Differences in displacement capabilities of formed and welded bellows are discussed in section 2.1.3.2.3.

2.1.3.2 BELLOWS ELEMENTS

2.1.3.2.1 General Requirements

Design configurations for metal bellows are illustrated and discussed in handbooks and design manuals such as reference 3. Analytic techniques and design practices reviewed in references 69, 70, and 76 through 78 generally are applicable when supplemented by the results of experience with space-flight hardware.

The least predictable aspect of bellows design is operational cycle-life endurance capability. This difficulty exists primarily because the influence of resonant or forced vibration is not exactly predictable. Obtaining structural properties in metal bellows that are consistent with the functional and reliability requirements of a specific application frequently requires iterative design and development effort.

2.1.3.2.2 Interply Contamination

Both single-ply and multiple-ply metal bellows are in common use. Multiple-ply bellows elements are rated for higher allowable pressures than are single-ply elements with the same mechanical spring rate. Multiple-ply bellows elements have been used to satisfy redundancy requirements in critical applications. However, the theoretical advantages of multiple-ply elements over single-ply elements in obtaining smaller envelope size or greater reliability are offset by physical disadvantages. The major problem with multiple-ply elements is the difficulty in obtaining and maintaining the required degree of interply cleanliness throughout the processes of fabricating the elements and welding the elements to their end fittings. Interply contaminants can result in abrasion and rupture of bellows elements during operation or exposure to vibration (refs. 79 through 81).

The use of cleaning procedures originally specified for parts that will be exposed to liquid oxygen has been beneficial in obtaining the type and level of cleanliness required for successful bellows fabrication. These procedures are followed as general practice during fabrication, although other cleaning procedures may be specified for the finished product. Recleaning of bellows assembly parts is required whenever handling may introduce contamination.

2.1.3.2.3 Fabrication Effects

Metal bellows elements are fabricated in any one of several different ways: (1) mechanical or hydraulic die-forming, (2) machining, (3) deposition, or (4) welding of individual corrugations to each other.

For a given application, bellows with welded corrugations exhibit lower mechanical spring rates, have greater allowable working strokes, and are more compact than formed bellows. The welded types, however, are difficult to clean and inspect; they are subject to failures resulting from weld spatter, weld slag, or other contaminants lodged between closely nested corrugations; and they are subject to failures resulting from stress concentrations at the notches of the weld beads. Changes in material properties in the weld-heat-affected zones of bellows, as well as improper heat treatment, have resulted in premature fatigue failures. Automatically controlled welding equipment is required for adequate control of the welding process and the convolution-edge weld geometry. Bellows elements have been welded directly to actuator housings and other actuator elements but frequently are welded to end fittings for subsequent welding or fastening.

Formed bellows frequently are installed in an actuator so that for a full stroke of the piston approximately one-half the bellows deflection will be in compression and one-half in extension. This installation results in little or no impairment of fatigue life, and a formed bellows with axial length shorter than that of a bellows in compression only can be used. The fatigue life of welded bellows, however, is reduced drastically (ref. 70, Appendix n) when the bellows is deflected in extension. Nominally identical welded bellows also exhibit great variation in fatigue life as a result of variations in the welding process and variations in welding effects on materials properties. Tests indicate that increasing the internal pressure will decrease the fatigue life of welded bellows. Internal pressures, within limits that preclude bellows squirm (sec. 2.1.3.2.4), have negligible effect on the fatigue life of formed bellows.

Bellows elements have been corroded when strict cleanliness controls were not maintained before, during, and immediately subsequent to welding. Crevices at slip-on joints at end fittings require particular attention because they can trap contaminants or operating fluids that promote chemical or galvanic corrosion.

Brazing and soldering are seldom used in fabricating space-flight bellows because the required flux materials can cause corrosion if not completely removed.

2.1.3.2.4 Bellows Squirm

Bellows elements in actuator assemblies tend to be distorted by an internal pressure greater than the external pressure; this distortion is called squirm. For typical actuator bellows, the allowable pressure difference with internal pressure higher, as limited by squirming, is drastically less than the allowable pressure difference with external pressure higher. The problem of squirm is similar to the column-buckling problems encountered with long, slender bellows elements. In applications wherein internally applied pressures may result in squirm failures of simple bellows elements, or when buckling is a potential problem, the use of segmented elements with guides (articulated bellows) may be required. Figure 9 illustrates a typical articulated bellows in an actuator assembly.

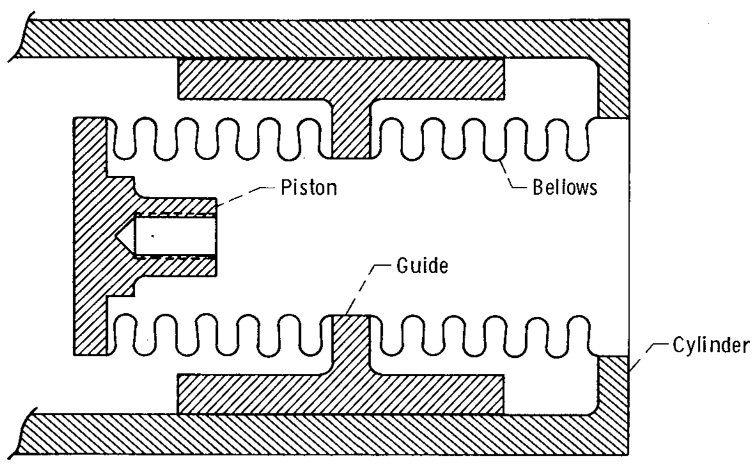


Figure 9. - Sketch of articulated bellows in actuator.

2.1.3.2.5 Quality Control

The thin sheet stock sizes used for bellows fabrication introduce functional sensitivity to minor surface irregularities and to corrosion. A 0.001-in. (0.025 mm)-deep scratch in a 0.005-in. (0.127 mm)-thick bellows element has the effect of a deep notch. Shallow scratches, dents, die marks, asperities, and corrosion spots can concentrate cyclic stresses and initiate structural failures. Bellows corrosion has resulted from failure to remove forming oils, soap, or detergent solutions and other fluids used during fabrication and testing. Fingerprint corrosion frequently occurs on high quality surfaces. Inadequate cleanliness control prior to, during, and subsequent to welding operations and heat treatment has resulted in rust pinholes and other forms of corrosion. Inadequate packaging, handling, and installation procedures have been a frequent cause of bellows distortion and other mechanical damage.

Formed bellows usually are subjected to visual inspection, leak tests, deflection tests, and proof pressure tests so that the quality of each unit may be verified. These limited checks are inadequate for bellows with welded convolutions. Visual inspection has been supplemented by dye-penetrant tests but, to prevent corrosion, all traces of penetrants must be removed prior to heat treatment or subsequent welding operations. Radiography has been used, but sensitivity in detecting small flaws in circumferential weld seams is limited. Inspection procedures usually include verification of bellows material and heat-treatment certifications. Adequacy of welding procedures is assessed by examining samples cross-sectioned from a production run in which all units are fabricated under the same controlled conditions.

2.1.3.3 BELLOWS SPRING CHARACTERISTICS

The discussion in section 2.1.1.4.2 on temperature-related variations in mechanical spring forces and spring rates of helical springs is generally applicable to metal bellows. Section 2.1.2.3.2 on compensation for temperature effects also is applicable.

2.1.3.4 BELLOWS MATERIALS

The bellows material for a given application is selected for compatibility with the fluids to which it may be exposed, strength and fatigue life throughout the temperature range, and good forming and welding properties. For space-flight applications, acceptable materials are limited to the metal alloys with good corrosion resistance. Cold working during forming increases the strength of some materials such as the 300-series CRES alloys; but annealed stress-relieved bellows have more-nearly-linear mechanical spring rates, lower hysteresis, greater fatigue life, and greater thermal stability. The most commonly used materials for space-flight service are 321 and 347 CRES, 17-7PH, A286, AM355, Inconel X, Inconel 718, and beryllium copper. As noted previously, 17-7PH is subject to long-term stress corrosion and to hydrogen embrittlement. 321 and 347 CRES are the most commonly used bellows materials for applications within their stress limitations. Inconel 718 bellows are compatible with hydrogen and with most reactive fluids, including fluorine (refs. 79 and 80). As in the case of diaphragms, the use of Inconel 718 is increasing as a result of its good forming and welding properties, its strength and fatigue resistance, and its corrosion resistance.

2.1.4 Solenoid and Torque-Motor Actuators

Solenoid actuators are devices that contain a magnetic flux path, a movable armature with on-off full-stroke displacement, and one or more coils of wire for conducting electricity and exciting a magnetic field. These devices convert an electrical-energy input into a mechanical-motion output. Only direct-current solenoid actuators are used in space-flight hardware. Solenoid actuators appear most frequently in single-coil configurations with a linear-motion plunger armature or a flat-plate armature. Many special-purpose solenoid actuator configurations have been developed, including multiple windings in a single coil, multiple coils, and a combination of coils and permanent magnets; some types have had mechanical latching features. Reference 3 illustrates and describes a variety of typical configurations. References 82 through 93 present fundamentals of electromagnetic materials, circuits, and design procedures.

Torque motors, in which the interaction of magnetic fields is controlled to apply a magnetic torque to an armature with a limited angular displacement, were originally developed as small servovalve actuators. Larger versions of this actuator concept (ref. 94) have been developed for direct actuation of fast-response flow-control valves, including mechanically linked bipropellant valves. Although torque motors differ in mechanical configuration from conventional solenoid actuators, both have the same problems in coil design, coil enclosure, electrical connector, and materials. The same features that are critical in the successful design of solenoid actuators are, in general, also applicable to torque motors.

2.1.4.1 ELECTROMAGNETIC CIRCUITS

2.1.4.1.1 Materials

The materials for external surfaces of an electromagnetic actuator must be sufficiently corrosion resistant to satisfy space-flight hardware requirements for salt-spray qualification testing. Internal or external materials must be compatible with reactive fluids or vapors. Corrosion resistance in the presence of contaminants, including water or water vapor, is essential in satisfying the high-reliability requirements of space-flight service. Corrosion-resistant surface platings on ferromagnetic materials that are not inherently corrosion resistant have failed to provide protection when an accidental scratch penetrated the plating and resulted in loss of corrosion resistance. Nonuniform plating thickness at corners or other surface discontinuities has resulted in inadequate corrosion resistance or in mechanical interference between close-fitting parts.

Space-flight solenoid actuators usually are designed to use ferromagnetic materials such as 446, 430, or 430F CRES that do not require plating for corrosion resistance when exposed surfaces are machined for smooth finishes. Surface finishes of 16 μin. (0.406 μm) AA, or smoother, usually are specified for critical surfaces exposed to reactive fluids. Because the magnetic properties of materials are altered significantly by cold working, it is common practice to anneal all machined parts subsequent to rough machining and prior to finish machining, care being taken to ensure that the finishing removes very little material.

2.1.4.1.2 Eddy Currents

Induced eddy currents frequently introduce a time lag between a change in coil field strength and the resulting change in working airgap flux; this effect limits high-speed response. When ferromagnetic materials with high electrical resistivity are used to minimize induced eddy currents, the exposed surfaces must be plated for corrosion resistance. The use of machined slots in ferromagnetic materials in the flux circuit, analogous to the use of laminated materials, has proven to be an effective means of increasing the electrical resistance to eddy currents; the accompanying loss of magnetic permeability is minor. Locating the center of an electromagnetic coil as close to a working airgap as is structurally feasible minimizes the energize and deenergize time lags that result from induced eddy currents in the flux path between the coil and the airgap. Computation of eddy current effects in actuator magnetic circuit configurations is complex and inexact, and the present state of the art requires reliance on experience with similar configurations and on cut-and-try techniques. References 91 through 93, and the bibliographies therein, are typical of the literature on this topic. The literature provides insight into eddy-current problems, but practical solutions remain empirical when high-speed response is required.

2.1.4.1.3 Residual Magnetism

Solenoid actuators for space-flight service are required to operate with electrical signal circuits in which there may be a leakage current when the circuit is in its deenergized mode. This requirement evolves from the use of solid-state switching circuits with allowable leakage currents and from the use of test circuits in which not all ground connections are made to a common ground potential. Deenergized-mode leakage current in a solenoid actuator coil excites a residual electromagnetic field that supplements the residual ferromagnetic field of the magnetic circuit materials. When a signal circuit is deenergized subsequent to actuation of a solenoid actuator, the residual magnetic forces act in the direction of holding the actuator in its actuated position, and consequently oppose deactuation. In many applications, positive deactuation must be obtained without the assistance of actuator working-load forces acting in the deactuation direction. Such is the case in a normally closed solenoid valve that must close subsequent to each test or checkout prior to application of fluid pressure. In many electromagnetic actuators, residual magnetic forces are opposed by return-spring forces that also oppose actuation. Residual magnetic forces can be reduced significantly by increasing the size of one or more air gaps in the magnetic circuit, with a consequent reduction in actuation force. A nonmagnetic spacer installed in a working air gap does not affect actuation force but does reduce residual magnetic force at the expense of actuator stroke. Residual magnetism has retarded solenoid deactuation when presumably nonmagnetic materials used as shims or spacers in an air gap were actually ferromagnetic in behavior (e.g., rolled 302 CRES shim stock that was not annealed subsequent to rolling).

2.1.4.1.4 External Magnetic Fields

Design requirements limiting the magnetic field surrounding electromagnetic actuators have not been imposed in most rocket engine and vehicle applications. Such requirements, however, may become more common in future spacecraft programs. For example, for the Pioneer F and G spacecraft thruster, solenoid-valve specifications require an overall magnetic field strength of less than 50 gamma (50×10^{-9} teslas) while energized and less than 10 gamma (10×10^{-9} teslas) while deenergized, at a distance of 1 foot (30.48 cm) in all directions. Other component magnetic field strengths are limited to 3 gamma (3×10^{-9} teslas) at a 1-ft (30.48 cm) distance to allow proper operation of onboard magnetometer experiments.

Calculation of the permeance of leakage flux paths and mapping of the magnetic field surrounding an actuator require simplifying approximations for practical computations. Although not precise, analysis is used to predetermine the effects of voltage transients and the distribution of ferromagnetic materials on leakage flux. Correlations between experimental and theoretical information for existing configurations are made to establish confidence levels for new designs.

2.1.4.2 ELECTRICAL CIRCUITS

2.1.4.2.1 Voltage Surge Suppression

Insulation breakdown has occurred in electrical systems as a result of the voltage surge that can be generated by the sudden collapse of the magnetic field in a solenoid actuator when the solenoid is deenergized. It is common practice to design solenoids to withstand insulation tests at 1000 volts, because surges of this magnitude can be generated. When solenoids are designed to withstand the maximum anticipated voltage transient, deenergize surges are not a problem within a solenoid actuator as a system component; but these surges can be a problem elsewhere in the system, particularly in the switching elements.

As an alternate to designing all circuit elements to withstand the anticipated voltage surges, voltage surges can be suppressed in the electrical circuit external to a solenoid actuator, or they may be suppressed within a solenoid actuator assembly. Reference 95 discusses various circuit designs for voltage-surge suppression within a solenoid-actuator assembly. These include combinations of resistors, capacitors, and diodes shunted across a coil; the use of slugs, sleeves, and concentric coils; and the use of bifilar coil windings, i.e., two wires wound together to form two closely coupled coils in one subassembly, one coil for actuation and one coil short-circuited to suppress voltage transients.

Hermetically sealed coils for space-flight applications usually are baked to remove moisture and coil-entrained gases, cure the coil varnish, and anneal the insulation materials. Processing temperatures, and coil temperatures when a solenoid actuator is energized, limit the use of resistors, capacitors, and diodes as integral parts of a coil assembly. Slugs, sleeves, and concentric coils are not as closely coupled to the actuation coil as a bifilar coil, and consequently are not as effective. Some surge-suppression circuits with diodes require a polarity-sensitive coil assembly, with consequent hazard of installation error. All surge-suppression devices within an actuator assembly add to its size, and solenoid actuator size and weight frequently are primary design considerations.

Most significantly, any design approach for suppressing deenergize voltage transients that can in any way also suppress energize voltage transients complicates the problem of designing for fast and repeatable energize and deenergize response timing. When repeatability in response timing is required or is desirable in solenoid-actuated devices (e.g., two propellant valves that must actuate nearly synchronously, or pilot valves that should be replaceable without alteration of the timing of a control system), repeatability often is obtained by designing for fast response. With fast response, relatively large percentage variations in response timing result in small variations in absolute timing. The need for voltage-surge-suppression features may be in conflict with the response requirements, and solenoid design must be coordinated with electrical system design in accommodating both needs.

2.1.4.2.2 Driving Circuits

The most simple driving circuit for a solenoid actuator is a power supply and a manually operated mechanical switch, as frequently used in test or checkout setups. In space-flight control systems, switching is in response to control system signals, and the switching function usually is performed by circuits with solid-state elements. The driving circuits frequently are designed with elements for suppressing radio frequencies and for compensating or shaping transient response. Compensating features for protection of circuit elements usually include devices for suppressing deenergize voltage surge. Response-shaping networks occasionally are used for applying full supply voltage to a solenoid for actuation and, following a time delay, applying a lower voltage to minimize solenoid steady-state holding current.

When the design of a solenoid actuator and its driving circuit are closely coordinated, actuator speed of response and steady-state current drain characteristics can be optimized. Driving-circuit design for response shaping may be a necessity when both fast response and low steady-state current drain are required with large solenoid actuators. Problems in the design of solenoid actuators and driving-circuits have resulted from insufficient design coordination. The most common problem arises when the design specifications for a solenoid actuator state or imply step inputs in applied voltage, as approximated by a snap-action switch and a well-regulated power supply, when in fact the driving-circuit transient response is appreciably different as a result of its transient-suppression and response-shaping characteristics. (Although solenoid driving circuits may be physically incorporated in an actuator assembly, a discussion of driving-circuit detail design is beyond the scope of this monograph.)

2.1.4.3 COILS

2.1.4.3.1 Corrosion Protection

For space-flight service, solenoid-actuator coil spaces must be kept dry and chemically neutral, because coil voltages in the presence of dilute acids promote destructive electrolysis. The presence of contaminants and moisture can result in destructive arcing between electrical connector pins. For these reasons, isolation and hermetic sealing of coil spaces have become standard practices. In some designs, coil protection is achieved by using static seals, which permit disassembly, or by varnish-impregnation, encapsulation, or potting. The term "hermetic sealing," however, usually implies a welded seal, with the coil space carefully dried and purged with a dry inert gas prior to final sealing.

As noted previously, hermetically sealed coil windings usually are varnished and baked to remove moisture and anneal the insulation materials; this action also provides secondary protection against loss of hermetic sealing. The hermetic seal frequently has been lost when glass insulation bonded to the electrical connector pins and to the connector shell has been subjected to thermal transients during welding, testing, or operation. Corrosion has occurred in hermetically sealed coils as a result of inadequate drying or from introduction of moisture during purging prior to final sealing. Formal process control is essential in preventing such failures.

2.1.4.3.2 Coil-Wire Insulation

2.1.4.3.2.1 Temperature Limitations

Both government (ref. 90) and industry standards have been established to specify a grade of insulated wire known as "magnet wire." This wire is fabricated in American Wire Gage (AWG) sizes specifically for use in electrical coils, and its quality is rigorously controlled. Insulated magnet wire is classified according to its maximum rated operating temperature, the type of insulation material, and the thickness of the insulation. The maximum hot-spot temperature in a coil is the determining factor in selecting magnet-wire insulation with a suitable temperature rating.

Various types of plastic-film insulation are in increasingly common use, and for space-flight hardware the plastic insulations are used almost exclusively. These plastics are available with temperature ratings of 105°, 130°, 155°, 180°, and 200°C (378, 403, 428, 453, and 473 K). Standards have not yet been established for magnet-wire insulations suitable for operation at temperatures exceeding 200°C (473 K); various commercial insulations are available, however, for operation in this temperature range.

Ceramic-film insulations overlaid on nickel-coated copper are usable at temperatures above 500°C (773 K). At these temperatures, a migration of conductor material into the insulation results in a continuous increase in the wire resistance and a decrease in the insulation dielectric strength; the rate of migration increases with temperature. The number of hours at which insulated wires of these types can be exposed to a given temperature usually is limited to the time required for a 10-percent increase in wire resistance.

Insulation in the form of sheets or tape to separate coil layers from each other or to separate coils from surrounding materials is made from a greater variety of materials than is available for wire insulation. Most insulation materials have trade name designations, and manufacturers' data should be consulted for complete listings and specific application information.

2.1.4.3.2.2 Electrical Resistance and Dielectric Strength

Several factors are considered in selecting insulation thicknesses. Dielectric strength related to insulation thickness must be adequate to withstand the maximum transient voltage that will be encountered. Surges approaching 1000 volts frequently occur when a coil is deenergized, either in service or during tests. Insulation thicknesses are limited by coil size limitations. The choice of wire insulation thickness for any wire size affects the number of coil turns in a unit cross section of a coil and, therefore, affects coil size and the ratio of coil turns to coil resistance (ref. 85).

Two separate tests of coil insulation are in general use for inspection purposes. One test measures the resistance to direct current between the coil conductor and the housing enclosing the coil; the other measures the dielectric strength of the coil insulation. The direct-current test is an accurate quantitative test of limited scope; the dielectric-strength test serves as a qualitative test to detect defective or marginal coils.

The direct-current resistance usually is measured with a "megger" that applies a differential voltage (usually 500 or 600 volts dc) to the coil conductor and its housing and measures the leakage current through the insulation. A microammeter in the megger is calibrated in units of megohms resistance. The required insulation resistance to direct current from conductor to housing will vary with specific applications but is generally 100 megohms or greater. In well-designed and well-constructed coils, resistances of several thousand megohms are achieved easily. For a coil with negligible capacitance and dielectric absorption, when 100 megohms has been specified as the minimum acceptable direct-current resistance for the coil insulation, the corresponding maximum leakage current at 1000 volts dc is approximately 10 microamperes. A reasonable allowable current flow during a 1000 volts ac test would be 200 to 500 microamperes; this current level allows for equipment limitations, capacitance, and dielectric absorption.

A dielectric-strength test is performed by measuring alternating-current leakage through the insulation. This test is performed at a higher electric potential than the megger test and usually is referred to as a "hi-pot" test. As a general practice, 1000 volts ac plus twice the rated voltage is applied as a test voltage. This practice, however, is adjusted to suit the needs of specific component applications. For 28-volt dc solenoid coils, a test voltage of 1000 (+100,-0) volts ac usually is specified. Torque motors in circuits that do not experience high voltage surges frequently are tested at 600 volts ac or less.

Dielectric-strength test equipment usually is designed to operate at 60 Hz with an adjustable output voltage. Defective insulation or too-close spacing of conducting materials will result in an abrupt increase in alternating-current flow as the test potential applied to a coil is increased. The test equipment is designed to limit the maximum current flow in the event of an insulation breakdown. In addition to measuring very small insulation-leakage currents, the test equipment also measures current flow resulting from capacitance and dielectric absorption.

Dielectric-strength tests tend to degrade insulation properties, and coils have been damaged by repetitive tests. It is now common practice to limit these tests in time duration and in number. Limiting the number of tests requires careful review and monitoring of all checkouts and tests to which a coil assembly is exposed throughout its fabrication, acceptance testing, and service life.

Although it is convenient for applying high voltages and for detecting insulation flaws not detectable with a megger test, the dielectric-strength test does not provide for accurate measurement of direct-current resistance as an isolated parameter. A megger test and a dielectric-strength test serve two different purposes; neither is adequate in itself, but in properly related combination they provide for an adequate inspection of coil insulation qualities.

2.1.4.3.3 Temperature-Effect Compensation

Although insulated copper magnet wire manufactured to government specifications (ref. 90) is the most commonly used current conductor for solenoid and torque motor coils, other metals and alloys are available in magnet quality wire form and frequently are useful in coil designs for space-flight hardware. When a coil must operate over a wide range of temperatures, the use of a wire material with a temperature coefficient of resistance much lower than that of coppper is used to limit variations in coil resistance, and therefore in coil current, resulting from temperature variations. Low temperature coefficients, however, are accompanied by high resistivities, as shown in table II.

A coil that satisfies the requirements for ampere-turns and for minimum current variation with coil temperature can be obtained with relatively few coil turns of a high-resistance, low-temperature-coefficient wire (e.g., constantan) in series with a relatively large number of coil turns of copper wire. With most of the coil resistance in a length of wire that has negligible change of resistance with temperature, a change in overall coil resistance results only from change in the low-resistance copper that provides the required number of coil

TABLE II. – Resistivity and Temperature Coefficient of Resistance for Typical Coil Wires

Material	Composition	Resistivity at 20°C (293 K) ohms/cir. mil. ft	Resistivity at 20°C (293 K) microhm-cm	Temperature Coefficient α, K^{-1}
Silver	Ag	9.79	1.63	0.0038
Copper	Cu	10.37	1.73	0.0039
Aluminum	Al	16.10	2.68	0.0044
Radio alloy 30	2 Ni, 98 Cu	30	5	0.0013
Radio alloy 60	6 Ni, 94 Cu	60	10	0.0005
Radio alloy 90	12 Ni, 88 Cu	90	15	0.0004
Radio alloy 180	22 Ni, 78 Cu	180	30	0.00018
Constantan	45 Ni, 55 Cu	294	49	0.00004

Notes: (1) Resistance at 293 K = R_{293} = resistivity $\times \frac{\text{wire length}}{(\text{wire diam.})^2}$

(2) R_T $(T \neq 293\ K) = R_{293}\ [1 + \alpha(T - 293)]$

turns. For example, consider a coil wound with 15 turns of constantan wire with a resistance of 25 ohms in series with 985 turns of copper wire with a resistance of 5 ohms. The coil has totals of 1000 turns and 30 ohms resistance. Because the constantan has a negligible temperature coefficient of resistance, only 1/6 of the total coil resistance is subject to significant variation with coil temperature, and coil current variations with temperature will be only 1/6 as much as would occur in a 1000-turn, 30-ohm coil wound entirely with copper wire of appropriate size and resistance.

Concentrating the coil resistance in relatively few coil turns also concentrates the conversion of electrical energy to heat energy, and the coil must be designed to accommodate the resulting hot spot. When a coil must operate at temperatures that promote rapid oxidation of copper, nickel-clad copper wire frequently is used. For temperatures approaching the melting point of copper, other metals or alloys must be used.

2.1.4.3.4 Coil-Wire Extensions

Extension wires between coils and their electrical connectors have failed because of vibration-induced flexing of the extension wires. These fatigue failures usually have occurred at the soldered attachments to electrical connector pins. Failures also have resulted from mechanical damage to coil-wire ends or to extension-wire ends when the insulation film was stripped to expose the wire ends for soldering or welding; nicks, scratches, or thinning of a wire increase the susceptibility to failure from vibration and to failure from localized excessive current density and overheating. When coil wires are small, it is common practice to attach heavier extension wires to the coil wires and mechanically restrain the attachment joints. The extension wires often must be flexible so that they can be nested easily, subsequent to attachment to electrical connector pins, as the connector is maneuvered into position for welding the connector to a coil housing. A combination of flexibility and mechanical strength in coil extension wires is obtained by using multistrand wires. Plastic-film insulation permits use of thermal techniques for stripping insulation at wire ends to preclude potential damage resulting from mechanical stripping.

2.1.4.4 ELECTRICAL CONNECTORS

2.1.4.4.1 Construction and Materials

Electrical connectors for hermetically sealed coil enclosures include a connector configuration that complies with electrical-system requirements for attachment of a mating connector and an adapter section that accommodates a weld joint, as illustrated in figure 10.

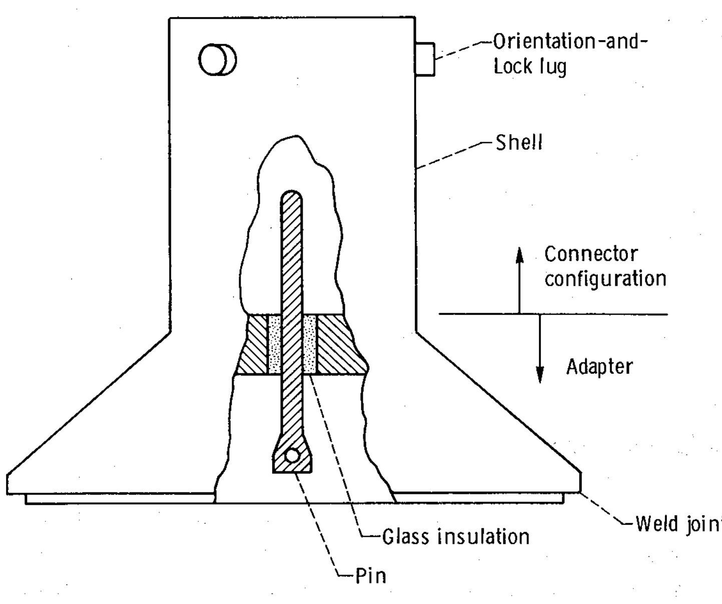


Figure 10. - Sketch of electrical connector for hermetically sealed coil enclosure.

The geometry of the connector configuration usually is established by system specifications that standardize the type of connector to be used throughout the system. The design of adapter sections for joining a connector to an actuator housing is an integral part of the actuator design activity.

Damage to connector-pin insulation or to actuator-coil insulation occasionally has resulted from excessive temperatures or temperature gradients during welding operations. For the connector shown in figure 10, these thermal problems are minimized by machining the connector shell from a single piece of bar stock and by locating the weld joint as far away from the pin insulation as practical size limitations permit. Connector temperature rise during welding is minimized by attaching a metal mass to the connector to act as a heat sink. Carefully controlled, automatic-welding procedures are used to limit heat input.

Short circuits have occurred and pins have corroded in electrical connectors exposed to external moisture or reactive propellants. Pin alloys with the lowest electrical resistance require plating for corrosion resistance. Pin plating has been an unsuccessful corrosion preventive, because the plating is discontinuous at the intersection with the bonded-glass pin insulators. Nonplated 300-series-CRES pins are now in common use.

The most commonly used connector shell materials are the 321, 347, 304L, and 316L CRES alloys. The same materials are used for corrosion-resistant pins.

Electrical circuits have been connected incorrectly when identical connectors were used for various components in a system. To preclude cross-connection errors, connectors of different sizes and connectors with different numbers of pins are employed in circuit locations where the possibility for these errors exists.

Redundant connector pins have been used to provide parallel paths through conduit and actuator mating connectors for greater assurance that reliable contact will be maintained under operating conditions.

Electrical harness or cable connectors have loosened or detached under vibration and have opened electrical circuits. This problem has been eliminated by lockwiring connectors to actuator housings.

2.1.4.4.2 Connector Breakdown

Electrical breakdown between connector pins or between pins and a connector metal shell has occurred during tests of coil insulation resistance and dielectric strength. When test specifications are adequate for identifying solenoid-actuator assemblies with marginal or deficient insulation resistance or dielectric strength, inadequate units are rejected during tests prior to acceptance for flight service. In some instances, breakdown has resulted from contaminated or improperly cleaned electrical connector insulation surfaces. Arcing has resulted from improper drying, purging, and gas filling of hermetically sealed coil spaces.

Hermetically sealed solenoid coil enclosures usually are designed with one or more small ports so that each enclosure can be pressurized with helium gas and the seal welds then tested with helium leak-detecting equipment. An acceptable enclosure is then purged with dry nitrogen gas, and the ports are sealed to enclose a nitrogen-gas fill at atmospheric pressure and nominal room temperature. Because the dielectric strength of helium is significantly lower than the dielectric strength of nitrogen under these conditions, connector-coil assemblies that are acceptable when thoroughly purged with dry nitrogen may be susceptible to arcing when substantial helium is present.

Table III lists the relative dielectric strengths of common gases at standard sea-level pressure and temperature (760 mm Hg, 288 K). In uniform electric fields, breakdown occurs at a

TABLE III. – Relative Dielectric Strengths of Common Gases at Standard Sea-Level Pressure and Temperature

Gas	Relative dielectric strength
Nitrogen	1.00
Air	0.95
Oxygen	0.90
Hydrogen	0.57
Argon	0.28
Neon	0.13
Helium	0.14

voltage that is a function of the gas pressure and the electrode spacing, as shown in figure 11 (Paschen's curves) for several gases at 20°C (293 K) (refs. 88 and 89). Dielectric strength varies with gas temperature only as the latter affects gas density, and the data in figure 11 can be adjusted accordingly.

As indicated in figure 11, for a given electrode spacing, breakdown voltage decreases with decreasing pressure to a minimum value. Further reduction in pressure results in an increase in breakdown voltage; thus, hermetically sealed enclosures containing a hard vacuum have been used to obtain high internal dielectric strength. The disadvantage of an enclosed vacuum is that it is susceptible to undetected leakage from the external environment.

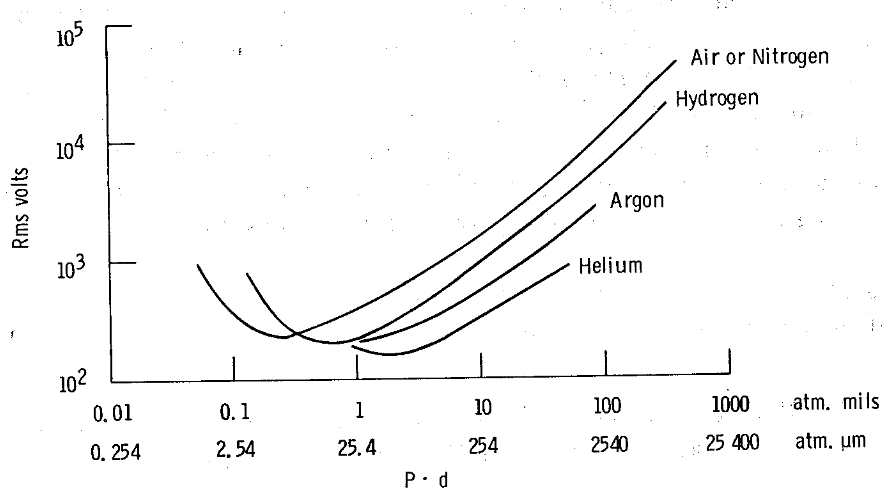


Figure 11. - Breakdown voltages of selected gases at 20° C (293 K) as a function of electrode spacing d and pressure P (refs. 88 and 89).

The Apollo Lunar Module Descent Engine (LMDE) employs a hermetically sealed torque-motor actuator for throttle-valve linkage. The sealed enclosure is filled with a mixture of 10-percent helium, 90-percent nitrogen at 0.25 ± 0.1 psia (1.72 ± 0.70 kN/m^2; the helium is employed as a leak-test tracer. The system does not experience high voltage surges, and internal dielectric strength is adequate. At the charge pressure, a snapover-disk pressure indicator in the assembly is concave with sea-level ambient pressure. If a housing leak develops and internal pressure increases to approximately 12 psia (82.7 kN/m^2), the indicator disk snaps into the convex position.

Other materials and methods have proven effective in providing high pin-pin or pin-case dielectric strength and insulation resistance when conditions permit. When the physical spacing between adjacent pins or between pins and the connector metal shell is small, as in miniature connectors, potting compounds have been used to fill the spaces between pins and the connector shell. Increased resistance to electrical breakdown thereby is obtained with a technique that also provides mechanical support of lead-wire attachments. External mating connectors frequently are designed with compressible-elastomer insulating seals that provide an increased dielectric path between adjacent pins and between the pins and the shell. External mating connectors are visually inspected and blown clean with filtered dry nitrogen gas immediately prior to making an attachment.

2.1.5 Snap-Action Actuators

Snap-action actuators have the characteristic of nonlinear response to an input signal. When a gradually changing input signal reaches a value at which actuation is initiated, the actuator responds by displacing rapidly to its full-stroke position. In hydraulic and pneumatic actuators, nonlinear response can be obtained when initiation of motion, in response to an input signal pressure applied to a primary pressure-sensing effective area, exposes a secondary pressure-sensing area to fluid pressure to produce an unbalanced opening force. In electromagnetic actuators of the plunger-solenoid type, plunger linear motion decreases the axial length of the working air gap, thereby increasing the air-gap permeance; and the applied magnetic actuation force increases exponentially as a function of plunger displacement once sufficient ampere-turns have been applied to initiate plunger motion.

In the preceding examples, snap action is obtained by applying actuation forces as a function of actuator displacement to supplement an applied signal force once motion commences. Alternatively, snap action can be obtained when the forces opposing actuation decrease as a function of actuator displacement when a signal force sufficient to initiate motion is applied and maintained. This type of snap-action actuator uses mechanical springs with negative spring rates.

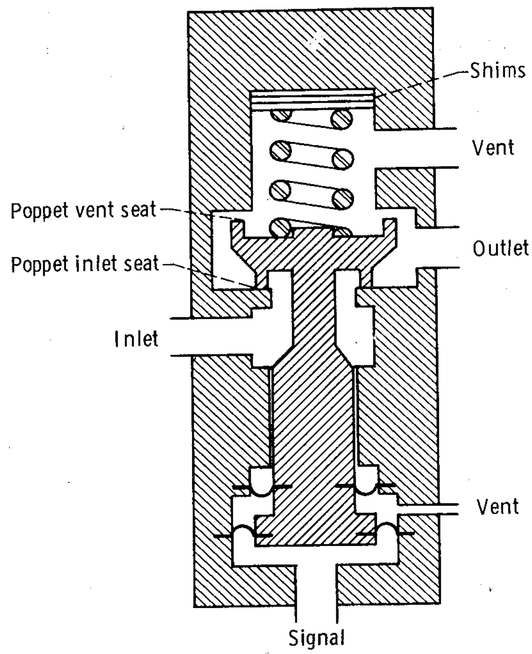
Snap-action concepts are most frequently employed in actuators for on-off pilot valves and switching devices when fast response and positive actuation without chatter are required.

2.1.5.1 REGENERATIVE-FORCE ACTUATORS

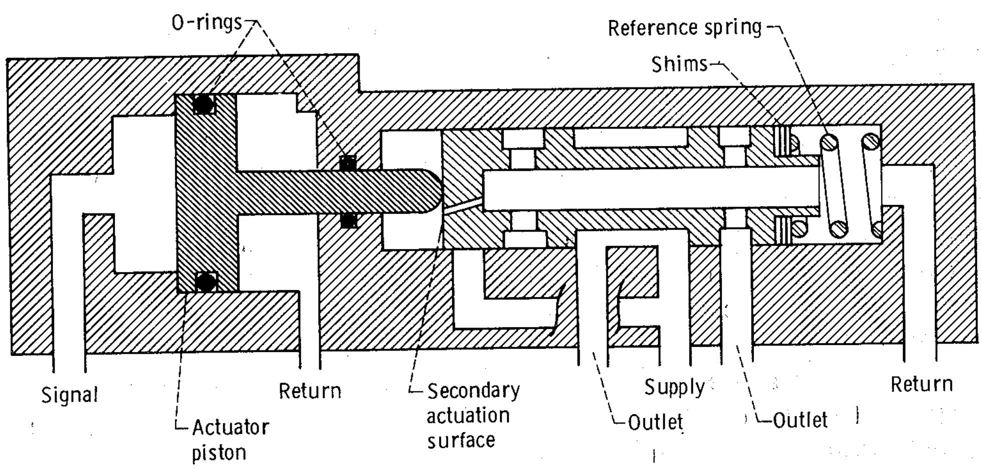
A variety of pneumatic and hydraulic snap-action valve actuators have been developed to utilize the valve effluent pressure for applying regenerative actuation forces as functions of valve displacement following initiation of valve motion by a signal pressure. Figure 12 illustrates two design concepts for regenerative-force actuators.

In the design in figure 12(a), pneumatic inlet pressure forces applied to the stem of a normally closed three-way valve are balanced when the valve is closed, because the diaphragm exposed to inlet pressure has an effective area nominally equal to the cross-sectional seating area of the inlet poppet. The signal-pressure level at which the poppet starts to move from its normal position is controlled by a shim-adjustable helical-spring reference force. When an increasing signal pressure is sufficient to unseat the inlet poppet, pneumatic flow from inlet to outlet and from outlet to vent results in a pressure increase acting as an unbalanced actuating force against the cross-sectional area of the poppet vent seat. As the poppet displaces, the inlet flow area increases, the vent flow area decreases, the unbalanced actuation pressure force increases exponentially, and snap-action results. It is essential that the increase in unbalanced actuation force resulting from inlet flow as a function of poppet displacement be greater than the increase in the opposing reference-spring force resulting from mechanical spring rate. In this design, the signal-pressure setting is a specified differential pressure above the vent-cavity pressure.

Figure 12(b) illustrates a design concept for a snap-action, hydraulic, four-way spool valve. Linear motion from the normal position commences when a signal-pressure force acting



(a) Normally closed three-way pnuematic valve with snap-action opening as a function of relative seat areas.



(b) Snap-action hydraulic four-way spool valve.

Figure 12. - Schematics of snap-action actuators utilizing regenerative forces.

against an actuator piston exceeds the opposing helical-spring reference force. A secondary actuation surface, one end of the spool valve at signal pressure, is ported to system return pressure when the valve is in its normal position. A small linear displacement of the spool valve uncovers a port in the spool sleeve and admits flow from supply pressure to the cavity at signal pressure. The resulting unbalanced actuation force produces snap-action, full-stroke displacement. In this design, the signal pressure setting is a specified differential pressure above system return pressure.

An inherent problem exists in all pressure-sensing snap-action devices when the device must operate at sensed-pressure levels near the actuation-pressure setting. Acceleration force inputs associated with vibration can induce full-stroke on-off cycling when the bandwidth between the pressure levels required for actuation and for deactuation is small. Interaction between sensed pressure and actuation of a snap-action device, as affected by acceleration forces and bandwidth, requires system design analysis beyond the scope of component design studies. A snap-action device that performs well in one system may exhibit on-off cycling in a different system. Time lags in dynamic response in pressure-sensing lines and passages require careful evaluation. Close coordination of system design and snap-action component design is essential in achieving successful performance.

2.1.5.2 NEGATIVE-RATE-SPRING ACTUATORS

Snap-action design concepts using negative-rate Belleville springs have been developed for on-off devices such as pressure switches, position-indicating switches, relief valves, pilot valves, and on-off pressure regulators. Figure 13 is a schematic of a simple on-off, or "bang-bang" pressure regulator that illustrates the basic actuator design factors.

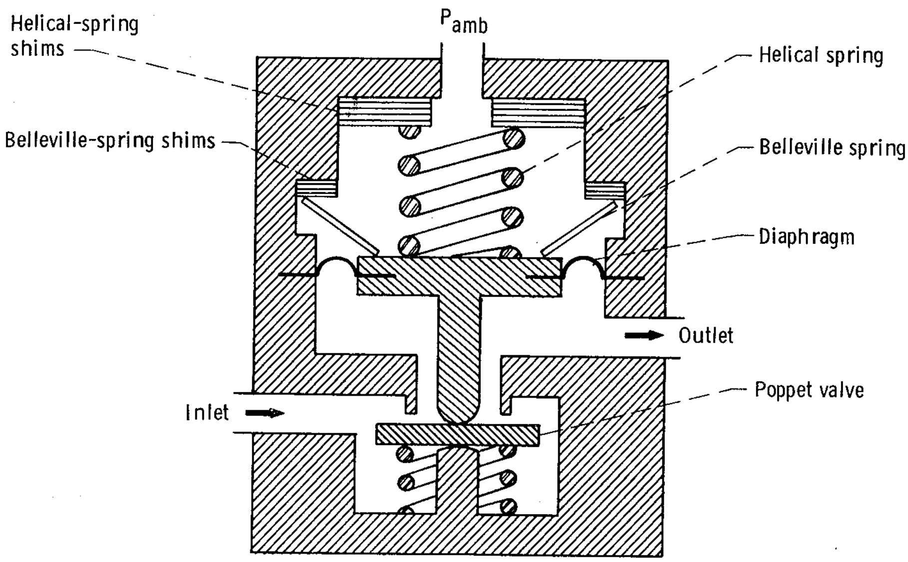


Figure 13. - Schematic of snap-action actuator with negative mechanical spring rate.

The diaphragm actuator in figure 13 is biased in the valve-open direction by a helical spring and by a Belleville spring designed and installed to operate in its negative-rate range of deflection. The negative spring rate of the Belleville spring is numerically greater than the sum of the positive spring rates of the helical spring and the diaphragm, so that the actuator net spring rate is negative. When the sum of the inlet and outlet pressure forces applied to the actuator exceeds the net installed spring force, as the actuator displaces from its normal position and the actuator net spring force decreases with displacement, the pressure forces that initiated displacement are greater than the opposing spring force, and the valve snaps closed. The valve will remain closed until the outlet pressure decays sufficiently to initiate displacement in the valve-opening direction. As the valve begins to open and the actuator net spring force increases with displacement, the net spring force exceeds the pressure force, and the valve snaps open.

Figure 14 illustrates the Belleville-spring force-vs-deflection characteristics. The mechanical spring rate, the slope of the curve, is a function of the spring deflection from free height. Point A in figure 14 is a typical point on the force-deflection curve in its negative-spring-rate region. At point A, a decrease in spring force ΔF accompanies an increase in spring deflection ΔX. If the spring is installed with a deflection from its free height that is greater than the deflection at point A, the slope of the curve is steeper and the spring has a greater negative spring rate.

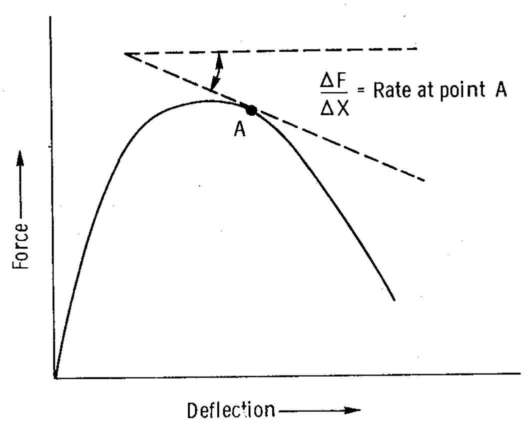


Figure 14. - Belleville-spring force-vs-deflection characteristics.

In the regulator shown in figure 13, the shims for adjustment of the Belleville-spring installed height provide for adjustment of the Belleville-spring rate to compensate for fabrication tolerances affecting the spring rates of the helical spring, the diaphragm, and the Belleville spring, and thus provide for adjustment of the actuator combined mechanical spring rate. When the actuator net spring rate has been adjusted, the net installed spring force is adjusted by changing the helical-spring shim thickness. Net installed spring force controls the outlet pressure setting, and net spring rate controls the bandwidth in regulated outlet pressure. References 3, 47, and 48 present basic design procedures and design data for Belleville springs. References 64 through 68, 96, and 97 discuss problems in the application of negative-rate Belleville springs in snap-action actuators and present problem solutions that have been developed.

Differences between computed and actual negative-rate spring characteristics frequently require iterative design and development effort when precision is required. Obtaining repeatable spring characteristics in production Belleville springs frequently entails a high rejection rate. Static friction and sliding friction at spring inner- and outer-edge support surfaces introduce significant hysteresis, and thereby limit repeatability. References 64, 65, and 96 review techniques for minimizing friction by the use of spacer rings at the inner and outer edges. In the actuator of figure 13, friction is maintained within acceptable limits by close control of spring edge radii and by the use of hardened spring supports with smooth surface finishes.

2.1.6 Rotary Actuators

Hydraulic, pneumatic, and electric rotary actuators offer the advantage of compactness when a rotary output motion is required and the actuator can be directly coupled to its loading device. A rotary actuator with gearing may be more compact than an equivalent linear actuator with a linear-to-rotary motion transmission. Rotary actuators with rotary-to-linear motion drive mechanisms may offer size advantages when long-stroke linear-motion output is required.

2.1.6.1 HYDRAULIC AND PNEUMATIC ACTUATORS

The primary disadvantages of hydraulic and pneumatic rotary actuators are associated with rotor-to-stator dynamic seals and with actuation-fluid leakage. Relatively high leakage flowrates and variations in leakage flowrates resulting from clearance changes as functions of pressure, temperature, and angular displacement introduce problems in control of actuator response and efficient use of operating fluids. In on-off applications with response time controlled by orifices or other flow restrictions, the response time varies with changes in actuator leakage. In modulating applications with actuator response controlled by a servovalve, the actuation system response-time constants and system stiffness are both affected by variations in leakage. The on-off rotary actuator thus is limited to applications in which timing control is not critical; the modulating rotary actuator may require a closed-loop system design that compensates for variations in leakage. References 98 through 100 review the current state of the art in the analysis and design of hydraulic and pneumatic rotary actuators.

Hydraulic motor assemblies, with or without speed-control features, are used extensively in aircraft systems and offer potential for use in rocket engine and spacecraft hydraulic control systems in which motor fluid can be recovered and recirculated. As discussed in reference 98, pneumatic motors have high gas consumption rates, compared with piston actuators, and offer little potential for use in space-flight stored-gas systems. The primary advantages of all-metal pneumatic motors are tolerance for environmental extremes and operational capability with hot gases.

2.1.6.2 ELECTRIC-MOTOR ACTUATORS

Electric motors are used in rocket engine control systems principally as valve actuators and as drivers for hydraulic pumps. Electric motors are protected against exposure to environmental conditions by isolating the motor assemblies in sealed enclosures with lip seals as output shaft dynamic seals.

Motor assemblies used as valve actuators incorporate speed-reducing gears; tachometers; position-limiting mechanical stops; and position-indicating switches, transducers, or potentiometers in a common sealed enclosure. Units are purged and prefilled with dry inert gas or are cooled by a recirculating dry inert gas so that a passive internal environment is obtained. On-off position control usually is obtained with conventional direct-current motors, whereas modulating position control usually is obtained with two-phase alternating-current motors, direct-current torque motors, or direct-current stepping motors. A variety of special motor designs is available for compliance with power supply, speed, torque, and special servosystem requirements. References 5 and 101 through 105 review electric motor designs and characteristics for control system applications. References 10 and 11 describe hydraulic-pump drive motors used in the Saturn II and Saturn S-IVB vehicles.

The intermittent and short-duration duty schedules for space-flight valve actuators and their low power requirements permit the use of dry-film lubricants for motor shaft bearings and for gears. Pump drive motors designed for many hours of operation at significant power levels (4.5 hp (3.36 kW) for Saturn) require the use of grease-packed bearings.

Thorough analysis of system requirements and motor loading under transient as well as steady-state conditions is the basis for defining the specific motor operational characteristics that will provide adequate margin for reliable operation. Thorough analysis of heat-energy inputs resulting from motor inefficiency and heat transfer to or from the environment and the motor attachments is required in maintaining motor temperature distributions within allowable limits. Acceleration and vibration inputs introduce structural and operational problems associated with cyclic relative motion at shaft bearings, gear-teeth contact surfaces, instrumentation contacts, and commutator brush contacts.

The discussions in section 2.1.4 on solenoid and torque-motor actuators are generally applicable to electric motors, allowing for differences in configuration.

2.1.6.2.1 Redundant-Electric-Motor Actuators

Rotary-electric-motor actuator assemblies with screw-jack linear output motion have been used as rocket engine valve positioners. Two or more motor rotors coupled to a common jackscrew provide for motor and motor-circuit redundancy.

The actuator for the LMDE throttle valve is controlled by three electronic channels that power three direct-current pancake motors mounted on a common shaft. The motor shaft powers a ball-screw rotary-to-linear-motion transmission assembly. Five potentiometers are geared to the motor shaft. Three of these potentiometers provide position-feedback information to the three motor electronic channels, one for each channel; the other two potentiometers provide actuator-position information for telemetry. This actuator configuration permits circuit design so that actuator performance will remain within specified limits in the event of a failure of any electrical system element.

Motors are designed to permit assembly prior to removal of magnetic keepers, and nonmagnetic housings are used to preserve permanent-magnet field strength. Spacing between motors is sufficient to prevent the magnets of one motor from affecting performance of an adjacent motor. Output-shaft displacement is limited by nonjamming mechanical stops.

Redundant-electric-motor actuators are reviewed in detail in references 106 through 108.

2.1.7 Advanced Concepts

There are currently available advanced-state-of-the-art actuators with potential applications in space-flight control systems. These advanced concepts include nutating actuators, digital actuators, and fluidic-logic actuators.

Nutating actuators. – Pneumatic actuators that convert the motion of a nutating disk into a rotary output motion have been developed as all-metal reversible-direction-of-rotation motors for high-temperature and nuclear-radiation environments. Primary problems with this design have been high leakage and scrubbing of the nutating disk against stationary parts (ref. 109).

Digital actuators. – Digital actuators are positioning devices that respond to input signal pulses or steps in producing discrete changes in controlled displacement. The electric stepping motor, with multiple windings that are energized sequentially for discrete increments in clockwise or counterclockwise rotation, is the most highly developed actuator of the digital category. Design concepts have been developed in which a stepping motor is used in an operator for a hydraulic or pneumatic actuator, thereby adapting a high-force-level analog actuator for operation with digital input signals. Multiple-piston, hydraulic, linear-displacement actuation devices have been designed to control output shaft position incrementally in response to the application or removal of sequenced input pressure signals (refs. 110 through 115).

Fluidic-Logic actuators. – Actuators in fluidic-logic systems are conventional hydraulic and pneumatic actuators with fluidic-logic-element operators. References 116 through 125 provide a survey of the published literature.

2.2 OPERATOR CONFIGURATIONS

Operators are classified as on-off operators for two-position control or as modulating operators for variable-position control. Operators reviewed in this monograph are pilot valves, servovalves, pressure dividers, and advanced-concept operators such as stepping motors and fluidic-logic devices. Switches used as actuator position indicators that also may be used in operator circuits are reviewed in section 2.5. Actuator and electrical-operator assemblies have been designed with the operator driving circuitry included in the assembly. The design of driving circuits, however, is beyond the scope of this monograph. Operator valve design information in this monograph supplements more detailed information in references 126 and 127.

2.2.1 Pilot Valves

A pilot valve is defined as an on-off operator that responds to a signal input for two-position, open-loop control of actuator fluid flow or fluid pressure. The most common types of pilot valves are solenoid valves, pressure-actuated valves, and position-actuated valves in two-way, three-way, and four-way configurations. Pilot valves provide for on-off control of large hydraulic or pneumatic actuation forces in response to relatively low power signals. They are used where full-stroke actuation and deactuation is required, as in the case of pilot-operated hydraulically or pneumatically actuated shutoff valves. Primary areas of concern in the design of a pilot valve are effective flow area, positive actuation, satisfactory response to pressure transients, control of stagnant-fluid temperature, and control of contamination.

2.2.1.1 EFFECTIVE FLOW AREA

The effective area for actuation-fluid flow through a pilot valve usually is sized to preclude excessive pressure drop through the pilot valve and its porting under the flow conditions required for actuator displacement transients. Control of on-off actuator response timing is obtained most easily when the primary flow resistance in a valve-and-actuator circuit is concentrated at a timing-control orifice. The orifice can then be used to compensate for differences in system flow resistance that occur in different assemblies of nominally identical systems. As discussed in section 2.1.1.2.1, an expensive, time-consuming, cut-and-try orifice-replacement approach to obtaining the required response timing for an actuator can be eliminated by the use of precision orifices. The cut-and-try approach cannot be avoided, however, if fabrication and assembly tolerances for pilot valves and plumbing introduce wide variations in system flow resistances among different assemblies. References 128 through 133 review procedures and data for sizing pilot valves.

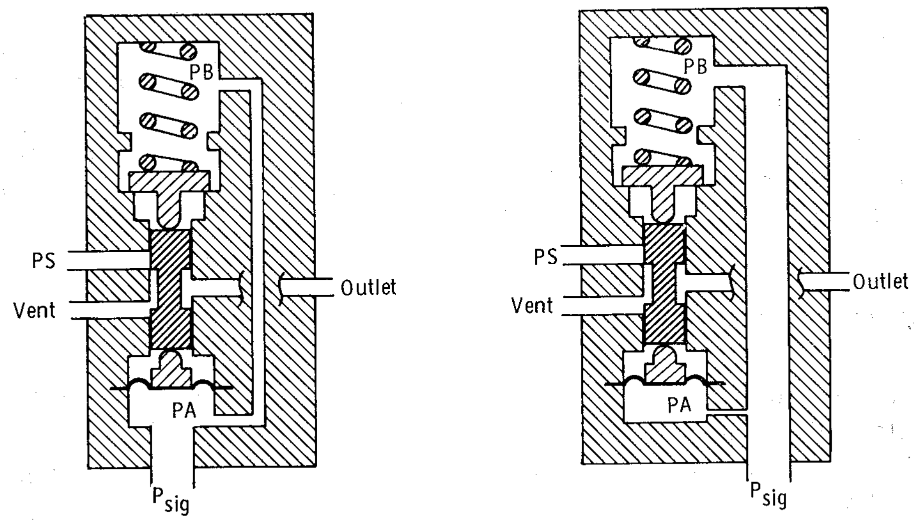
2.2.1.2 POSITIVE ACTUATION

Erratic operation and dynamic instability of pressure-actuated pilot valves have resulted from fluid-flow dynamic forces acting on the valves during actuation or deactuation transients. Similar problems have occurred when the sensed pressure is also the source for valve flow and when signal-pressure drops accompanying flow result in interaction between the sensed pressure and valve actuation. Section 2.1.5 illustrates and discusses snap-action actuators that ensure positive full-stroke actuation of pilot valves once actuation has been initiated by a sensed signal pressure. References 3 and 17, and published documents cited therein, present design-analysis procedures for predicting fluid-flow dynamic forces that may contribute to valve chatter or dynamic instability, and describe design techniques for minimizing or eliminating such forces.

2.2.1.3 RESPONSE TO PRESSURE TRANSIENTS

Pressure-actuated pilot valves have failed to perform within actuation-pressure limits when time lags in pressure sensing resulted in the valve responding both to rate of change in sensed pressure and to the level of sensed pressure. Thus, a pilot valve that had been set to actuate within a specified pressure range when sensed pressure was changed gradually during test has actuated at a significantly different sensed-pressure level in operation when the sensed pressure changed rapidly. Ideally, there should be no time lag in the response of pilot-valve motive forces to sensed-pressure transients. In many pilot-valve applications, however, fluid-compressibility effects and the presence of flow restrictions between the sensed-pressure location and the actuation pressure within a pilot-valve assembly introduce response time lags and consequent sensitivity to rate of change in sensed pressure. In the case of a pilot valve situated remotely from the sensed-pressure location, time lags in signal-pressure transmission can result in transient differences between sensed pressure and valve actuator pressure. In hydraulic systems, fluid-compressibility effects resulting from entrapped gases can introduce time lags in pressure response and problems in response to rate of change in sensed pressure similar to those encountered in pneumatic systems.

Figure 15 shows two designs for porting arrangements that provide a pilot valve with the desired transient response. Figure 15(a) illustrates a pneumatic pilot valve configuration in



(a) Different transient responses in primary and secondary actuation.

(b) Identical transient responses in primary and secondary actuation.

Figure 15. - Schematics of pilot valves with porting arranged to provide desired transient response.

which primary and secondary actuation pressures PA and PB act against a diaphragm in the direction of valve actuation and also against the valve cross section to oppose actuation. The spring-cavity volume and the restriction of the flow passage for that volume result in significant lag in the dynamic response of pressure PB to transients in pressure PA. If the actuation pressure setting is adjusted with PA changing gradually, and with PB nearly equal to PA, the valve will actuate at a significantly lower applied pressure when PA is changing rapidly and the transient in PB is lagging the transient in PA. The sensitivity to rate of change in PA can be reduced by minimizing the volume associated with PB and by enlarging the flow passage for that volume.

In figure 15(b), the volume and the flow restriction associated with primary actuation pressure PA are designed to have the same time constant as the volume and flow restriction associated with secondary actuation pressure PB. As a result, pressures PA and PB have identical transient response to externally applied pressure P_{sig}. The extent to which pressures PA and PB both lag fast transients in the externally applied pressure, and the consequent sensitivity to rate of change in applied pressure, is a function of the size of the pressurized volumes and flow passages.

2.2.1.4 STAGNANT-FLUID TEMPERATURE

Stagnant fluids in flow passages of tight-shutoff pilot valves or in pressure-sensing cavities of pressure-actuated pilot valves are susceptible to thermal-environment effects. Variations in liquid viscosity or in gas temperature, related to variations in localized environmental temperatures, can affect actuation system response. Freezing of liquids in low-temperature environments can result in response failures. Elevated environmental temperatures and heat soakback during or subsequent to an engine firing can result in liquid vaporization and in overheating of nonflowing pilot valves. Solutions to these problems have included relocation of valves, the use of thermal insulation, and the design of flow paths for heat energy with thermal barriers to minimize heat transfer. In hydraulic systems, pilot valves have been designed for controlled leakage or for a small continuous circulation flow in order to control temperature. Pilot-valve designs frequently are incorporated in the design of actuator assemblies or other components that can provide shielding or temperature control for stagnant pilot-valve fluids.

2.2.1.5 CONTAMINATION

Pilot-valve assemblies frequently include filter elements for protection against contamination damage that can affect actuation or valve leakage. The use of filters usually is limited to valve ports for inflow from the fluid supply source and from actuators. Filters in a valve assembly protect against contaminants inadvertently introduced during handling, installation, and servicing as well as contaminants in the operating fluid. Ports with undirectional outflow usually are not provided with a filter, because filters for all ports prevent outflow of contaminants, and contaminants in a valve, including particles that migrate from the filter elements, remain trapped in the valve. Filtration and filter problems are discussed in further detail in section 2.1.1.2.3.

2.2.2 Servovalves

A servovalve is defined as a modulating operator that amplifies system signals for variable-displacement, closed-loop control of actuator position. The various servovalve configurations are illustrated and discussed in handbooks such as reference 3. Reference 17 and references 21 through 32 present a comprehensive review of the state of the art in servovalve design technology.

2.2.2.1 FLOW GAIN

The flow gain of a servovalve is defined as the change in valve flowrate that accompanies a change in servovalve displacement. Flow gain of a servovalve assembly also can be defined as the change in valve flowrate that accompanies a change in valve input signal. This definition is particularly useful for characterizing torque-motor valve assemblies and pressure-actuated valve assemblies.

Obtaining the optimum flow gain by properly relating actuator velocity to servovalve input signals is a frequent problem in the design of new closed-loop actuation systems. When the flow gain is too low, actuation-system response is too slow. When the flow gain is too high, an actuator may be overly responsive to system noise, and the actuation system may be oscillatory or dynamically unstable. The solution to this problem frequently requires a servovalve designed for a nonlinear flow gain, with a low flow gain for small displacements from the valve neutral position and a high flow gain for large displacements. This design minimizes actuator response to system noise and provides for closed-loop dynamic stability under steady-state operating conditions with the servovalve near its neutral position. It provides also for fast response under transient conditions with the servovalve displaced from neutral.

2.2.2.2 PRESSURE GAIN

The pressure gain of a servovalve is defined as the change in actuator pressure, or differential pressure, that accompanies a change in servovalve displacement. The pressure gain of a servovalve at or near its neutral position can be used to relate changes in actuator force to small servovalve input signals. If the pressure gain is too low, significant actuator steady-state position errors will result from actuator load changes. The pressure gain must be sufficiently high to maintain the required positioning accuracy and system stiffness in the presence of applied load changes under all test, checkout, and operating conditions. Obtaining a sufficiently high pressure gain with the servovalve at or near its neutral position may be in conflict with a requirement for a low flow gain near neutral. Both flow gain and pressure gain must be considered in the design of a servovalve, an actuator, and a closed-loop system. Reference 17, and documents referenced therein, provide extensive coverage of this topic, including the contouring of valves and ports in shaping the flow-gain and pressure-gain characteristics.

2.2.2.3 FLUID-FLOW DYNAMIC FORCES

In a servovalve, especially in an hydraulic spool valve, fluid-flow dynamic forces may result in valve chatter, erratic operation, or dynamic instability. These problems are accentuated in pressure-actuated servovalves with the potential for interaction between sensed pressures, valve displacement, and valve flow. Occasionally, available driving forces are inadequate to overcome fluid-flow reaction forces that are additive to valve static forces. This condition usually occurs in a design based on inadequate analysis of the flow dynamics and reaction forces involved, or in a configuration in which dynamic effects are not predictable with sufficient accuracy. Most of the problems involving fluid-flow reaction forces are overcome by designing the servovalve so that the reaction forces contribute dynamic damping and oppose any tendency toward valve oscillation (refs. 17 and 21 through 32). Also, the servovalves can be designed with lateral self-centering features to minimize side forces that contribute to valve friction.

2.2.2.4 SPOOL-SLEEVE MATERIALS AND FINISHES

Linear-displacement three-way and four-way spool valves frequently are used as servovalves or as pilot valves. To ensure precise control of spool valve diametral clearances in guide bores and precise control of relative locations of spool and guide metering edges, spool valves are match fitted to guide sleeves, and finished valves and sleeves are maintained as matched sets for installation in assembly housings. The most commonly used material for spool valves and sleeves is 440C CRES, heat treated for full hardness prior to finishing. The strength, hardness, and dimensional stability of this material, in its fully hardened condition, are ideal for obtaining the sharp metering edges and smooth finishes required for servovalve applications and for obtaining high resistance to damage by fluid contaminants throughout wide temperature ranges. Although attainment of dimensional stability in 440C requires an accelerated aging process for removal of residual stresses caused by heat treatment and finishing operations, the one problem with this material in spool-sleeve applications is its susceptibility to corrosion in service or in storage if all surfaces are not smoothly finished. Corrosion has not been a problem when parts are finished smoothly without machining scratches and are protected from undue exposure to moisture or corrosive fluids during processing or storage.

2.2.2.5 SPOOL-SLEEVE CLEARANCES

The diametral clearance between a precision spool valve and its guide sleeve usually is specified in the range of 0.0002 to 0.0004 in. (5.08 to 10.2 μm). As a general practice, the sleeve is fabricated with a slightly undersized inside diameter, and the spool is fabricated with a slightly oversized outside diameter. The sleeve inside diameter is then finished by honing to obtain a spool guide surface that is a true cylinder with 0.00005 in. (1.27 μm). The spool is finished by lapping to match the sleeve diameter, and the spool and sleeve thereafter are maintained as a matched set. Because accurate verification of the specified clearance throughout the spool-sleeve axial length is difficult to obtain by dimensional inspection, flow tests are performed in which measurements of flow gain, pressure gain, and null leakage verify dimensional accuracy. Measurements of force inputs and hysteresis under flowing and nonflowing conditions provide additional verification data on friction or binding.

Diametral clearances in the vicinity of 0.0002 to 0.0004 in. (5.08 to 10.2 μm) are practical minima in the present state of the art; they provide maximum wiping action that prevents contaminants from lodging in the clearance, minimizes silting effects on operation, and minimizes leakage.

Larger diametral clearances frequently are specified for applications in which leakage or control of spool-valve functional characteristics is not critical. Larger clearances and dimensional tolerances simplify fabrication procedures, reduce costs, and lessen dimensional-stability effects and distortion effects on clearances, at the expense of greater leakage and greater sensitivity to contamination. Excessive friction and binding in close-fitting spool valves has resulted from dimensional instability or distortion of mating parts. Distortion usually results from lack of symmetry in the housing surrounding the sleeve, housing deformation, excessive static-seal compression forces applied to the sleeve outer surfaces, lack of rigidity in the sleeve when exposed to applied pressures, and thermal gradients.

Attempts to relate filtration requirements to spool-valve clearances, available valve-driving forces, system contamination levels, and contaminant size distributions on a statistical basis have been notably unsuccessful. Filters as a source of contamination are discussed in section 2.1.1.2.3. Over-emphasis on size and weight has resulted in the use of filters with insufficient surface area and with consequent inadequate contaminant-retention capacity and inadequate moisture tolerance. The use of undersized filter elements in situations with pressure-drop limitations has resulted in selection of coarse filtration when fine filtration was required. Establishing requirements for filter-element surface area and fineness of filtration is an empirical process influenced strongly by experience and judgment.

2.2.2.6 RELIABILITY

Statistically, the most unreliable components in modulating actuation systems are the feedback transducer and the torque-motor-actuated servovalve assembly. Traditionally, torque-motor-actuated servovalves have been designed for small size and weight, low electrical input power, minimum leakage, and maximum frequency response. Unnecessarily restrictive specification of these parameters results in reliability compromises involving contamination sensitivity and low operating forces and force margins. Significant improvements in reliability have been obtained by development of operator assemblies with mechanical feedback and redundant torque-motor-actuated servovalves.

Eliminating the feedback transducer eliminates its wiring, solder joints, and electrical connections as well as the power supplies, feedback amplifiers, and summing networks. Servovalve assemblies designed for mechanical position feedback can provide the fail-safe feature of actuator centering or displacement to a preferred extreme position in the event of an electrical failure. The principal disadvantages of mechanical feedback, compared with electrical feedback, are reductions in actuation-system linearity, threshold, resolution, and flexibility for shaping system-gain. With mechanical feedback, improved linearity and tolerance control is required in the servoamplifier; amplifier drift, torque-motor hysteresis, and null shift result in significant actuator positioning errors, and calibration of actuator position versus amplifier input signal must be reestablished for each system checkout. As the servovalve and actuator are an integrated assembly, gain changes require redesign, and servicing is more complex. The use of mechanical feedback has been limited to actuation systems requiring slow response and only fair positioning accuracy. The S-IC gimbal actuation system has a required frequency response flat to 1 Hz with an allowable phase lag of 25° at 1 Hz. Mechanical feedback servoactuators satisfy those requirements.

References 9 through 11 and 134 through 137 review the present state of the art relative to the emphais on reliability in space-flight modulating actuation systems and discuss mechanical feedback and redundancy concepts for torque-motor-actuated servovalves.

2.2.3 Pressure Dividers

The term "pressure divider" frequently is applied to a pneumatic or hydraulic circuit in which an intermediate pressure between two flow restrictors in series is used as a control signal. One type of pressure divider has two fixed orifices in series, and the fluid flows from a source pressure through the orifices to a receiver pressure. This type has been used for generating a signal pressure that is a function of the source pressure (sec. 2.5.2.4). Most pressure dividers in service, however, consist of one fixed flow restrictor and one variable flow restrictor in series, so that the signal pressure is a function of the source pressure, the receiver pressure, and some system parameter that controls the variable flow restrictor. The variable flow restrictor is frequently a pilot valve or a servovalve. In many applications, the variable flow restrictor is a needle valve or a vane attached to an actuator for generating a control signal pressure as a function of actuator position. Protection against contamination of a flow restrictor is discussed in section 2.1.1.2.3.

2.2.3.1 TEMPERATURE-EFFECT COMPENSATION

Temperature-related transient and steady-state calibration shifts are a common occurrence in pneumatic pressure dividers. These shifts are accentuated when gas temperature is a variable and when gas temperature is significantly different from environmental temperature. Calibration shifts result from differential expansions or contractions that affect the ratio of inlet-restrictor flow area to outlet-restrictor flow area, from temperature effects on restrictor flow coefficients, and from heat transfer between flowing gas and surrounding parts. In most cases, heat transfer is the predominant factor.

Figure 16 illustrates a prototype pressure divider that was tested to illustrate the effectiveness of counterflow passages in minimizing temperature-related calibration shifts (ref. 138). The design has not been used in rocket vehicles, but ramjet engines have utilized the principles developed. Gas at inlet pressure traverses the length of the divider through an outer jacket, reverses direction, and flows through an inner jacket and through ports in the wall supporting the inlet flow restrictor. The direction of flow then reverses again for flow through the inlet flow restrictor, the divided-pressure cavity, and the outlet flow restrictor to exhaust. The jacketed divided-pressure cavity is isolated from the environment by the counterflow passages.

In this triple-pass pressure divider, there is significant heat transfer between the inflowing gas and the walls of the outermost flow passage. There are, however, smaller temperature

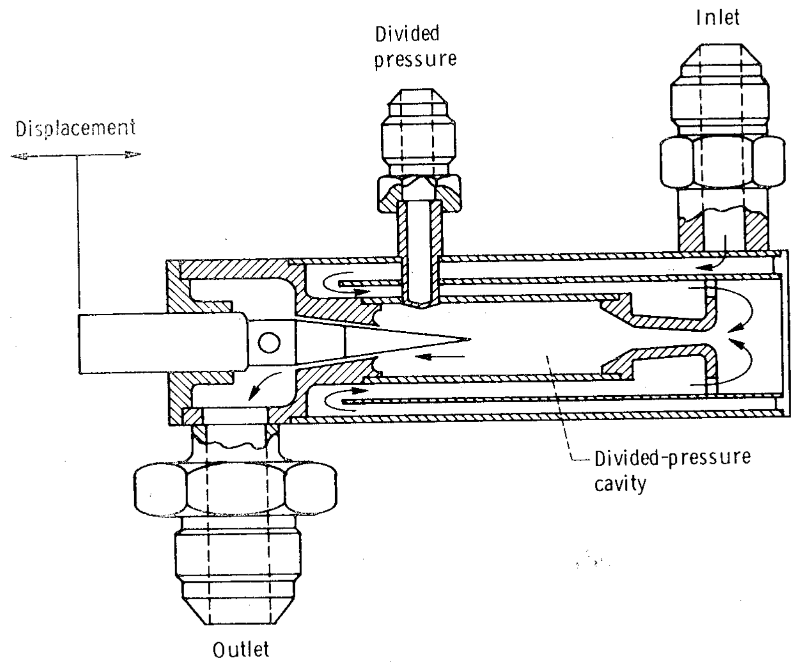


Figure 16. - Schematic of pneumatic pressure divider with temperature-effect compensation (ref. 138).

differences for heat transfer between the gas in the reverse-flow passage and its walls, and there are still smaller temperature differences for heat transfer between the gas in the divided-pressure cavity and its walls. Because heat transfer is a function of temperature differences and not of temperature level, the pressure divider of figure 16 minimizes heat-transfer effects on calibration over a wide temperature range. Heat-transfer effects under temperature transient conditions are minimized by the use of thin walls and minimum metal masses to minimize time lags in establishing equilibrium conditions. In tests of the figure 16 prototype with gas inlet temperatures ranging from 65° to 635°F (292 to 608 K), the maximum steady-state shift in the ratio of divided pressure to inlet pressure was 0.2 percent, with a room-temperature environment and with no flow demand on divided pressure. The maximum transient shift obtained was 0.8 percent.

2.2.3.2 FLOW-RESTRICTOR CONFIGURATION

Calibration shifts that resulted from variations in restrictor flow coefficients related to variations in gas flowrates, pressures, and temperatures have been observed in pneumatic pressure dividers. Because the ratio of divided pressure to inlet pressure is a function of the ratio of the effective flow areas of the inlet and outlet flow restrictors, variations in flow coefficients are as significant as variations in physical flow areas. Flow restrictors with rounded entries are subject to flow-coefficient variations related to pressure-level effects on the thickness of boundary layers at the restrictor throat. All pneumatic flow restrictors experience flow-coefficient variations as functions of pressure ratios and Reynolds number. A flow restrictor with a sharp-edged throat experiences the least variation in flow coefficient, because compressible boundary layers do not form upstream from the throat. With increasing flowrates and corresponding increasing Reynolds numbers, flow coefficients approach constants. Tests of the pressure-divider configuration shown in figure 17 (ref. 139) demonstrate negligible pressure-level effects over a wide flow range. The use of conical restrictors extends the range for restrictor sonic flow and thereby extends the range through which pressure ratios have no effect on calibration.

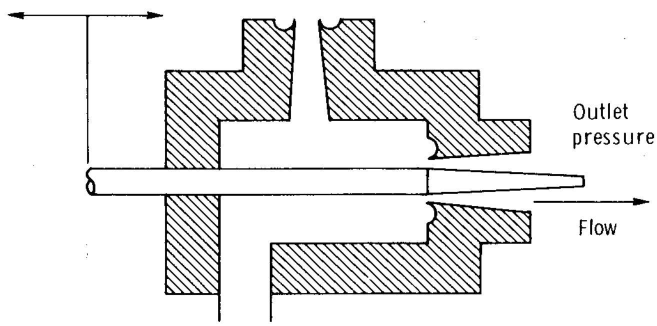


Figure 17. - Schematic of pneumatic pressure divider.

2.2.4 Advanced Concepts

Stepping-Motor operators. – The use of stepping motors as actuation elements in operators for hydraulic and pneumatic actuators is covered briefly in section 2.1.7. In addition, the discussion in section 2.1.4 on electromagnetic actuators is generally applicable to stepping motors used as operators.

Fluidic-Logic operators. – The bulk of the design and development effort related to fluidic-logic circuits and hardware for space-flight applications to date has been experimental. References 116 through 125 provide a brief survey of the published literature on this topic.

2.3 SEALS FOR ACTUATORS AND OPERATORS

The state of the art in the design and application of sealing devices and sealing techniques is reported extensively in the technical literature (refs. 3 and 140 through 147). This monograph, therefore, is concerned only with specific problems and problem solutions in the design of seals for actuators and operators used in space-flight systems.

2.3.1 General Considerations

2.3.1.1 COMPATIBILITY WITH FLUIDS

Material chemical compatibility has been a problem in bipropellant systems when a seal that was compatible with the propellant to which its exposure was intended was not compatible with the other propellant. Cross-compatibility problems in bipropellant systems usually have been associated with leakage of propellant isolation devices or with reverse-flow transients. These problems also have occurred when bipropellant system components that were similar or identical except for the seal materials were assembled incorrectly, and the seals were exposed to the wrong fluids.

Inadequate control of procedures for fabrication, handling, installation, and servicing frequently has resulted in seal compatibility problems. For example, although Kel-F is compatible with oxygen and hydrogen and is used in many seal applications, it is subject to cracking at stress levels of about one-fifth its yield strength when exposed to commonly used halogenated cleaning and flushing solvents. Improper use of lubricants for seal installation has resulted in compatibility problems during subsequent exposure to operating fluids or solvents.

Components designed for service with hazardous fluids frequently are tested with less hazardous fluids. The use of volatile solvents as test fluids avoids component contamination during testing, but requires that the seal material be compatible with the test fluids.

2.3.1.2 LEAKAGE MEASUREMENT

Among the most common problems associated with the use of seals are the failure to detect deficient seals and the rejection of adequate seals as a result of inaccurate or nonrepeatable leakage measurements. Frequently, these problems result from failure to design for convenient and sufficiently accurate leakage testing. Very small leakage flowrates through seals in pneumatic actuators cannot be measured accurately when testing requires measurement of leakage from a pressurized cavity into a relatively large reservoir cavity. The length of time required for a small leakage to increase the reservoir pressure sufficiently for a flow measurement permits the effects of temperature changes during a test to confuse the measurements. Assembly designs frequently require measurement of the combined leakage of more than one seal into a common cavity with no provision for separating the measurement of leakages through critical and noncritical seals. Poor control of test equipment and procedures results in inconsistent test results when tests are performed in different manufacturing, test, and laboratory locations. Problems arise when the same leakage limits are specified for all tests. An assembly that barely passes an acceptance test at the component level may fail to pass an engine-system leakage test, when the same limits are specified, as a result of differences in test conditions or equipment.

2.3.1.3 SEEPAGE AT LOW DIFFERENTIAL PRESSURE

Hydraulic seepage has occurred when seals that were designed to be loaded by large differential pressures were loaded by small differential pressures under system standby or pressurized-storage conditions. This type of failure often can be attributed to seals that were installed with insufficient compression or that lost part of the installed compression as a result of yielding. Careful attention to dimensional tolerances affecting seal compression and to seal loading under static head as well as operational conditions is required.

2.3.1.4 INADVERTENT CONTACT SEALING

Variations and discrepancies in breakaway-response timing of piston actuators have occurred when a piston mechanical stop in an actuator inadvertently acted as a seal or semiseal. As illustrated in figure 18, when the piston is in the position shown, the applied pressure cannot actuate the piston until sufficient fluid has leaked through the mechanical-stop contact surface to raise the cylinder pressure to the level required for breakaway against opposing forces. A similar response problem may exist in the opposite direction when the piston has been fully actuated. In the actuated condition, the inside diameter of the piston in contact with the housing may constitute a seal that retards pressurization of the piston area between the inside and outside diameters. Because of fluid-compressibility effects, variations and discrepancies in breakaway-response timing are more pronounced when gas is the actuation fluid or when an hydraulic actuator is not primed. This kind of problem is avoided when the piston stops are designed to ensure free flow of actuation fluid to the entire effective area of the piston.

2.3.2 Dynamic Seals

2.3.2.1 SEAL CONTACT SURFACES

Dynamic seals exhibit a tendency to fuse to cylinder walls during long-term periods of inactivity. This problem usually occurs when seal lubricants have been removed by flushing

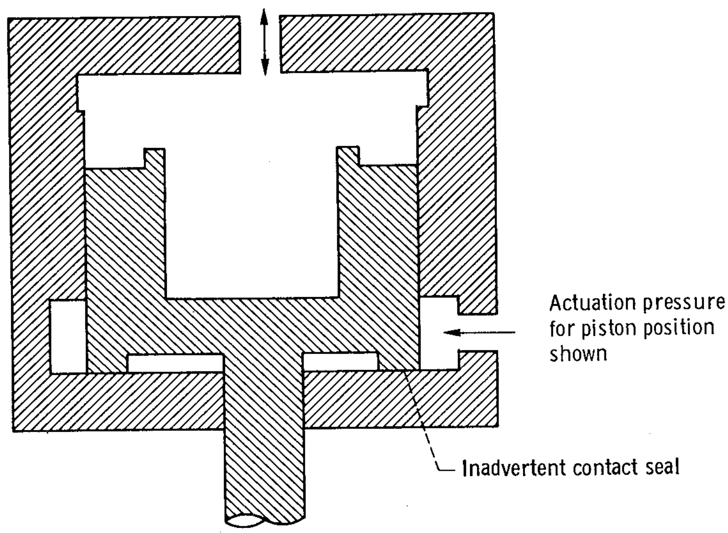


Figure 18. - Sketch showing inadvertent contact sealing in a piston actuator.

or when an operating liquid has dried. A significant increase in actuator breakaway force may be required for the first subsequent actuation, and seal damage may impair sealing capabilities. If a contact surface is too rough, a pliable seal that conforms to surface irregularities may experience excessive adhesion, breakout resistance, sliding friction, and seal abrasion. Similar problems have occurred when surface finishes were too smooth to support lubrication films. Surface finishes of parts in contact with dynamic seals usually are specified to be consistent with requirements that have been established for the specific seal material and seal configuration to be used. Seal lubrication procedures sometimes are specified for use in preparing components for storage or other periods of inactivity.

System contamination has resulted from solid-particle breakout from dynamic seals. Elastomeric seals usually are more pliable than plastic seals, and thus have a greater tendency to conform to surface irregularities, to adhere to contact surfaces, and to act as contaminant generators. The sensitivity of pliable dynamic seals to damage resulting from contact with system contaminants is generally recognized, but the potential for component or system contamination by particles from dynamic seals frequently is overlooked. Success in preventing or avoiding seal-particle contamination requires specific emphasis on the design of the seal and the seal contact surfaces and on the choice of seal material.

Seal damage or excessive wear has resulted from inadequate specification or from misinterpretation of drawing notes identifying sealing surfaces. The American Standards Association (ASA) specification for surface roughness (ref. 148) permits surface flaws unless a surface-finish symbol is accompanied by a drawing note defining limits for surface imperfections. When pistons or piston rods coated with hard materials are in contact with dynamic seals, the surface porosity in materials such as the oxides and carbides often results in rapid degradation of the seals. The use of inherently porous or abrasive materials at seal contact surfaces usually is avoided, a practice that sometimes results in the use of bellows or diaphragm dynamic seals with no sliding contact.

2.3.2.2 LIP SEALS FOR CRYOGENICS

Plastic lip seals as illustrated in figure 19 remain pliable and provide adequate sealing in pneumatic actuators that operate at cryogenic temperatures. Mylar, Kel-F, and Teflon have been used successfully as lip seal materials. Mylar is the preferred material for the configuration of figure 19 because it has good room-temperature forming characteristics and excellent low-temperature flexibility.

View A in figure 19 depicts a dynamic seal for a double-acting piston that also is used as a static seal for the interface between the actuator end cap and housing. The metal washer that separates the sets of seals in view A has a single serration on each face for localized compression of the flat portion of the Mylar in obtaining a static seal. Each seal element consists of a double layer, one layer in contact with the piston for sealing and one backup layer to load the sealing layer and to provide deformation resistance when pressurized. The seal is designed to withstand system proof pressure applied to either side of the actuator piston. Optimum seal material thicknesses and dimensions for the seal and its retaining details are established by development testing to obtain or verify sealing capabilities and resistance to deformation throughout the full range of applied pressures.

View B in figure 19 illustrates the piston-rod seal. Although dynamic sealing is required in one direction only, a dual element is used, with one element acting as a wiper to protect the active element from exposure to potential contamination from the piston rod linkage cavity. The metal washer between the active element and the wiper element has single serrations for compression of the static portion of each seal element. This seal configuration has been used successfully at nominal operating pressures of 400 and 500 psi (2.76 and 3.45 MN/m^2).

For high-differential-pressure applications, when the lip seal design illustrated in figure 19 does not have sufficient strength and rigidity to resist damage or deformation, the design concepts illustrated in figure 20 have been used. In the design in figure 20(a), a Teflon backup ring supports the seal. At room temperature, the diametral clearance between the backup ring and the shaft is the minimum clearance required to avoid interference as a result of dimensional tolerances. Tests indicate an operational-pressure upper limit of 1500 psi (10.3 MN/m^2) for successful use of this seal configuration.

A program to develop lip seals for pistons and shafts in support of M-1 engine actuator and valve design demonstrated the need for relatively heavy cross sections with backup

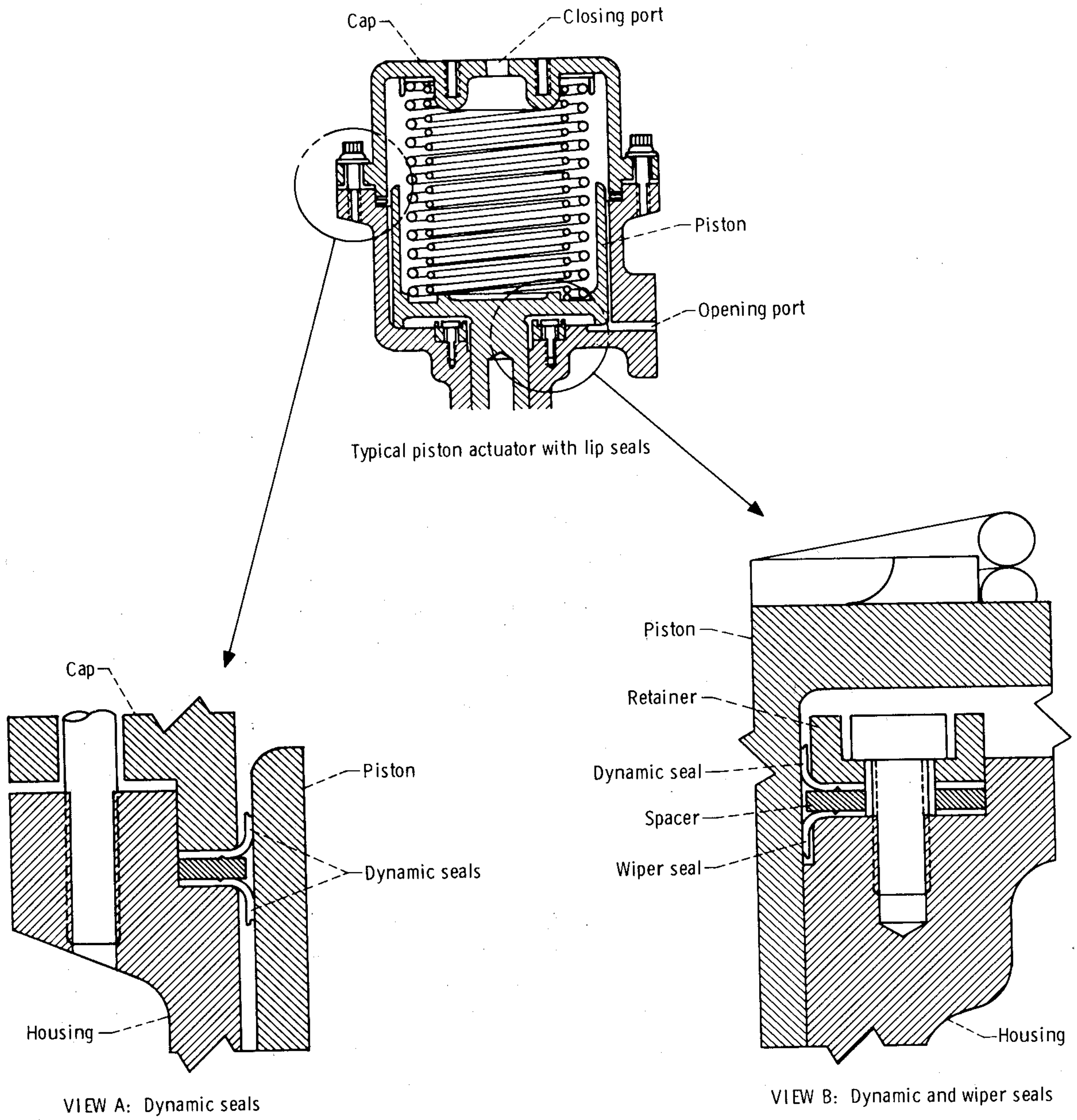
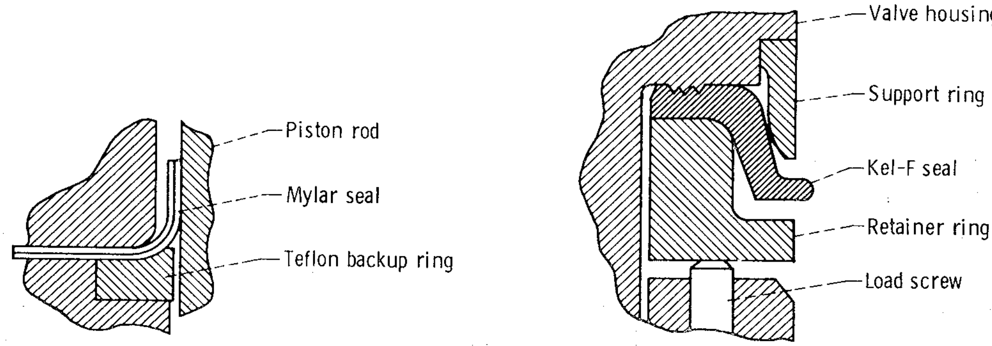


Figure 19. - Schematic of piston actuator with lip seals.



(a) Adequate to 1500 psi (10.3 MN/m^2).

(b) Adequate to 2000 psi (13.8 MN/m^2).

Figure 20. - Lip seal designs for high-pressure cryogenic conditions.

supporting elements for successful operation in high-pressure systems. Kel-F seals of the configuration illustrated in figure 20(b) performed successfully under cryogenic conditions at differential pressures of 2000 psi (13.8 MN/m^2) (ref. 149).

2.3.2.3 WIPER SEALS

Plastic and elastomeric dynamic seals are vulnerable to damage by solid-particle contaminants. Dynamic seals on piston rods exposed to an actuator external environment or to cavities containing contaminant-generating mechanical linkages or other transmission elements are particularly vulnerable. Contaminants generated by mechanical springs have resulted in numerous dynamic-seal failures. Degradation of piston seals in hydraulic actuators may be tolerable when seal leakage is additive to through-piston circulation flow for priming. In pneumatic actuators, however, and in single-acting hydraulic piston actuators with no circulation flow, piston-seal damage has resulted in response-timing failures as well as leakage-limit failures. Visible external leakage through piston-rod seals has resulted in rejection of actuators in which the leakage was insufficient to impair performance.

When exposure to contaminants is anticipated, wiper seals frequently are used to protect critical dynamic seals, as illustrated in view B of figure 19. The only function of the wiper seal is to scrub contaminants from piston or rod surfaces that contact dynamic seals. Contamination damage of a wiper seal is acceptable as long as protection of the dynamic seal is maintained. The primary problem with wiper seals is maintaining their protective capability when particles break out from edges exposed to contaminants. The wiper seal in view B of figure 19 has an appreciable axial length of contact surface with the piston, and particles that break out from the leading edge cannot readily pass through the contact surface and migrate to the dynamic seal.

2.3.3 Static Seals

2.3.3.1 SEAL COMPRESSION FORCES

Static-seal configurations that require relatively large compressive forces for proper installation (e.g., metal 0-rings with or without coatings, and metal gaskets requiring indentation by serrations) are prone to leakage through distorted contact surfaces. Static-seal leakage has resulted from displacement of seal contact surfaces or from loss of seal compressive force under pressure loads or impact loads when seal retaining surfaces had not been fully seated during installation. Seals requiring large compressive forces perform satisfactorily when fastener forces are sufficiently high, retaining parts are designed for rigidity, and assembly processes are controlled to ensure adequate and uniform seal load distribution.

2.3.3.2 SEAL CONTACT SURFACES

Static-seal leakage as a result of flaws in seal surfaces or in seal-contacting surfaces is a common occurrence. Because the ASA specification for surface roughness (ref. 148) permits surface flaws that can damage seals, production drawings frequently identify sealing surfaces, and surface-finish symbols are accompanied by drawing notes that limit surface imperfections. Drawings often refer to machining specifications in which fabrication and inspection requirements for sealing surfaces supplement the ASA surface-roughness specification.

2.3.4 Seal Welds

2.3.4.1 PROVISIONS FOR PRELIMINARY TESTING

Seal welding of actuator and operator joint seams and of plumbing connections is common practice for minimizing the potential for external leakage; a seal weld is intended only for sealing a seam or joint, not for supplying structural strength. Some seal-welded assemblies such as pressure-actuated devices that must operate at preset pressure levels require internal adjustments during preliminary testing, and designs must provide for tests and partial disassembly for adjustment prior to welding. Well-designed seal-welded assemblies provide for temporary plumbing connections and temporary static seals for convenient preliminary screening tests prior to welding and component formal acceptance testing.

2.3.4.2 PROVISIONS FOR DISASSEMBLY AND REASSEMBLY

Major parts have been damaged when removal of seal welds was required for servicing or overhaul of assemblies. Such damage usually results from deep weld penetration that requires removal of an excessive amount of adjoining material or from failure to design assembly joints in a manner that permits seal-weld removal and rewelding a reasonable number of times. Actuators and operators usually are designed so that structural loads will not be imparted to seal-welded joint seams, and minimum weld penetration is required for fluid pressure sealing. Joints usually are designed to accommodate weld-seam removal by machining and to provide sufficient material for multiple weld-removal and reweld operations.

2.4 MECHANICAL TRANSMISSION

The technical literature provides extensive illustration and discussion of the state of the art related to mechanical transmission of actuator forces and displacements. Discussion in this monograph, therefore, is limited to a review of specific problems and solutions that are critical in the design of mechanical transmissions for space-flight hardware.

2.4.1 Common Problems

2.4.1.1 MOISTURE IN VENTED CAVITIES

In many actuator assemblies, the power transmission elements are situated in cavities that are vented to the external environment. For space-flight service, these cavities usually are protected by low-cracking-pressure relief valves against entry of moisture from the environment; these valves, frequently referred to as vent port relief valves or as vent port check valves, permit outflow from the cavities and seal against inflow and moisture migration. Vented cavities often serve as receivers for pneumatic-fluid leakage or operator discharge flow, and vent port relief valves are sized to accommodate gaseous outflow with very little increase in cavity pressure. Occasionally, moisture has migrated from the environment into vent cavities when partial vacuums accompanied hardware chilldown or when vent-cavity gases were displaced during an actuation cycle. Vent port relief valves have failed to relieve vent-cavity pressure when frozen external moisture impeded valve opening. Frozen moisture in vent cavities containing transmission elements has been suspected as a cause of erratic actuator operation. Instrumentation elements (e.g., actuator position indicators and electrical connectors) exposed to vent cavities are especially sensitive to moisture damage.

Design features that eliminate or minimize problems with these valves are presented in reference 150.

2.4.1.2 FASTENER RETENTION

One of the most common failure modes encountered in the mechanical transmission of actuator power is loosening or disengagement of fasteners. Fastener failures have occurred when excessive impact forces distorted fasteners and linkage elements. More frequently, fastener failures have resulted from lack of positive retention features, from inadequate retention features, or from improper installation of fasteners with locking features.

Self-locking nuts and self-locking tapped bolthole inserts that rely on mechanical interference of distorted threads have gained acceptance as the preferred methods for retention of internal fasteners in an actuator and linkage assembly. Fasteners with elastic or plastic inserts that depend on insert compression and deformation forces for retaining the fasteners have been less reliable than the distorted-thread types, especially when fasteners are reused or when fasteners are exposed to large temperature ranges.

The most common problem with self-locking fasteners has been the uncertainty of the torque specification for securing the fasteners. The running torque required for rotation against the resistance of the locking feature can vary over a wide range. If the total applied torque is specified, the clamping force will vary as a function of the interference running torque. Better control of the clamping force is obtained when the torque required for clamping is specified as a torque value to be applied in addition to whatever torque is required during installation for overcoming the resistance of the locking feature.

Lockwires, cotter pins, and bent-tab washers used in mechanical transmissions exposed to impact loads and vibration have been damaged during installation, causing stress concentrations that resulted in subsequent failures. Thread-locking adhesive compounds are available for wide ranges of temperature; their use is increasing, especially for subassemblies that can be serviced by replacement rather than by overhaul. Shoulder bolts with self-locking nuts are preferred over linkage pins with snap-ring retainers, which require sharp-cornered grooves in hardened parts exposed to impact and vibration and are vulnerable to improper installation.

Set screws rarely are used in space-flight hardware because their scrubbing mode of contact generates contaminants and because they are loosened by vibration. Staking of fasteners produces metal chips and is an unreliable fastener retention method. Roll pins and spiral lock pins have been used successfully where a completed assembly prevented or limited pin disengagement.

2.4.1.3 GUIDES

2.4.1.3.1 Materials and Configurations

Many space-flight applications for linear-motion actuators and power transmissions require the use of nonlubricated guide surfaces with sliding contact. Unacceptable wear, abrasion, or galling can be reduced or eliminated by the use of material for pistons, piston rods, and linkage members that can be heat treated for hardness, and by the use of guide materials suitable for nonlubricated sliding contact. The design concept of figure 21, in which a combination of Mylar lip seals and Teflon guides in an aluminum housing supports a piston and piston rod made from Inconel 718, has been used successfully in pneumatic actuators exposed to cryogenic temperatures. The metallic materials were selected to minimize damage if inadvertent metal-to-metal contact should occur.

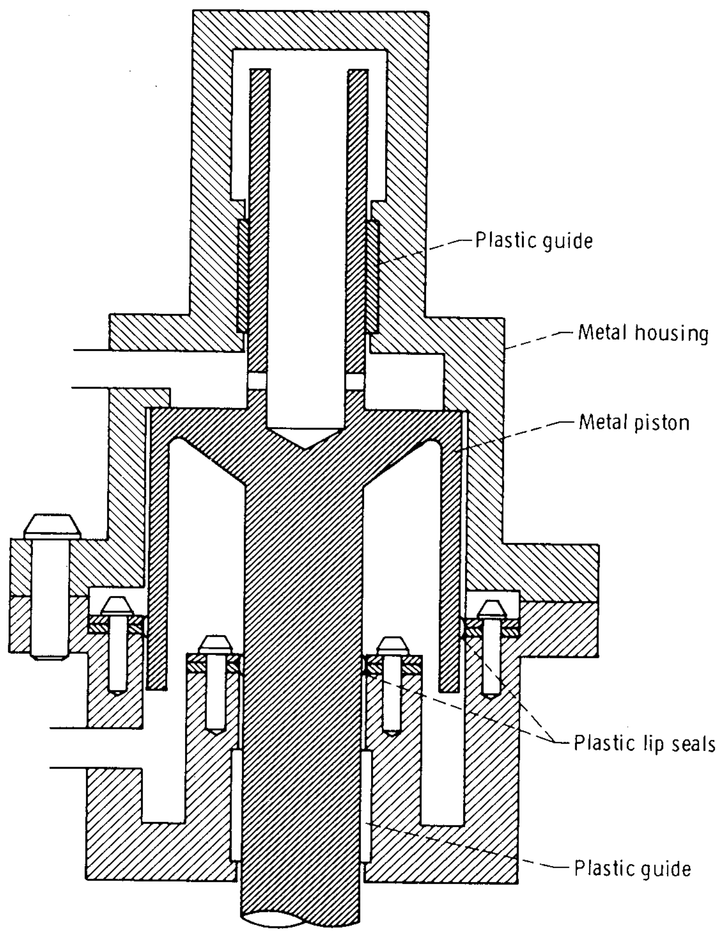


Figure 21. - Schematic of actuator with nonmetal seals and guides combined with metal housing and piston.

In some actuator applications, such as in piston-actuated valves for reactive fluids, design for nonlubricated metal-to-metal relative motion between moving elements and their guides may be required. Material chemical compatibility may be reduced by friction, seizing, wear, scoring, or galling at metal-to-metal contact surfaces. Damage to dynamic seals is inherent if sealing surfaces are subjected to intentional or inadvertent metal-to-metal contact. Problems associated with metal-to-metal sliding contact, friction forces, misalignment, rotation of power-transmission elements, and applied vibration in short-stroke actuators have been minimized or eliminated in many actuator and power-transmission designs by the use of metal flexure guides. Metal flexure guides are designed for flexibility in the axial-displacement direction and for stiffness in the directions of lateral and rotational displacement. Figure 22 illustrates a typical flexure-guide application in which two flexure elements suspend and guide a solenoid-actuated valve stem. The complex geometry of flexure elements of this configuration frequently requires development effort in obtaining elements with the desired mechanical spring rates.

2.4.1.3.2 Lengths and Clearances

As general practice intended to minimize binding or chatter, linear-motion actuators are designed for an effective guide length that satisfies one of the following conditions (refer to fig. 23):

$$\ell_1/D \geq 1$$

$$\ell_2/d \geq 3$$

$$L/D \geq 1$$

where

ℓ_1 = piston guide length

ℓ_2 = piston-rod guide length

L = effective guide length with guiding at ℓ_1 and ℓ_2

D = cylinder bore diameter

d = piston-rod diameter

If we consider the following additional parameters

C_1 = diametral clearance at axial length ℓ_1

C_2 = diametral clearance at axial length ℓ_2

e_1 = eccentricity of piston skirt to piston rod

e_2 = eccentricity of cylinder wall to piston-rod guide

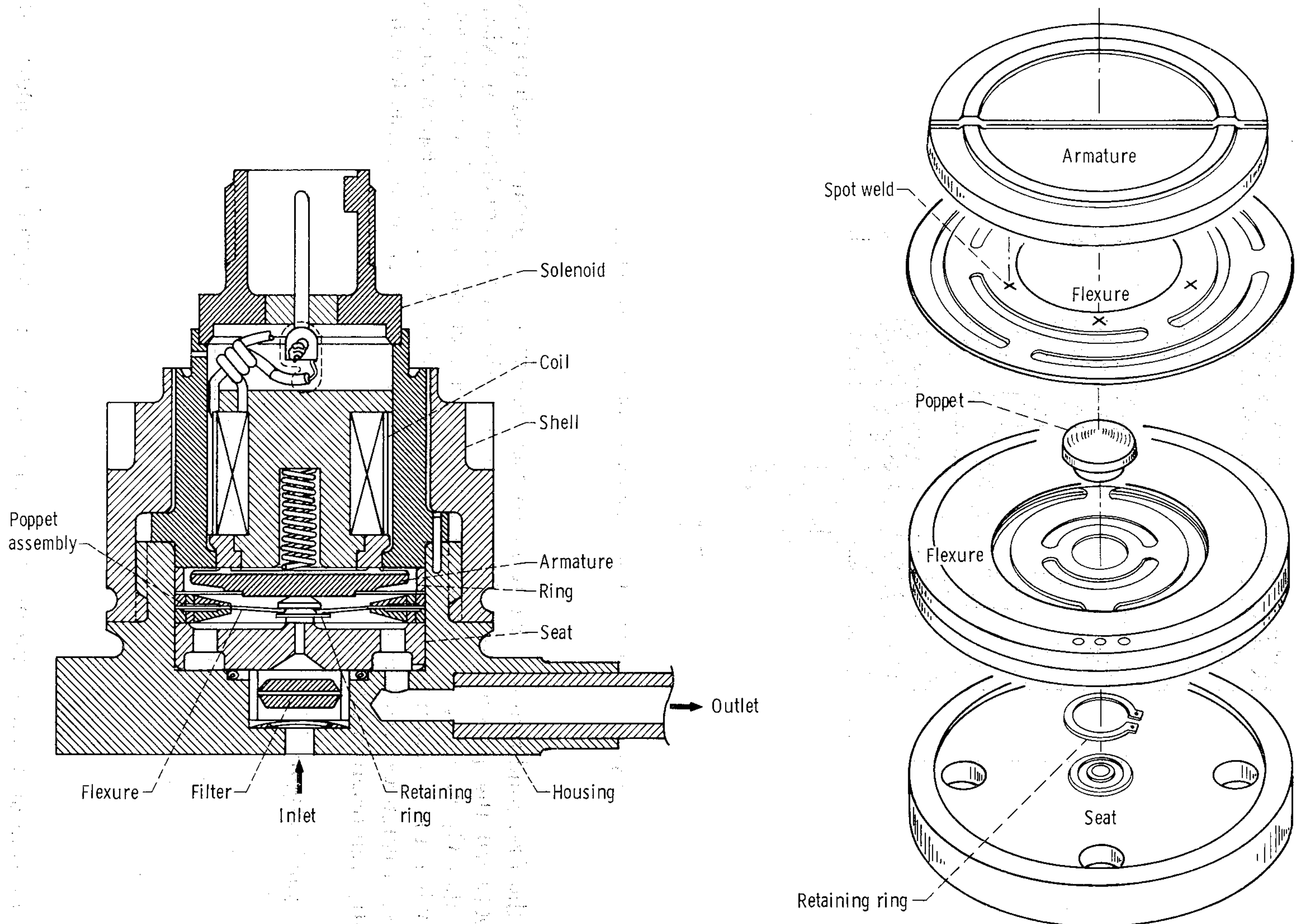


Figure 22. - Typical use of flexure guides in a solenoid-actuated valve.

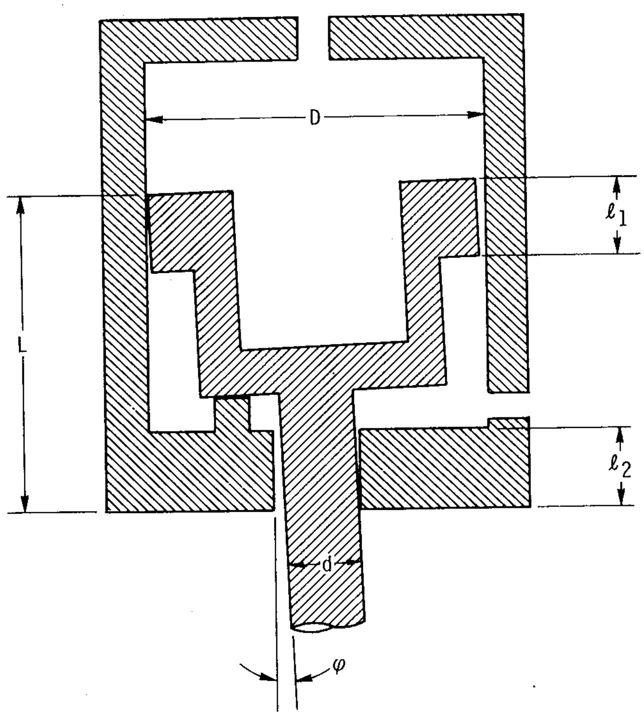


Figure 23. - Sketch illustrating parameters involved in piston guiding.

then, for the first condition to exist, it is also necessary that C_2 be great enough to preclude contact between the piston rod and its guide. For the second condition to exist, it is necessary that C_1 be great enough to preclude contact between the piston skirt and the cylinder. When the design intent is that the third condition exist, piston chatter can occur if dimensional tolerances result in length ℓ_1 or ℓ_2 becoming the effective guide length; also, eccentricities can result in binding.

Referring again to figure 23,

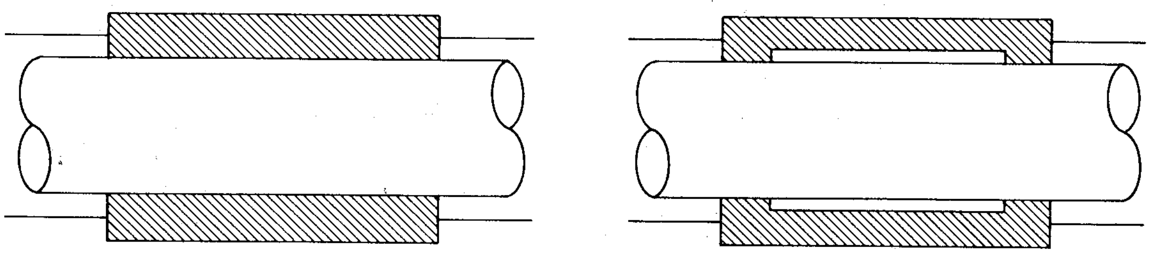
$$\tan \phi = \frac{1/2\,(C_1 + C_2) + e_1 + e_2}{L}$$

where

ϕ = maximum possible piston axial angularity with respect to cylinder axis for clearances C_1 and C_2 over length L

If $C_1/\ell_1 < \tan\phi$, the piston rod will not contact its guide (assuming sharp corners), and ℓ_1 becomes the effective guide length. If $C_2/\ell_2 < \tan\phi$, the piston will not contact the cylinder wall, and ℓ_2 becomes the effective guide length. For length L to be the effective guide length, the eccentricities, normalities, and minimum clearances must be adequately controlled, with allowances for differential expansion or contraction throughout the temperature range and for deflections under all loading conditions.

Transient and steady-state temperature distributions throughout an actuator assembly frequently result in excessive friction or in binding when guide lengths and clearances do not accommodate differential thermal expansion or contraction of adjoining parts or temperature-related distortions. Figure 24 shows two basic types of rod guides. Relatively long continuous guides for axial-displacement rods, (fig. 24(a)), are more susceptible to thermally induced binding than are the undercut guides (fig. 24(b)), which have the same effective guide length but less surface for potential contact and interference.



(a) Continuous rod guide.

(b) Undercut rod guide.

Figure 24. - Basic types of rod guides.

2.5 INSTRUMENTATION FOR ACTUATORS AND OPERATORS

A discussion of the state of the art in the design of instrumentation *per se* is beyond the scope of this monograph; references 3, 101, 151, and 152 are typical of the extensive literature available on this specialized subject. This section, therefore, is limited to a review of problems and solutions that are critical in the successful use of instrumentation with space-flight actuators and operators.

2.5.1 General Requirements

2.5.1.1 ISOLATION OF FUNCTIONAL INSTRUMENTATION

Flight instrumentation associated with actuators and operators may be required for system functioning or for system monitoring. In addition, flight hardware may require provisions for the installation of checkout and test instrumentation that is not required for flight.

The reliability of a space-flight hardware system is enhanced when instrumentation for system functioning and for system monitoring are designed and installed so that a malfunction of monitoring instrumentation cannot affect operation of the system. The prevailing philosophy is that functional operation of a flight-hardware system is of primary importance and that the addition of instrumentation for information purposes should in no way detract from functional reliability. For example, the design requirements for an actuator may include a requirement for electrical indication that the actuator displacement is within a specified percentage of the full-stroke position; the indication is to be used both for initiating another event in operation of the flight system and also for monitoring of flight performance. If one position switch is used to serve both purposes, the functional and the monitoring instrumentation subsystems become interrelated at that switch, and a malfunction in one subsystem could affect performance of the other. The use of two actuator position switches, one for each subsystem, adds to the complexity of the actuator design but eliminates a location for potential interraction between the subsystems

2.5.1.2 SAFE FAILURE MODES

Structural failure in a pressure-sensing instrument, such as rupture of a pressure transducer diaphragm, has resulted in system fluid outflow with consequent fire hazard or system pressure loss. Similar failure modes have resulted from structural failures in signal-pressure lines to remotely located instruments. Operational failures have occurred in mechanisms with position-indicating devices when mechanical failure of an indicator has interfered with actuation of the mechanism.

The reliability of a space-flight system is enhanced when instrumentation is designed and installed so that instrumentation structural or operational failures do not impair system functioning. To the maximum feasible extent, instrumentation is designed for fail-safe system operation (i.e., safe operation of a system continues in the event of an instrumentation failure). Redundant indicators often are used when instrumentation signals are essential in system operation.

2.5.1.3 ELECTRICAL CONNECTORS

Electrical connectors for flight instrumentation for actuators and operators include both hermetically sealed connectors that are integral parts of indicating devices (e.g., pressure transducers and position potentiometers), and connectors that are not hermetically sealed but usually are attached to actuators with a flanged joint and a static seal. Connectors with direct exposure to actuator internal cavities have failed when moisture and other contaminants corroded or short-circuited the pins and wire attachments, or when the connectors were mechanically damaged by inadvertent application of excessive pressure. These problems are minimized when an electrical connector can be attached to an actuator in a location that is exposed only to a vented cavity with an inert environment.

2.5.1.4 ACCESSIBILITY

When flight instrumentation is required for actuators and operators, the attachment locations, electrical connector locations, electrical conduit routing, and conduit support locations require the same consideration that is given to hydraulic or pneumatic plumbing systems. Providing accessibility for installation, calibration, and servicing often is in conflict with the need to safeguard instrumentation-system components from exposure to handling damage or accidental impact damage.

Requirements for checkout and test instrumentation at the system level frequently are not established fully in the component design phase, and the needs for test instrumentation accessibility do not always receive due consideration. Instrumentation locations that are adequate for component level testing are not always accessible when components are installed in a system. There are no general rules governing instrumentation accessibility. Problems are avoided only when all component and system instrumentation needs are investigated early in a design program, due account being given the physical orientation of components as installed in a system.

2.5.2 Pressure Measurement

2.5.2.1 PRESSURE-SENSING LOCATIONS

Continuous measurement of actuator internal pressures permits computation of the actuator forces that exist during motion transients. Oscillograph traces of the internal pressures during motion transients indicate variations in actuator forces as functions of actuator displacement, and indicate discontinuous forces resulting from mechanical binding or from static friction that exceeds dynamic friction. When timing-control flow restrictors, plumbing fittings, or housing flow passages introduce significant pressure differences between externally applied pressures and actuator internal pressures during motion transients, the external-pressure measurements provide erroneous indication of internal pressures; and information thus obtained can be misleading in a performance analysis. External-pressure measurement may require static-pressure recovery corrections to allow for differences in flow velocity in the actuator and at the pressure-sensing location. Whenever possible, therefore, actuators are designed so that internal pressure may be measured directly.

2.5.2.2 TEST-INSTRUMENTATION EFFECTS

The addition of pressure-measuring instrumentation to an operator-controlled actuator for test or checkout purposes can alter its performance to the extent that the information obtained under dynamic conditions is erroneous or to the extent that dynamic performance is seriously impaired. The effect of instrumentation is especially critical in modulating systems, in pneumatic systems with small pressurized volumes, and in hydraulic systems in which fluid-compressibility or bulk-modulus effects are critical. The addition of instrumentation to a system with compressible fluids can result in an increase in a pressurized volume and thereby increase the time required for pressure changes in that volume. In hydraulic systems, instrumentation that has not been primed (gases removed) or instrumentation with flexible pressure-sensing elements can increase the time required for pressure changes and can reduce the hydraulic stiffness of the system. Accordingly, the possibility of any of these factors affecting the specific test setup and instrumentation is evaluated, and the effects of the instrumentation on system performance are established. References 153 through 157 review the dynamic response characteristics of signal-pressure transmission lines.

2.5.2.3 FLIGHT-INSTRUMENTATION EFFECTS

When actuators and operators are designed for applications in which flight-system instrumentation is included, the same considerations given to the effects of test instrumentation on actuator and operator performance are applicable, except that the effects of instrumentation volumes and flexibilities must be factored into the basic design considerations for satisfying overall system performance requirements.

2.5.2.4 PRESSURE-TRANSIENT EFFECTS

The transient response of pressure-sensing instrumentation occasionally has been impaired when the instrumentation was situated remotely from sensed-pressure locations. Dynamic response in hydraulic systems and in pneumatic systems has been impaired as a result of fluid-compressibility effects. Figure 25 illustrates a technique that has been used for remotely located pressure switches in an hydraulic system; the technique ensures self-priming of a signal-pressure line by removing gases from the line prior to operation of a switch.

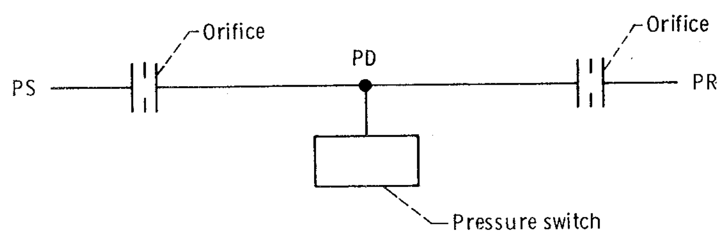


Figure 25. - Schematic of self-priming instrumentation line for hydraulic system.

Under standby conditions prior to operation, gases are removed by a circulation flow from the sensed-pressure location PS through two identical orifices in series to system return pressure PR. As the sensed pressure increases during system operation, flow through the identical orifices results in a pressure PD at the pressure switch that is midway between the sensed pressure and system return pressure. The pressure switch is preset to actuate at approximately one-half the desired sensed-pressure level for actuation. A pressure transducer similarly situated in an hydraulic system would indicate approximately one-half the sensed pressure, a primed sensing line being utilized to minimize fluid compressibility effects. If the sensed-pressure level is high relative to system return pressure, variations in system return pressure will have only minor effect on the accuracy of instrumentation calibration. Pressure-dividing circuits are discussed in greater detail in section 2.2.3.

2.5.3 Position Indication

2.5.3.1 BACKLASH AND HYSTERESIS

In closed-loop control of actuator position, backlash or hysteresis that results in instrumentation deadband will limit positioning accuracy and can result in hunting or dynamic instability. After a dynamic analysis of a closed-loop system has been made, design requirements are established to define the maximum allowable deadband allotted to the combination of actuator, linkage, and indicating device. An indicating device and its attachment method then are selected or designed for performance within the allowable limits.

2.5.3.2 ATTACHMENT MECHANICAL INTEGRITY

Attachment of a position-indicating device such as a potentiometer or transducer to an actuator or its mechanical linkage usually requires some provision for accommodating minor misalignment resulting from assembly clearances and dimensional tolerances. In cases wherein a limited mechanical backlash is allowable, attachments sometimes are pinned, pin clearance providing an allowance for misalignment. In cases wherein backlash is undesirable, a flexible element firmly attached to the indicator and to the actuator (e.g., a relatively long thin rod with threaded ends) sometimes is used. Position indicators have been designed with a spring-loaded follower element bearing against an actuator piston or its linkage, or with a spring-biased pinned attachment.

The most common problems with attachments are (1) buckling, extension, or distortion of flexible attachment members under fast displacement transients; (2) enlargement of mechanical clearances under displacement-transient and vibration conditions; (3) loosening of attachment fasteners; and (4) indicator forces (including friction, mechanical binding, pressure, and inertia forces) exceeding bias-spring forces under shock, vibration, and displacement-transient conditions. Attachments that rely on spring forces alone to maintain follower contact are the least reliable. Most attachment problems result from errors in predicting the maximum transient forces that will be applied to an indicator attachment. These problems are handled effectively only when adequate dynamic-force analyses are carried out and when the design emphasizes mechanical reliability in complying with performance requirements.

2.5.3.3 POSITION-INDICATOR RELIABILITY

As discussed in section 2.2.2.6, actuator position transducers, including potentiometers, are among the least reliable components in flight hardware systems. Mechanical position-feedback methods were developed for the gimbal actuation systems on the Saturn vehicles, so that failure modes associated with electrical position transducers and their circuits were eliminated. Approximately 90 percent of all unsatisfactory condition reports recorded for the F-l engine piston-actuated main propellant valves pertain to the position-indicating potentiometers in those valve assemblies. Because these valve-position indicators are used for information only, they have not caused any operational failures; but they have introduced servicing problems.

Position transducers are useful in the checkout or testing of open-loop actuation control systems when actuator response timing is being established or verified. Frequently, electrical switches that indicate the two extremes of piston displacement can serve the same purpose. Actuation-fluid pressure measurements, or differential-pressure measurements across a flow restrictor, can provide indirect indication of the velocity of the actuator piston. Hydraulic and pneumatic pressure-divider circuits, for on-off or for continuous position indication, provide mechanical alternates for electrical switches and potentiometers.

Reliability problems with position indicators are being solved by designing systems to eliminate the need for such indicators and by using redundant indicators and instrumentation circuits.

2.5.4 Temperature Measurement

Temperature-measuring instrumentation for space-flight actuators and operators usually is limited to thermocouples for monitoring temperature during component environmental tests and system development tests. Special design provisions for the attachment of thermocouples seldom are required, because thermocouples for test purposes can be attached temporarily to production units with tape or adhesives. For development tests, assembly fasteners can be modified for more secure attachment.

Misleading results have been obtained in environmental tests of actuators and operators when measurement of the ambient temperature has been substituted for direct measurement of housing temperatures. When component tests are performed in environmental chambers, the time required for component temperature stabilization may be many hours. Large rocket engine piston-actuated main propellant valves submerged in liquid nitrogen require approximately 45 minutes for internal-temperature stabilization, about twice as long as the elapsed time during which active bubbling in the nitrogen is visible evidence of continuing heat transfer. When environmental tests are intended to provide thermal-effect information on critical performance features such as actuator response timing and seal leakage, it is essential that actual temperature distributions in the hardware be known. For true indication of hardware temperatures, thermocouples are attached directly to components. Usually, extensive tests are requred to determine the most appropriate attachment points and to establish the elapsed time required for temperature stabilization.

2.5.5 Vibration-Test Measurement

Vibration testing of space-flight actuators and operators, at the component level or system level, ordinarily is required in qualification testing and occasionally is required in acceptance testing. Additional vibration tests may be performed during the development phase of a hardware program.

Acceleration levels are measured during vibration testing of components and systems by accelerometers attached at carefully selected points. The preferred attachment points are at locations where maximum accelerations occur as a result of amplification of input vibration. When a design configuration did not provide for convenient firm attachment of accelerometers, attachment has been difficult, accelerometers have detached during tests, and test results have been inaccurate; these problems are avoided when the instrumentation needs are anticipated in the design of a component. When a configuration does not permit strap-on attachments, mounting pads sometimes are provided to permit drilling and tapping for attachment of accelerometers. When the spring mass of a system is small, special precautions are taken to avoid altering component vibration characteristics by the attachment of instrumentation and wiring. Proper procedures and instruments are established by careful analysis of the test setup.

3. DESIGN CRITERIA and Recommended Practices

3.1 ACTUATOR CONFIGURATIONS

3.1.1 Piston Actuators

3.1.1.1 GENERAL CONSIDERATIONS

3.1.1.1.1 Pneumatic Modulating Control Dynamics

Pneumatic piston-cylinder actuators for servovalve control shall satisfy requirements for speed of response and load stiffness as well as requirements for system dynamic stability.

It is recommended that quantitative design requirements for a pneumatic piston-cylinder actuator for modulating control be established on the basis of nonlinear analysis of closed-loop system performance. Computer techniques similar to those described in references 14 through 16 should be employed in evaluating the effects of the critical nonlinear and discontinuous functions. Pistons should be designed so that the restrictor plenum compensation technique illustrated in figure 2 can be used if required by results of analysis or by hardware tests. Actuator seals and guides should be designed for control of static and dynamic friction forces within the limits established by system analysis. It is recommended that analysis and design procedures be planned to allow for a number of iterations in establishing design requirements, preparing preliminary designs, modifying the mathematical model as hardware designs evolve, revising the original design requirements for the various system components, revising preliminary designs, and so forth until all system and component design and performance problems have been redistributed and solved.

3.1.1.1.2 Hydraulic Modulating Control Dynamics

Hydraulic piston-cylinder actuators for servovalve control shall satisfy requirements for speed of response and load stiffness as well as requirements for system dynamic stability.

It is recommended that quantitative design requirements for an hydraulic actuator be established on the basis of a computerized nonlinear dynamic analysis of closed-loop system performance. Modulating actuators should be designed for continuous removal of trapped or fluid-entrained gases or vapors (sec. 3.1.1.1.4). Actuator seals and guides should be designed for control of static and dynamic friction forces within the limits established by system analysis. Firm actuator design requirements should not be established until there has been sufficient interaction between system analysis and preliminary design to distribute and solve system design and performance problems.

3.1.1.1.3 On-Off Control Dynamics

Actuator response timing shall not be affected adversely by differences in piston effective areas, actuator volumes, or swept volumes.

Hydraulic on-off actuators should be designed for continuous self-priming to minimize fluid-compressibility effects in obtaining repeatable response timing (sec. 3.1.1.1.4). Hydraulic-fluid flow-control restrictors must be sized to accommodate differences in piston effective areas and volumetric displacements on both sides of a double-acting piston.

For a pneumatic on-off actuator, with response-timing control required in both directions, it is essential that (1) the volume on each side of the piston, with the piston at each extreme of its displacement, (2) the piston effective areas and corresponding swept volumes and, (3) the flow-restrictor sizes for each side be adjusted so that the time for motion start in addition to the full-stroke motion time in each direction will satisfy the response requirements. Figures 26 and 27 (ref. 42) provide a convenient method for computing pneumatic-pressure response time in an actuator cavity with a response-control flow restrictor. A cavity volume, a restrictor effective flow area, and gas properties are used to compute a response-time constant τ. Figure 26 can then be used to relate cavity pressure to a step increase in signal pressure. Figure 27 can be used to relate cavity pressure to the initial cavity pressure in response to a step opening of a vent flow path. Response time is plotted in terms of the computed time constant.

The parameters involved in figures 26 and 27 are treated therein in customary units. In SI units,

$$\tau = \frac{V_2}{k\,CA\,\sqrt{RT_1}\quad S_{sonic}} \qquad \text{(fig. 26)}$$

$$\tau = \frac{V_2}{k\,CA\,\sqrt{R(T_2)_0}\quad S_{sonic}} \qquad \text{(fig. 27)}$$

where

τ = time constant, sec

V_2 = cavity volume, mm^3

k = ratio of specific heats

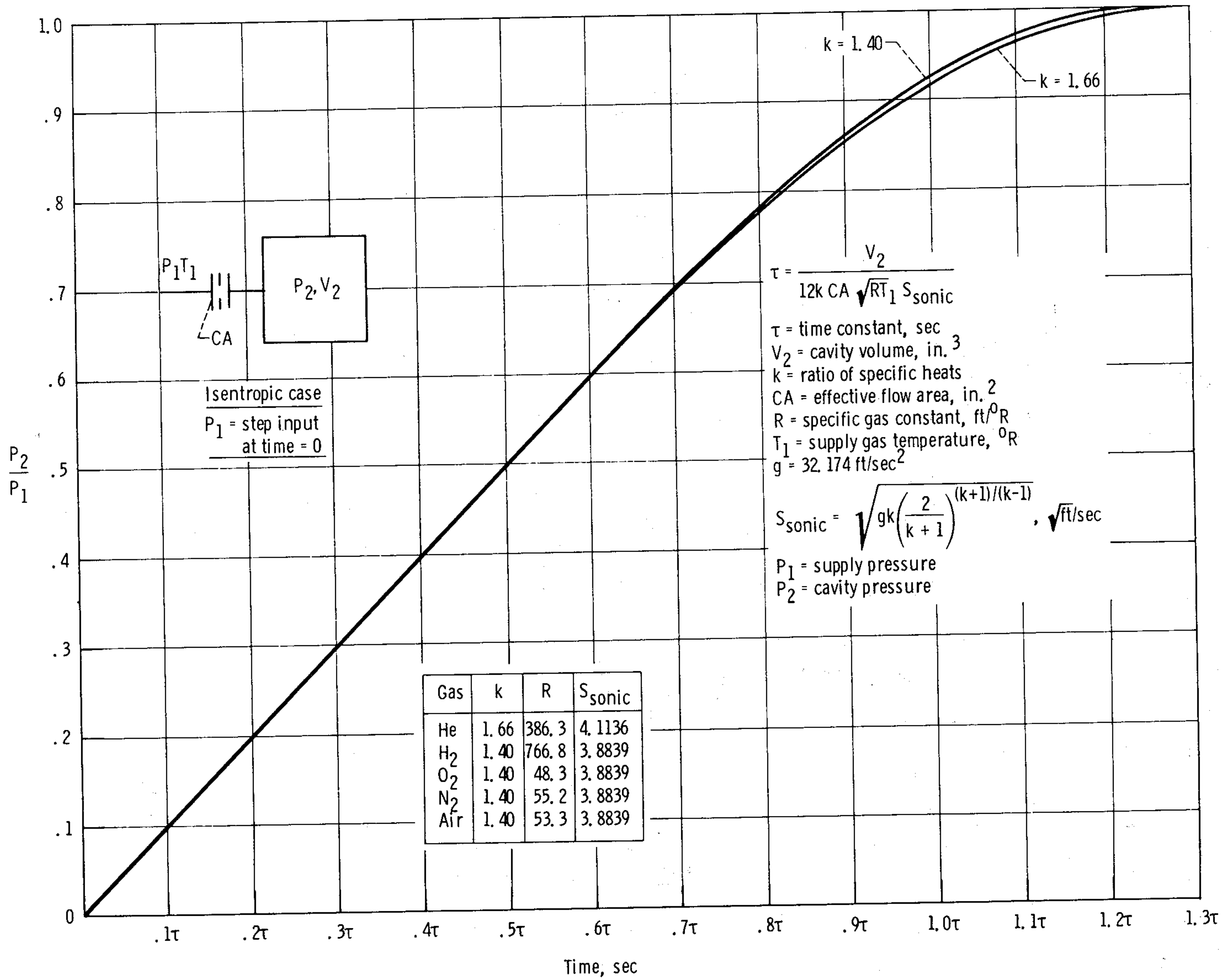


Figure 26. - Plot of gas pressurizing transients in a fixed volume (ref. 42).

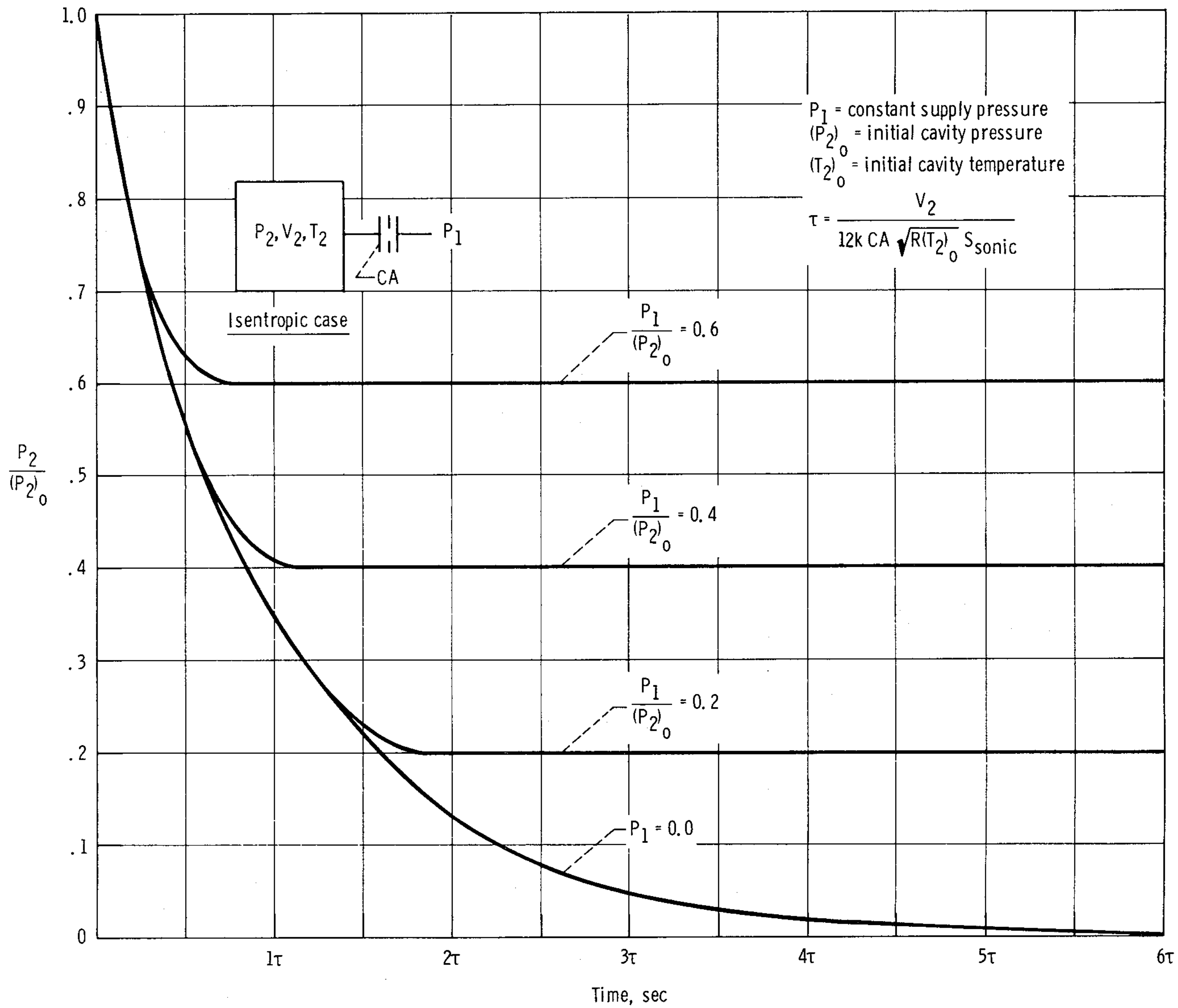


Figure 27. - Plot of gas venting transients in a fixed volume (ref. 42).

CA	=	effective flow area, mm^2
R	=	specific gas constant, N-m/K-kg
T_1	=	supply gas temperature, K
$(T_2)_o$	=	initial cavity temperature, K

$$S_{sonic} = \sqrt{k\left(\frac{2}{k+1}\right)^{(k+1)/(k-1)}} \quad \text{, dimensionless}$$

3.1.1.1.4 Hydraulic-Actuator Priming

Actuator dynamic response and actuation system stiffness shall not be affected adversely by trapped or fluid-entrained gases or vapors.

The most reliable technique for removing gases in an hydraulic actuation system, and for maintaining this primed condition, is to design the actuator for a controlled circulation flow of hydraulic fluid prior to and during operation. This design feature can ensure removal of gases initially in the system and removal of entrained gases and vapors released from the actuation fluid as the fluid pressure decreases from supply pressure to the actuator internal pressure during actuation. When efficiant self-priming is not feasible, design analysis and test results should verify that the response and stiffness meet requirements under all potential conditions of partial priming.

With a single-acting hydraulic piston actuator, continuous priming should be obtained with (1) a three-way valve operator that delivers hydraulic-system return pressure to the actuator cavity during standby and delivers supply pressure during operation and (2) a controlled-leakage flow path from the actuator cavity to a drain system that is at a lower pressure than the hydraulic-system recirculating return pressure. For a direct-acting actuator, the sensed-pressure source can be used for circulation flow if pressure is available prior to actuator operation. Avoiding depletion of the fluid supply is a primary consideration in designing single-acting actuators for continuous priming.

With a double-acting hydraulic piston actuator, continuous priming should be obtained by (1) using a four-way-valve operator that provides hydraulic system supply pressure to one actuation cavity and hydraulic system return pressure to the other cavity and (2) interconnecting the two actuation cavities with a controlled-leakage flow path. The inflow port and passages and the outflow path must be located to maintain a circulation flow that will remove gases with the actuator in any operating attitude. The actuator should be designed so that there are no cavities for gas entrapment at an elevation higher than that of any outflow path. Clearances in piston-cylinder seals and orifices or drilled holes for flow through a piston are convenient paths for priming flow. If an operational attitude for an actuator is such that gas may be trapped in a location that will not be primed by circulation flow through the piston, it may be necessary to add priming flow passages for outflow from those locations. Inflow passages may be located tangentially with respect to the cylinder wall to assist priming by creating a swirling circulation flow.

Exceptions to these practices occasionally are necessary for pressure-sensing actuators exposed to a system pressure that must be contained with little or no leakage. In this case, the entrapped-gas compressibility effects must be accommodated in the system design, usually by minimizing the actuator volume and displacement.

3.1.1.1.5 Hydraulic-Fluid Temperature Control

Actuator fluid temperature shall remain within allowable limits under standby as well as operational conditions.

The simplest combination of actuator materials, heat-transfer path, and fluid-circulation flow path that satisfies this requirement should be used. Electrical heaters or hydraulic heat exchangers requiring separate plumbing are alternates that should be considered only if simple actuator design concepts are not feasible.

Actuator-fluid temperature is controlled most simply by sizing the circulation flow resistances provided for actuator priming flow so that the priming flow, from a temperature-controlled source, is sufficient to prevent heat transfer from changing the fluid temperature beyond established limits.

When the circulation flowrate required for temperature control is large relative to the flowrate required for actuator displacement transients, parallel-circulation flow paths are recommended. The actuation fluid should flow through a timing-control flow restrictor, and the temperature-control fluid should flow through heat-exchanger passages in the actuator housing. Flow restrictors in a temperature-control flow path should be located where the greatest localized heat-transfer coefficients, accompanying localized high fluid velocities, are most beneficial in transferring heat energy to or from the hydraulic fluid.

3.1.1.1.6 Safety Requirements for Test and Operation

Test and operation of pressurized components shall comply with requirements for personnel safety as well as with those for structural integrity.

It is recommended that proof pressures be specified with a tolerance of ±2 percent. The fluid to be used, the test setup, the filtration and moisture-limiting requirements, and the test procedure should be specified in detail for protection of personnel as well as for preservation of hardware structural integrity. The proof pressure should be applied for a minimum time of 2 minutes and then reduced to zero at specified limiting rates of pressure change. Each proof-pressure test should be repeated for a total of five pressurizing and depressurizing cycles.

The safe operating pressure for personnel exposure should be a function of the pressurizing media, the pressures the component has previously experienced, and the component burst-pressure rating. If a safety-factor policy has not been established, the following practices are recommended. For components that may be pressurized with poisonous, corrosive, flammable, cryogenic, or compressible fluids and that are not proof-pressure tested, the safe operating pressure should not exceed 1/6 the calculated or demonstrated minimum burst pressure, whichever is lower. When such components have been proof-pressure tested, the safe operating pressure should not exceed 2/3 the proof pressure or 1/4 the calculated or demonstrated minimum burst pressure, whichever limit is lowest. For components that will be pressurized with nonhazardous liquids only, and that are not proof-pressure tested, the safe operating pressure should not exceed 1/4 the calculated or demonstrated minimum burst pressure, whichever is lower. When such components have been proof-pressure tested, the safe operating pressure should not exceed 2/3 the proof pressure or 2/5 the calculated or demonstrated minimum burst pressure, whichever limit is lowest.

Whenever feasible within design constraints, each spring-loaded actuator should be designed so that disassembly results in every spring with significant force returning to its free length with the assembly fasteners still engaged. This practice requires careful attention to fastener engagement lengths and to spring lengths. Shorter springs with greater spring rates frequently can be used to eliminate disassembly problems while obtaining the required installed load. When an assembly procedure requires the application of spring forces prior to engagement of spring-retaining features, with consequent disassembly hazard, a standard warning tag must be used. A durable metal warning tag should be attached to the actuator in a prominent location in a manner that minimizes the potential for inadvertent damage or removal.

3.1.1.2 FLOW RESTRICTORS

3.1.1.2.1 Flow Control

Flow restrictors shall satisfy actuator response requirements with the least possible testing and fitting.

When precise control of actuator response is required in an open-loop system, it is recommended that response-control flow restrictors (primarily orifices) be designed for installation with negligible leakage in parallel with the primary flow paths through the restrictors. The use of flow-test procedures to size orifices is recommended only when the orifice configuration is not amenable to control of the effective flow area by precise control of dimensions. The use of sharp-edged orifices is recommended for ease of control of orifice entry geometry and attendant flow coefficients. The dimensional tolerances and surface finishes should be established with sufficient precision to eliminate the need for orifice flow testing and cut-and-try sizing. It is recommended that the smoothness of surface finish specified for the orifice bore and the upstream face be consistent with the desired sharpness of the entry. These recommendations are illustrated in figure 28 for a response-control orifice that is designed for symmetry to eliminate potential assembly error.

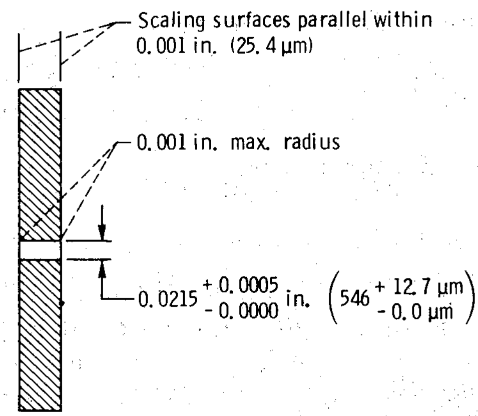


Figure 28. - Recommended tolerances for a precision orifice.

As discussed in section 2.2.1.1, if cut-and-try procedures are to be eliminated in satisfying actuator response requirements, it is essential that pilot valve effective flow areas and plumbing line sizes be great enough to concentrate system flow resistances at the response-control orifices.

When an orifice must be flow tested to obtain and verify the desired effective flow area, the test setup of figure 29 is recommended. When the same test setup is used for checking all orifice test specimens, the effective flow area of a specimen is proportional to the ratio of P2 to P1 for pneumatic flow, or is proportional to the square root of the ratio of (P2-P3) to (P1-P2) for hydraulic flow. With this setup, pressure P1 is regulated at a specified nominal

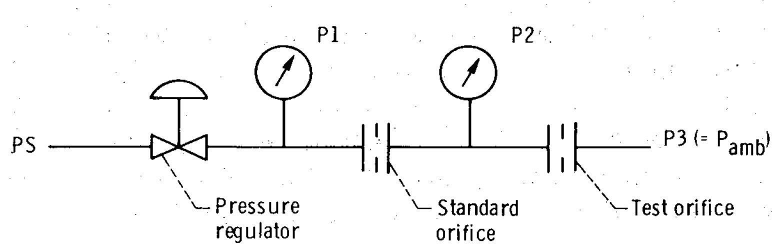


Figure 29. - Schematic of setup for orifice flow test.

value, and the effective flow area of a test specimen is compared with the effective flow area of an orifice used as a comparison standard. The test requirements can be based solely on two pressure measurements and the appropriate ratios, eliminating the need for flow measurements with their inherent limitations on accuracy.

It is recommended that precision orifices be finished by precision lapping procedures for the bores and faces. With orifice-entry edges controlled by the intersection of two lapped surfaces, deburring of the orifice entry is not necessary, and precise control of entry geometry is obtained. To prevent mechanical damage and to obtain the required degree of inspection accuracy, precision orifices should be inspected by high-power microscope only; the use of go/no-go pins and other mechanical inspection techniques should be prohibited by drawing-controlled specifications.

3.1.1.2.2 Temperature-Effect Compensation

Actuator dynamic response as a function of fluid flow shall remain within allowable limits throughout the full range of fluid and environmental temperatures.

When changes in fluid temperature affect the flow through a fixed flow restrictor to the extent that actuator dynamic response fails to comply with response requirements, a restrictor with temperature-effect compensation should be considered as an alternate to a fixed restrictor. If a design concept like that of figure 5 is used, obtain nearly linear compensation throughout the temperature range by careful apportionment of the parallel flowrates through the fixed and variable restrictions under room-temperature conditions.

3.1.1.2.3 Contamination

Actuator dynamic response shall not be affected adversely by flow-restrictor contamination.

In hydraulic systems, wherever a small-size, response-control flow restrictor is used, a strainer or filter should always be located to protect the restrictor against contamination resulting from fluid inflow. If the hydraulic actuator is designed to minimize internal contaminant generation and is assembled under clean conditions, filtration of outflow is not required. This practice applies when an orifice diameter is 0.070 in. (1.78 mm) or smaller; for a needle valve, it applies when the radial clearance between the needle and the valve seat is 0.070 in. (1.78 mm) or smaller. A filter located near a flow restrictor must be shielded from direct fluid-jet impingement against the filter element to preclude filter-element damage when flow is in the direction of the restrictor to the filter.

In pneumatic systems, the use of individual filters for critical flow restrictors requires evaluation of each particular case to determine whether a filter will enhance or detract from restrictor reliability. In debatable situations, the design should include provision for flow-restrictor filters that can be retained or deleted as test and operating experience is accumulated.

A strainer or filter and a timing-control flow restrictor incorporated in an actuator assembly is the preferred configuration because it protects the assembly against inadvertent contamination or inadvertent overspeed actuation during component checkout. A filter should be sized for low pressure drop and for adequate capacity to retain contaminants, so that changes in filter flow resistance will have only minor effect on overall flow resistance.

3.1.1.3 SPRINGS

3.1.1.3.1 General Requirements

A mechanical spring and its installation shall satisfy actuator reliability requirements as well as spring mechanical design requirements.

The use of mechanical springs in piston actuators should be avoided whenever feasible. When springs are required, it is recommended that all helical springs be designed for solid-height stresses within the allowable stress limits for the spring materials; this practice will preclude spring deformation under vibration conditions in which coil clashing may occur. Stress limits must allow for the maximum spring temperature that maybe occur under all test as well as operational conditions. Potential vibration-induced deflections should be minimized by designing each spring so that the deflection from free length to minimum working length is approximately 80 percent of the deflection from free length to solid height.

3.1.1.3.2 Generation of Metal Particles

A mechanical spring and its retention features shall not generate metal particles.

Clearances between springs and adjacent parts should be designed to preclude contact resulting from acceleration and vibration deflections as well as from distortion deflections that accompany spring installation and piston stroking. Details of spring end-coil design should be specified explicitly to eliminate sharp edges and to control the thickness and structural integrity of end-coil tips. It is recommended further that helical-spring end coils and end-coil retainers be designed to minimize potential lateral displacement and consequent relative motion. Diametral clearance between a spring end coil and its retainer, required to accommodate coil-diameter forming tolerances, can be reduced by machining end-coil inside or outside diameter to closer tolerances than the forming tolerances. Additionally, it is recommended that an end-coil retainer for a helical spring be fabricated from relatively hard metals only. Spring retainer configurations can be machined directly into a steel piston or housing. When a relatively soft metal such as aluminum is used for a piston housing, separate steel spring retainers, restrained by pins or other mechanical interlock against motion relative to the piston or the housing, should be used.

3.1.1.3.3 Helical-Spring Rotation

Helical-spring angular displacement that accompanies axial deflection shall not produce adverse effects.

Helical-spring rotation that occurs as the spring is compressed to its installed length during assembly of an actuator should be accommodated by one of three methods: (1) allow the piston to rotate; (2) use assembly fixtures that permit spring rotation; or (3) use a thrust bearing in compression under a spring end coil. Coil rotation that accompanies a long working stroke should be accommodated by a design that allows piston rotation or by the use of a thrust bearing under one end coil. The preferred location for a thrust bearing is at the fixed end of a helical spring rather than at the moving piston end. Coil rotation accompanying compression of adjacent coils into contact with each other (coil clash) as a result of acceleration or vibration should be minimized by designing springs as recommended in section 3.1.3.1.

3.1.1.4 MATERIALS

3.1.1.4.1 General

Material structural integrity shall be maintained throughout the storage and service life of an actuator.

Materials known to be susceptible to stress-corrosion cracking (table I) should not be used when the expected environment is conducive to that failure mode. Parts made from aluminum alloys should be chromic- or sulphuric-acid anodized to enhance corrosion resistance. Surface finishes of 63 μ in. (1.60 μm) AA, or smoother, should be specified for all machined surfaces. Materials selections for specific types of actuator and operator elements are discussed under separate topic headings for those elements.

3.1.1.4.2 Helical-Spring Material

Helical spring material shall be compatible with the environments to which it will be exposed.

For maximum reliability in space-flight hardware, materials for helical springs should be limited to the corrosion-resistant alloys.

When design analysis reveals temperature-related spring-force or spring-rate variations resulting in failure to comply with actuator performance requirements, the use of a Ni-Span-C helical spring, heat treated to minimize temperature-related variations in its spring rate, is recommended. When Ni-Span-C is used, the solid-height maximum fiber stress should be limited to the range of 60,000 to 80,000 psi (414 to 552 MN/m^2), with 60,000 psi (414 MN/m^2) as the limit in applications where spring vibration is anticipated and shift in installed load is a critical design consideration. The use of Ni-Span-C or similar spring materials to minimize temperature-related variations in mechanical spring rate requires strict quality control procedures to verify the effectiveness of the material heat treatment. Because the allowable stresses for Ni-Span-C are appreciably lower than the allowable stresses for other commonly used spring materials, greater installation space must be provided for Ni-Span-C springs; this space requirement should be anticipated early in the design activity.

When differential expansion or contraction of actuator component parts results in excessive variation in the installed length of a reference spring, the temperature-effect compensation techniques of section 3.1.2.3.2 are recommended.

It is recommended that the use of 17-4PH and 17-7PH for springs be discontinued because these materials are susceptible to long-term stress corrosion.

3.1.2 Diaphragm Actuators

3.1.2.1 GENERAL CONSIDERATIONS

3.1.2.1.1 Fluid-Inflow Effects

Diaphragm structural integrity shall be maintained under all conditions of fluid inflow.

It is recommended that pressurized cavities and inflow porting associated with a diaphragm be designed so that high-velocity fluid inflow during initial pressurization or during operation does not impinge directly on the diaphragm. In designs requiring small clearances between a movable diaphragm surface and fixed surfaces, the fluid inflow should be filtered to preclude damage resulting from trapping solid-particle contaminants between the diaphragm and the fixed surfaces. When test or operational conditions introduce the potential for ice formation in the clearances between a movable diaphragm and adjacent fixed surfaces, the clearances should be designed to accommodate this form of solid-particle contamination, and system operational specifications should be established and implemented to minimize this problem.

3.1.2.1.2 Vibration

Diaphragm cycle-life endurance capability shall not be degraded by vibration-induced dynamic oscillations.

It is recommended that diaphragm reference springs and their installations, especially for modulating diaphragms, be designed to eliminate resonance modes under vibration conditions. Diaphragms should be isolated from fluctuating driving forces associated with sensed-pressure dynamics. It is recommended that modulating diaphragm actuators be designed for dashpot damping, using the diaphragm as the dashpot piston and sizing the associated volume and orifice for the maximum damping consistent with the required speed of response.

3.1.2.1.3 Installation Mismatch

Diaphragm cycle-life endurance capability shall not be degraded by mismatch of diaphragm elements with contacting surfaces.

Alignments and concentricities of all parts that affect installation of a diaphragm should be obtained by controlling tolerances on dimensions and by designing for self-alignment of parts that contact each other. To obtain concentricity of inner and outer clamping surfaces, take into account the diametral clearance between the diaphragm piston rod and its guides. The assembly procedure, and the use of any required tools and fixtures for the positioning of parts, should be formally specified and controlled. When nonmetallic diaphragms are used, the use of initially flat diaphragms of pliable plastic materials, with convolutions formed in place by the application of controlled pressures following installation, should be considered.

3.1.2.1.4 Interply Pressures

Actuation pressures shall not result in interply pressurization of a multiple-ply diaphragm.

Actuators with multiple-ply diaphragms should be designed to preclude exposure of interply voids to applied pressures. When multiple-ply diaphragms are retained by clamping, diaphragm inner and outer edges should be vented to ambient pressure. In mechanical design, utilize the basic concepts illustrated in figure 6. Seal welding of metal diaphragms, alternate diaphragm clamping features, or supplementary static seals may be used to advantage, the choice depending on specific actuator configuration requirements.

3.1.2.1.5 Convolution Reversals

A diaphragm actuator shall withstand the maximum anticipated differential-pressure reversal without loss of required cycle-life endurance capability.

The stiffness of metal diaphragm convolutions may be sufficient to resist deformation when exposed to low-level differential-pressure reversals. A factor of 1.5 should be applied to the anticipated differential-pressure reversal for purposes of analysis or test. Each diaphragm-actuator application must be analyzed carefully to determine the maximum potential diaphragm differential-pressure reversal, under transient and surge pressure conditions, as well as under quiescent test and operating conditions, so that the actuator can be designed to accommodate the most severe differential-pressure reversal that may be experienced in service. The use of opposing diaphragms with a vented intermediate space, as illustrated in figure 6(b), eliminates the potential for diaphragm reversal resulting from applied differential-pressure reversals.

3.1.2.1.6 Overstroking

A diaphragm shall not be subject to overstroking and overstressing in any phase of an assembly process.

If an assembly process requires a diaphragm-actuator subassembly that at any time can be separated from stroke-limiting features in the complete assembly, the actuator subassembly should have mechanical stops to limit potential diaphragm axial displacement to no more than 90 percent of its calculated maximum allowable displacement. The maximum allowable axial displacement of a diaphragm from its neutral position can be used most efficiently if the diaphragm is installed in an actuator assembly so that its neutral position corresponds to the midstroke position of the actuation mechanism. Under these conditions, the working stroke of the diaphragm will be one-half the full stroke of the actuation mechanism. Stated conversely, the full stroke of the actuation mechanism can be two times as great as the allowable axial displacement of the diaphragm. For a given diaphragm design, such installation minimizes diaphragm deflection stresses and provides the greatest allowance for dimensional tolerances related to axial positioning of the diaphragm when an actuator is assembled.

3.1.2.2 DIAPHRAGM ELEMENTS

3.1.2.2.1 General Requirements

A diaphragm and its installation shall comply with actuator reliability requirements as well as with diaphragm mechanical design requirements.

Convolution annular width, depth, blend radii, and the material thickness must be designed to accommodate applied pressures and axial deflections within the material stress limitations. Retainer corners adjacent to diaphragms should be smoothly rounded with a minimum radius dimension at least double the diaphragm thickness. Retainer radii should be smaller than mating diaphragm-convolution blend radii to minimize the potential for mechanical interference. Convolution annular widths must be large enough, and material thicknesses must be small enough, to preclude convolution wrinkling during forming or as a result of axial displacement.

Tests of representative diaphragms, as installed in an actuator assembly, should verify conformance with design requirements for mechanical spring rate, deflection, and applied pressures; and life-cycle tests and vibration tests should verify reliable conformance with durability requirements.

3.1.2.2.2 Interply and Interface Contamination

Fabrication, processing, and installation of diaphragm elements shall minimize the potential for failure resulting from interply or interface contaminants.

The use of single-ply diaphragm elements provides the most direct approach to eliminating diaphragm contamination problems. When multiple-ply diaphragm elements are used, they should be formed and maintained as matched sets; and formal procedures for forming, processing, and installing should be specified and rigorously controlled so that diaphragm interply cleanliness is maintained. With single-ply or multiple-ply diaphragms, effective process controls are required in obtaining cleanliness at interfaces between diaphragms and contacting parts. When a metal diaphragm, single-ply or multiple-ply, is welded to a retainer, the interface between the diaphragm and the retainer should be designed to minimize the potential for interface contamination during welding; effective process controls should be implemented to obtain and maintain interface cleanliness.

3.1.2.2.3 Diaphragm Forming

3.1.2.2.3.1 Die Forming

Dies and forming procedures for preformed diaphragms shall control diaphragm geometry within acceptable limits.

Forming dies should be designed with an allowance for material springback when forming forces are removed. In all cases, representative samples of diaphragms formed with a particular die design should be precisely inspected and tested to prove conformance with design requirements. Inspection of diaphragm proof samples may indicate consistent production of diaphragms with acceptable dimensions. If those dimensions differ from the dimensions originally specified, as a result of material springback, minor revision of dimensional specifications for parts that clamp the diaphragm may be required. The need for such minor revisions should be anticipated in a design program when a new preformed-diaphragm design is to be used, and springback cannot be predicted precisely. Rigorous production controls are required to ensure that all preformed diaphragms used in actuator assemblies have been formed with proven dies and that no changes in the dies, the diaphragm material, or the forming process occur without a repetition of the die-proving procedure. To supplement the production controls, sample diaphragms from each production lot should be inspected dimensionally and burst tested.

3.1.2.2.3.2 Forming in Place

A formed-in-place diaphragm shall withstand actuator operating pressure.

When a diaphragm is to be formed in place, the initial application of pressure should be a forming pressure that is greater than the maximum anticipated operating pressure, including surge pressures. The forming pressure should be equal to or greater than the required proof pressure for the actuator. It is recommended that the forming pressure be applied gradually, maintained for several minutes, and then removed gradually. This cycle should be repeated several times to ensure complete forming (sec. 2.1.1.1.6). Diaphragms thus formed should never be reused. Formal specifications should require the installation of new diaphragms and a repetition of the forming-pressure cycles any time an actuator has been disassembled. If a diaphragm is to be formed at room temperature, the specified forming pressure should take into account the material properties at maximum operating temperature and maximum operating pressure. The diaphragms must be designed to preclude excessive yielding and localized thinning under the forming conditions, due account being given the actuator stroke as well as the forming pressure. Development test effort in establishing the optimum material thickness should be anticipated for new designs that are significantly different from proven designs.

3.1.2.3 SPRINGS

3.1.2.3.1 Spring-Force Control

Lateral components of spring forces applied to a diaphragm shall remain within acceptable limits throughout the required cycle life.

Actuator design details should be carefully controlled to ensure uniform application of axial spring forces with respect to a diaphragm axis. Lateral spring forces applied to a diaphragm should be minimized, and a diaphragm should have adequate support against lateral deflection. When helical-spring lateral forces resulting from nonparallel spring ends in the installed condition can be transmitted to a diaphragm, springs should be fabricated for parallelism of the end coils at the installed compressed height. It is recommended that all drawings for helical compression springs be supplemented by a specification document that establishes spring classifications, design requirements for each classification, and quality control requirements for obtaining spring precision commensurate with specific application requirements.

3.1.2.3.2 Temperature-Effect Compensation

A diaphragm actuator shall function satisfactorily throughout the range of temperatures, temperature gradients, and temperature transients to which it is exposed.

When actuation at a precise pressure setting is required and when compensation for temperature effects is necessary, conduct a thorough analysis of environmental conditions and of transient and steady-state temperature gradients throughout the actuator assembly. The design techniques discussed previously are recommended in preference to the more elaborate techniques described in references 64, 65, and 75, unless design complexity can be justified in specific design situations.

3.1.2.4 MATERIALS

Diaphragm materials shall not be affected adversely by installation, storage, test, and operational conditions.

Factors that must be considered in selection of material include heating during welding, potential galvanic corrosion, stress corrosion, stress relaxation, thermal stability, cyclic stresses, and chemical compatibility. Cold working during forming increases the strength of some metals such as the 300-series CRES alloys, but if cold-workable diaphragms are to be welded to retainers, they should be stress relief annealed to obtain diaphragms with predictable and repeatable properties. It is recommended that the use of 17-7PH for diaphragms be discontinued because of its susceptibility to long-term stress corrosion. Hydrogen embrittlement of 17-7PH, and consequent drastic reduction in cycle life, eliminates this material from consideration when exposure to hydrogen is required. Inconel 718 is recommended for applications requiring high strength or extensive cycling. The use of nonmetal diaphragms should be limited to applications in which very low mechanical spring rates are necessary and excellent cleanliness control is assured. Materials for nonmetal diaphragms should be selected for reliable quality control as well as for physical properties.

3.1.3 Bellows Actuators

3.1.3.1 GENERAL CONSIDERATIONS

3.1.3.1.1 Fluid-Inflow Effects

Bellows actuator structural integrity shall be maintained under all conditions of fluid flow.

It is recommended that pressurized cavities and inflow porting associated with a bellows element be designed so that fluid inflow does not impinge directly on the bellows. In designs requiring small clearances between convolutions, fluid inflow should be filtered to preclude damage resulting from trapping solid-particle contaminants between convolutions. When test or operational conditions introduce the potential for ice formation between convolutions, generous clearances between convolutions should be provided, and system operational specifications should be established and implemented to minimize any possibility of ice formation.

3.1.3.1.2 Fluid-Flow Dynamic Forces

Bellows cycle-life endurance capability shall not be degraded by fluid-flow dynamic forces.

When an actuator design requires exposure of a bellows element to fluid flow that may apply continuing dynamic forces to the bellows, the actuator should be designed to attenuate those forces. In a pressure-actuated valve assembly with a bellows exposed to the valve effluent, it is recommended that the bellows be located in a semistagnant region out of a turbulent flow path or located in an enclosure or shield sleeve that creates a semistagnant environment for the bellows. It is recommended that a pressure-sensing bellows be located in an isolating cavity with an orifice or small port for snubbing sensed-pressure fluctuations.

3.1.3.1.3 Fatigue Life

Bellows elements shall satisfy full-stroke cycle-life endurance requirements and vibration endurance requirements.

This criterion is satisfied best by designing a bellows so that its natural frequencies are above the range of the driving frequencies with significant energy. If this technique is not feasible, as a result of design constraints, it is essential that the actuator and its bellows element be designed for vibration damping. Vibration damping can be obtained by the use of an articulated bellows, using the dividing rings or guides as friction dampers or as viscous damping vanes. Articulated bellows guides with controlled guide-to-cylinder clearances for fluid flow, and with orifices through the guides if necessary for flow control, can be used to introduce vibration damping forces opposing guide velocity but independent of steady-state displacement, as in the case of fluid dashpots. In designing for full-stroke cycle-life endurance requirements, generous allowance should be made for factors that can reduce the fatigue life of a particular bellows element, e.g., localized convolution deformations incurred during fabrication, processing, or installation. When extended cycle-life endurance is required, especially under conditions of severe vibration, the recommendations in sections 3.1.3.2.2 and 3.1.3.2.3 for the use of single-ply formed bellows are emphasized.

3.1.3.1.4 Overstroking

Bellows elements shall not be subject to overstroking and overstressing in any phase of an assembly process.

If an assembly process requires the use of a bellows-actuator subassembly that at any time can be separated from stroke-limiting features in the complete assembly, the actuator subassembly should contain mechanical stops that limit potential bellows axial displacement to no more than 90 percent of the calculated maximum allowable displacement. The maximum allowable axial displacement of a formed bellows from its neutral position can be used most efficiently if the bellows is installed in an actuator such that its neutral position corresponds to the midstroke position of the actuation mechanism. Under these conditions, the working stroke of the actuator will be one-half the full stroke of the actuation mechanism. Stated conversely, full stroke of the actuation mechanism can be two times as great as the allowable axial displacement of the bellows from its neutral position. For a given formed-bellows design, such installation minimizes bellows deflection stresses and provides the greatest allowance for dimensional tolerances related to axial positioning of the bellows when an actuator is assembled. When bellows with corrugations welded to each other are used, displacement in extension from the neutral position should always be avoided. Bellows handling and installation procedures should be specified and controlled to protect bellows elements and to prevent installation of damaged elements.

3.1.3.2 BELLOWS ELEMENTS

3.1.3.2.1 General Requirements

> *Bellows characteristics and bellows installation shall satisfy actuator reliability requirements as well as bellows mechanical design requirements.*

General design practices for bellows should be supplemented with the specific design criteria and recommended practices presented in section 3.1.3.1.

Tests of representative bellows, as installed in an actuator assembly, should verify conformance with design requirements for mechanical spring rate, deflection, and applied pressures; and life-cycle tests and vibration tests should verify conformance with durability requirements.

3.1.3.2.2 Interply Contamination

> *Fabrication, processing, and installation of bellows elements shall minimize the potential for failure resulting from interply contaminants.*

This criterion is satisfied best by designing actuators for the use of single-ply bellows elements. When the use of multiple-ply bellows elements is essential, on the basis of objective and critical analysis of actuator design requirements, formal procedures for ensuring interply cleanliness during bellows element fabrication and welding to end fittings should be specified and rigorously controlled. When multiple-ply bellows are heli-arc fusion-welded to an end fitting, the plies should be resistance-welded together at the bellows ends prior to fusion welding to the end fitting in order to prevent entry of gases or contaminants between the plies.

3.1.3.2.3 Fabrication Effects

The structural characteristics and endurance capabilities of bellows elements shall not be degraded by fabrication effects.

This criterion is satisfied best by designing actuators for the use of formed bellows elements. When the use of bellows elements with corrugations welded to each other is essential, on the basis of actuator size or mechanical spring rate limitations, welding process procedures should be carefully specified and rigorously controlled.

When a bellows is made from heat-treatable metal alloys, the bellows should be welded to compatible non-heat-treatable end fittings prior to heat treatment. When bellows end fittings are welded to an actuator housing or to other actuator elements, bellows heating should always be avoided because localized heating may alter the bellows material properties.

When bellows with welded corrugations are used, actuators should be designed for essentially all bellows operation in compression only.

It is recommended that all welding procedures include cleanliness control procedures as a safeguard against potential corrosion. For minimizing corrosion potential, welding at end fittings is recommended in preference to soldering or brazing.

When exposure to reactive fluids is required, actuators should be designed so that crevices at bellows end-fitting joints are not exposed to the fluids.

3.1.3.2.4 Bellows Squirm

The actuator configuration shall preclude the occurrence of structural failures resulting from bellows squirm or column buckling.

It is recommended that actuators be designed so that the maximum pressure difference to which a bellows element is exposed is applied with the higher pressure external to the bellows element. The actuator design should provide sufficient guiding of movable bellows ends to preclude lateral offset to an extent that can appreciably reduce the resistance to buckling. When squirm or buckling problems with simple bellows elements cannot be avoided, the use of articulated bellows elements (fig. 9) is recommended.

3.1.3.2.5 Quality Control

The quality of bellows elements and assemblies shall be consistently acceptable.

It is recommended that bellows elements, and parts to which they are joined, be designed with accessibility for inspection as a primary consideration. When inspection for cleanliness or defects is difficult or impossible, as in bellows with welded corrugations or multiple plies, fabrication processing must be adequately controlled for quality assurance. Formal packaging, handling, test, and installation procedures should be specified and implemented to prevent installation of a defective bellows in a flight actuator.

3.1.3.3 BELLOWS SPRING CHARACTERISTICS

A bellows actuator shall function satisfactorily throughout the range of temperatures, temperature gradients, and temperature transients to which it is exposed.

Bellows elements should be designed for structural integrity throughout the temperature range to which they are exposed, and it should be demonstrated that temperature-related variations in mechanical spring force and spring rate do not adversely affect actuator performance. When analysis or test reveals temperature-related spring-force or spring-rate variations that result in failure of the actuator to comply with performance requirements, the temperature-effect compensation techniques discussed in section 3.1.2.3.2 should be employed.

3.1.3.4 BELLOWS MATERIALS

Bellows materials shall not be affected adversely by installation, storage, test, and operational conditions.

Factors that must be considered in the choice of material include heating during welding, potential galvanic corrosion, stress corrosion, stress relaxation, thermal stability, chemical compatibility, and cyclic stresses. The use of stress-relief-annealed bellows elements is recommended for minimizing potential weld-heating effects and for long-term retention of the as-fabricated material properties. It is recommended that the use of 17-7PH for bellows be discontinued because of its susceptibility to long-term stress corrosion. Hydrogen embrittlement of 17-7PH, and consequent drastic reduction in its cycle-life capability, eliminates this material from consideration when exposure to hydrogen is required. Inconel 718 or its equivalent is a good choice for most applications.

3.1.4 Solenoid and Torque-Motor Actuators

3.1.4.1 ELECTROMAGNETIC CIRCUITS

3.1.4.1.1 Materials

Electromagnetic-circuit materials shall be compatible with fluids or vapors to which they may be exposed.

It is recommended that the design of electromagnetic circuits call for corrosion-resistant ferromagnetic materials that do not require protective platings. If plating can be justified for parts requiring special consideration in a circuit, design allowances must be made for nonuniform plating thicknesses at internal and external corners or at other surface discontinuities. Smooth machined finishes on exposed surfaces should be specified to enhance corrosion resistance. Surfaces exposed to reactive fluids should be designed to eliminate fissures or fluid traps that impair flushing or degrade corrosion resistance.

3.1.4.1.2 Eddy Currents

Eddy currents in a solenoid actuator shall not degrade speed of response.

When high-speed response is required, it is recommended that the center of the coil be located as close to the working airgap as is structurally feasible. This location reduces the time lag between a change in coil field strength and the resulting change in airgap flux, the lag being due to induced eddy currents in the ferromagnetic material in the intervening flux path. When high-speed response is required, and chemical-compatibility considerations require the use of a ferromagnetic material that has a relatively high electrical conductivity, use slotted parts in the flux path. The slots should be located to parallel the direction of the lines of flux and thus introduce electrical resistance to induced eddy currents in the transverse direction.

3.1.4.1.3 Residual Magnetism

Residual magnetism in a solenoid actuator shall not adversely affect deenergize response.

As a general practice, it is recommended that each solenoid actuator in a nominal 28-volt system be designed for positive deactuation, subsequent to actuation, at an impressed voltage equal to or greater than 2 volts, as the impressed voltage is gradually reduced from its maximum limit. The maximum limit on impressed voltage is the maximum steady-state value resulting from voltage tolerances in the system power supply. If actuator working-load forces that are not always present (e.g., fluid pressure forces) assist deactuation, they should not be applied in tests intended to verify deactuation capability; if actuator working-load forces retard deactuation, they should be applied. This practice ensures positive deactuation, subsequent to actuation, under checkout as well as operating conditions, in the presence of maximum residual magnetism and with an allowance for signal-circuit leakage current in the deenergized mode. The 2-volt deactuation limit is a rule-of-thumb value for 28-volt space-flight solenoid actuators and is subject to revision on the basis of design requirements for specific actuator and signal circuits. Systems operated at other voltage levels should be analyzed to determine appropriate specification dropout levels.

3.1.4.1.4 External Magnetic Fields

The strength of the external magnetic field produced by an electromagnetic actuator shall be limited as required by system specifications.

Limits on the external magnetic field should be considered only to the extent that performance of other components is affected. In situations wherein external-field limits are specified, high-permeability enclosures of ferromagnetic materials may be required for magnetic shielding. Coil voltage-surge suppression may be required for limiting transient field strength. It is recommended that the need for iterative design and development effort be minimized by an analysis of the magnetic field supported by comparison with existing designs for which field-strength measurements have been obtained.

3.1.4.2 ELECTRICAL CIRCUITS

3.1.4.2.1 Voltage-Surge Suppression

Circuit designs to suppress deenergize voltage surge in solenoid actuators shall not adversely affect functional or structural integrity of production units.

Incorporation of surge-suppression circuits in the solenoid actuator assembly is not recommended for space-flight hardware. Any surge-suppression elements in an actuator assembly must be capable of withstanding not only the operational temperature range, but also the temperatures to which the elements will be subjected during the fabrication of the coil assembly and welding for hermetic sealing. Capability for verifying coil insulation resistance with high-voltage tests should be maintained. Surge-suppression design utilizing diodes that require a polarity-sensitive coil assembly should not be used. If voltage-surge suppression is required for protection of driving-circuit elements, it is recommended that surge-suppression features be incorporated in the driving circuit.

3.1.4.2.2 Driving Circuit

The combination of solenoid actuator and driving circuit shall provide the required dynamic response.

It is recommended that the required response characteristics for a solenoid actuator be specified in terms of response to voltage step inputs. For design analysis and for testing, determining the response to step inputs is convenient and provides the most meaningful measure of response. The response to voltages throughout the tolerance range for applied voltages should be determined, so that nonlinearities may be identified. Requirements for the response of a solenoid actuator to step inputs should be based on the response of the driving circuit to its inputs. Several iterations may be required in specifying actuator and driving circuit individual requirements, performing design analyses and preparing preliminary designs, redefining individual requirements, and so forth in arriving at a successful hardware design.

3.1.4.3 COILS

3.1.4.3.1 Corrosion Protection

Solenoid and torque-motor coils shall not be subject to chemical or galvanic corrosion.

The most positive method for obtaining coil corrosion resistance is to enclose the coil in a compartment, remove the air in the process of drying the compartment, fill it with an inert gas, and seal it hermetically. True hermetic sealing requires the use of an electrical connector in which contact-pin insulation is fused to the pin and to the connector shell, and all joints must be welded or brazed. It is essential that the design specifications include detailed specification of the procedures for drying, purging, gas filling, and completion of the seal welding. Because the results of these critical procedures cannot be determined by inspection of a completed assembly, formal control of the procedures is required.

3.1.4.3.2 Coil-Wire Insulation

3.1.4.3.2.1 Temperature Limitations

Coil insulation materials shall be suitable for the coil temperature range.

It is recommended that coil-wire insulation materials for all applications be limited to non-reactive plastics with known temperature ratings. The actuator assembly should be tested to verify the adequacy of the insulation material in withstanding localized heating under all operating conditions.

3.1.4.3.2.2 Resistance and Dielectric Strength

Coil insulation in a complete coil assembly shall withstand the maximum anticipated transient voltage without degradation.

It is recommended that all elements of a solenoid coil assembly be designed for, and acceptance tested with, conditions at least as severe as those discussed in section 2.1.4.3.2.2. It is recommended further that each material thickness specification take into account the thickness tolerances and the thinning effects resulting from coil-wire tension during coil winding. Single-film-thickness coil-wire insulation should not be used unless such use can be justified by limitations on coil space. Dielectric-strength test specifications should limit the test duration to the minimum time required for efficient performance of these tests. It is recommended that dielectric-strength tests be performed only during acceptance testing of new or overhauled components. All checkout and test procedures in which coil assemblies are involved should be controlled carefully to prevent needless repetition of dielectric-strength tests.

3.1.4.3.3 Temperature-Effect Compensation

A coil assembly shall provide sufficient ampere-turns for operation at maximum coil temperature and sufficient resistance to prevent excessive current drain at minimum coil temperature.

It is recommended that the coil design use only conventional copper magnet wire if considerations of coil temperature, available space, and required response time permit. The use of constantan wire in series with copper wire is recommended when the required temperature range results in excessive current variation in an all-copper-wire coil.

3.1.4.3.4 Coil-Wire Extensions

Coil-wire extensions for attachment to an electrical connector shall withstand vibration-induced flexing.

For coil-wire diameters smaller than 0.010 in. (254 μm), extension wires with diameters of at least 0.010 in. (254 μm) should be welded to the coil-wire ends; exposed wire in the vicinity of the welds should be covered by insulation sleeving; and the insulated welds should be firmly secured to the coil by wrappings or bindings in a manner that restrains any potential motion of the smaller coil wire. Coil-wire extensions from the coil to the solder joints at the electrical-connector pins should be enclosed in insulation sleeving, or should be surrounded by potting compound, or should be otherwise supported and mechanically restrained to prevent vibration-induced flexing of the extension wires. Procedures should ensure that insulation at wire ends is stripped by thermal techniques only. When multistrand extension wires are used, for greater wire flexibility, procedures should preclude use of wires with damaged strands at exposed wire ends and ensure that attachments are made with all strands active.

3.1.4.4 ELECTRICAL CONNECTORS

3.1.4.4.1 Construction and Materials

Electrical connectors for solenoid actuators shall satisfy structural requirements as well as system interface requirements.

An electrical-connector weld joint should be as far as possible from the connector-pin insulation material. The connector-to-housing adapter section should be designed to permit attachment of a metal mass to the connector as a heat sink during welding so that temperatures and temperature gradients at the pin insulation are minimized. The actuator and the connector should be designed to minimize heat transfer to the coil and to the extension-wire insulation during welding.

Electrical-connector materials should be nonmagnetic and must be compatible with the weld-joint requirements, including requirements for welding dissimilar metals, as the nonmagnetic connector may be welded to a ferromagnetic casing. The 300-series CRES pins are recommended over plated pins of alloys with lower electrical resistance when overall distribution of electrical-system resistance permits.

It is recommended that each actuator with an electrical connector include provision for lockwiring the mating harness or cable connector to the actuator. System assembly specifications should include requirements for lockwiring all completed connector attachments in a flight system.

The potential for errors in attaching electrical conduit connectors to control-system operator and instrumentation connectors, and the consequences of such errors, should be evaluated in determining the extent to which connectors in a system should be noninterchangeable. Although lockwiring and noninterchangeability are primarily system design considerations, their impact on component design should be anticipated in component preliminary-design activity.

3.1.4.4.2 Connector Breakdown

Electrical-connector pin arcing or flashover shall not endanger solenoid-actuator functional integrity.

Compliance with the requirements of this criterion emphasizes the need for the acceptance tests recommended in section 3.1.4.3.2.2 so that units with marginal or deficient dielectric strength can be detected and rejected during the manufacturing process. Obtaining and maintaining adequate cleanliness of parts is mandatory.

Pure, dry nitrogen gas is the generally recommended fill gas for hermetically sealed enclosures. When nitrogen is used, it is recommended that procedures be specified and implemented to ensure removal of all helium test gas and removal of all moisture prior to final seal closure.

With sealed units, the effects of gas temperature on enclosed-gas pressure, and consequently on gas dielectric strength, should be evaluated. When gas mixtures or gases other than nitrogen are used, the consequent reduction in gas dielectric strength should be evaluated. When high-potential tests of coil insulation are performed in the presence of a gas that is different from the gas that is present under operating conditions, or when these tests are performed with different gas conditions, the resultant variations in connector-pin dielectric strength should be evaluated in determining the adequacy of the test.

When the insulation distance between connector pins or between pins and the connector housing results in marginal dielectric strength, the use of internal potting compounds and external compressible pin seals is recommended.

3.1.5 Snap-Action Actuators

3.1.5.1 REGENERATIVE-FORCE ACTUATORS

In snap-action actuators that utilize fluid pressures for generating actuation forces as functions of actuator displacement, extraneous forces shall be accommodated in complying with pressure setting requirements.

Satisfaction of this criterion requires accurate analysis of all forces participating in the actuator force balance, and of the effects of design tolerances on those forces. In the design concept of figure 12(a), friction forces are minimized with diaphragms for balancing inlet-pressure forces and for signal-pressure sensing. Any difference between the effective cross-sectional areas of the poppet inlet seat and the inlet-pressure force-balancing diaphragm introduces an axial force that is proportional to the inlet pressure. The reference spring is adjusted for actuation at a specified signal pressure with inlet pressure at its nominal value. Tolerances on inlet pressure and tolerances affecting force-balancing areas introduce extraneous forces affecting the actuator force balance and signal-pressure setting. Retarding forces resulting from flow-related increases in vent-cavity pressure during the actuation transient must be accounted for, and are minimized by minimizing vent flow resistance. In the design concept of figure 12(b), the effects of O-ring friction on signal-pressure setting are minimized by sizing the signal-pressure sensing piston so that friction forces are a small percentage of the force required for actuation. Pressure in the secondary actuation cavity, acting against the spool cross-sectional area, introduces an actuation force as a function of leakage and supply pressure. Spool valve axial forces resulting from flow and changes in flow are additional extraneous forces affecting the actuation force balance. Extraneous axial forces at nominal supply pressure are accommodated by adjusting the reference spring with nominal supply pressure applied. The effects of tolerances on supply pressure and variations in O-ring friction are factors to be accounted for in designing for the required pressure-setting accuracy.

3.1.5.2 NEGATIVE-RATE-SPRING ACTUATORS

In actuators with negative-spring-rate mechanical springs, spring contact friction and variations in spring characteristics shall not affect compliance with pressure-setting requirements.

When repeatable Belleville-spring characteristics are required, it is recommended that the springs be fabricated entirely by machining rather than by forming or by partial forming. Testing of each spring is necessary for rejection of springs that comply with dimensional specifications, but not with load-deflection requirements. When precise control of the net mechanical spring rate is required for precise control of the difference in forces required for actuation and for deactuation, Belleville springs should have shim adjustment of the installed height for spring-rate adjustment as discussed and illustrated in section 2.1.5.2. The use of hardened, smoothly finished spring support surfaces is recommended for minimizing friction at spring contact surfaces.

3.1.6 Rotary Actuators

3.1.6.1 HYDRAULIC AND PNEUMATIC ACTUATORS

Actuation-fluid leakage and variations in that leakage shall remain within specified limits under all operating and test conditions.

Satisfaction of this criterion requires examination of the effects of pressures, temperatures, and rotor angular displacement, and of transients in these parameters, on actuation-fluid leakage. The design requirements for a hydraulic or pneumatic rotary actuator should include limits for variations in leakage in addition to a maximum leakage limit so that effective control of response can be achieved. Design of an actuator that satisfies all environmental and performance requirements may require a number of iterations in the process of establishing acceptable operational and leakage flowrates and optimizing the design requirements for the system, the operator, and the actuator.

3.1.6.2 ELECTRIC-MOTOR ACTUATORS

Electric-motor performance and structural integrity shall remain within acceptable limits under all storage, test, and operational conditions.

Requirements for specific operational characteristics and design features for an electric-motor assembly must be coordinated closely with overall system requirements and with the requirements of the device that will be positioned or powered by the motor. In many applications, e.g., in motor-actuated valves for cryogenic fluids or for hot gases, the need for standoff attachment posts and thermal insulation barriers must be evaluated, and heat transfer to or from a motor must be accommodated.

It is essential that electric-motor assemblies, including instrumentation, be enclosed in sealed housings or containers fabricated with corrosion-resistant materials, so that a dry inert internal environment is maintained.

The high vibration levels associated with space-flight service require careful attention to the design of motor mounts and supports so that vibration amplification at a motor is minimized. The use of preloaded or spring-loaded angular contact bearings is recommended for minimizing vibration-induced oscillatory relative motion between the rotor and the stator in larger motors. Close-clearance journals and sleeve bearings must be designed to accommodate extensive cyclic relative motion. The design practices for solenoid actuators (sec. 3.1.4) generally are applicable to electric motors, allowances being made for differences in configurations.

3.1.6.2.1 Redundant-Electric-Motor Actuators

Installation of a redundant motor shall not result in reduction of magnetic field strength or degradation of adjacent motor performance.

The use of nonmagnetic housings is recommended. Motor installations should be designed to permit assembly of the rotor and the armature prior to removal of magnetic keepers. Redundant motors should be spaced sufficiently far apart to preclude interference in the magnetic fields. Magnet north and south poles should be oriented by mechanical design features that preclude incorrect installation.

3.2 OPERATOR CONFIGURATIONS

3.2.1 Pilot Valves

3.2.1.1 EFFECTIVE FLOW AREA

Pilot-valve fluid flow resistance shall not impede control of actuator response.

A moderately oversized effective flow area and provision for a replaceable orifice in the valve assembly or in the valve-and-actuator circuit is recommended. This practice allows trimming the overall system flow resistance when accurate actuator response-time control is required. In addition, it is recommended that pilot-valve fabrication and assembly tolerances be controlled to maintain the effective flow area of production valves within a tolerance band that is consistent with the use of precision orifices (sec. 3.1.1.2.1).

3.2.1.2 POSITIVE ACTUATION

Pilot-valve actuation shall be sufficiently positive and chatter-free to preclude adverse effects on system dynamics.

Prediction of pilot-valve forces should be based on analysis of dynamic as well as steady-state flow and pressure distributions. In the design of pressure-actuated pilot valves, potential interaction between valve displacement and the signal pressure, as sensed internally by the valve actuator, must be evaluated. The use of snap-action actuator concepts is recommended for pressure-actuated pilot valves whenever the required differential between actuation pressure and deactuation pressure can be tolerated in system operation.

3.2.1.3 RESPONSE TO PRESSURE TRANSIENTS

Operation of a pressure-actuated pilot valve shall remain within set pressure limits throughout the range of applied pressure rates of change.

A pilot valve and its actuator should be close coupled to the sensed-pressure location, and intermediate flow restriction and capacitance should be held to a minimum. When more than one actuation surface is exposed to signal pressures, the effects of pressure transients acting on each surface must be recognized. In the design of hydraulic systems as well as pneumatic systems, make allowances for fluid-flow and compressibility effects when relating dynamic forces acting on the pilot valve to transients in sensed pressure. Dynamic analysis and tests should verify that actuation will occur within the specified sensed-pressure range when the sensed pressure is changing at its minimum or at its maximum rate.

3.2.1.4 STAGNANT-FLUID TEMPERATURE

Localized temperature effects on stagnant fluid in a pilot valve and its actuator shall not degrade performance.

It is recommended that preliminary design activity include heat-transfer analysis to the extent that temperature extremes of the pilot-valve fluid are known and are provided for under all valve exposure conditions. Thermal problems are especially significant in tight-shutoff hydraulic pilot valves designed as separate assemblies. When heat transfer to or from a pilot valve can result in fluid temperatures exceeding allowable limits, relocate the valve, or insulate it, or otherwise minimize heat transfer.

3.2.1.5 CONTAMINATION

Protection against contaminants in a pilot valve shall not result in contaminant traps.

When filters are included in a pilot valve assembly for protection against contaminants, it is recommended that their use be limited to ports through which valve inflow occurs in normal operation. It is also recommended that passages and ports for unidirectional fluid outflow be designed to facilitate contaminant outflow. Section 3.1.1.2.3 is generally applicable to pilot valves as well as to flow restrictors.

3.2.2 Servovalves

3.2.2.1 FLOW GAIN

Servovalve flow/displacement characteristics shall satisfy system requirements for speed of response as well as requirements for quiescent steady-state operation.

It is recommended that the nonlinear flow/displacement characteristics of a servovalve, inherent or deliberately contoured, be evaluated in the analysis and design of any proposed closed-loop actuation system. It is recommended further that evaluation of the flow/displacement characteristics and their effects on actuator displacement response be accompanied by evaluation of the pressure/displacement characteristics near the servovalve neutral position and their effects on system response to actuator load changes. As the flow/displacement and pressure/displacement characteristics of a servovalve are interrelated, the design criterion of this section and the design criterion of section 3.2.2.2 are also interrelated, and both must be satisfied within the same design.

3.2.2.2 PRESSURE GAIN

Servovalve pressure/displacement characteristics shall satisfy system requirements for system load stiffness as well as requirements for quiescent steady-state operation.

This criterion and the criterion of section 3.2.2.1 must be satisfied concurrently to obtain a successful design. It is recommended that design evaluation be based on realistic nonlinear characteristics, allowance being made for the effects of design tolerances, rather than on idealized characteristics that frequently are misleading in analyzing system performance.

3.2.2.3 FLUID-FLOW DYNAMIC FORCES

Fluid-flow reaction forces acting on a servovalve shall be accommodated in maintaining dynamically stable operation.

Satisfying this criterion requires detailed analysis of the fluid-flow dynamics associated with a servovalve design under all steady-state and transient operating conditions. It is recommended that servovalves be designed so that any unbalanced flow-reaction forces act in the direction of dynamic damping of valve displacement oscillations.

3.2.2.4 SPOOL/SLEEVE MATERIALS AND FINISHES

Materials, hardnesses, and surface finishes for a spool valve and matching sleeve shall preclude surface or metering-edge damage as a result of storage, handling, or service exposure to environmental or operational fluids and fluid contaminants.

The use of fully hardened 440C CRES is recommended for spool valves and sleeves except for those applications wherein high temperatures or chemical incompatibility precludes use of this material. With 440C CRES, all spool and sleeve surfaces should be free of machining marks or scratches, and all surfaces should have a finish of 16 μ in. (0.406 μm) AA or smoother. Because smooth surfaces are essential in obtaining corrosion resistance, a 16 μ in. (0.406 μm) AA finish, or smoother, should be specified for all surfaces including those of through holes and flow passages.

3.2.2.5 SPOOL/SLEEVE CLEARANCES

Spool/sleeve clearances shall remain within allowable limits under all test and operating conditions

Design specifications controlling spool-to-sleeve diametral clearance should be based on an evaluation of the importance of leakage relative to susceptibility to distortion as well as on the need for precise control of functional characteristics. Spools and sleeves must be sufficiently rigid to preclude spool-valve binding as a result of strains associated with installation forces and pressure forces under all test and operating conditions. Sleeve housings should be designed to minimize thermal gradients or housing strain at the sleeve locations. Filtration requirements should be based primarily on successful experience with similar applications under the full range of test, operation, and service-life conditions.

3.2.2.6 RELIABILITY

Servovalves and actuators shall satisfy system reliability requirements and component functional requirements.

Specifications on servovalve size, weight, electrical input power, leakage, and frequency response should be based on an assessment of their importance relative to reliability. Reliability should not be compromised for size and weight reductions of secondary importance. Input-power limitations should be based on the capabilities of modern electronics in obtaining signal level ranges that extend well beyond system noise levels. Design specifications for servovalve frequency response should be related to the actuation-system needs so that torque motor size and the inertia of moving parts will not be restricted needlessly. Consideration of mechanical-feedback servoactuators requires system dynamic analysis that accounts for torque-motor and servovalve time lags, phase lags, hysteresis, and accuracy limitations. Consideration of redundancy requires system evaluations beyond the scope of this monograph.

3.2.3 Pressure Dividers

3.2.3.1 TEMPERATURE-EFFECT COMPENSATION

A pneumatic pressure divider shall comply with all performance requirements throughout the range of temperatures and temperature transients to which it is exposed.

It is recommended that design analysis of pneumatic pressure dividers take into account the effects of temperature distributions and heat transfer on calibration accuracy. When temperature-effect compensation is required, the design techniques in section 2.2.3.1 are recommended. Figure 16 illustrates fundamental concepts that can be adapted as required for specific hardware configurations.

3.2.3.2 FLOW-RESTRICTOR CONFIGURATION

The flow-restrictor configuration shall have no adverse effect on pressure-divider calibration throughout the range of operating pressures and flowrates.

Flow restrictors with sharp-edged throat sections are recommended. However, if a pneumatic pressure divider is required to operate through a range of operating conditions, the inlet flow restrictor may experience subsonic flow if sharp-edged orifices are used. In this event, a sharp-edged conical flow restrictor should be used to extend the choked-flow range. The use of converging-diverging flow restrictors to extend the choked-flow range is not recommended for applications with a wide range of inlet pressures, because of pressure-level effects on flow-throat boundary layers that form upstream from and extend through the throat. The practices in section 3.1.1.2.1 are applicable to all flow restrictors.

3.3 SEALS FOR ACTUATORS AND OPERATORS

3.3.1 General Considerations

3.3.1.1 COMPATIBILITY WITH FLUIDS

Seal materials shall be chemically compatible with all fluids to which they may be exposed.

Compatibility problems resulting from seal and lubricant exposure to cleaning, flushing, and test fluids emphasize the necessity for complete and detailed specification of processing, installation, test, and servicing procedures as part of the design process. In bipropellant systems, when a seal material is compatible with one propellant and not with the other, a seal should be designed so that it cannot be installed inadvertently in a component in which the seal could be exposed to a fluid with which it is not compatible. Cross-compatibility problems with elastomer seals usually can be eliminated by the use of metal seals, Teflon seals, or Teflon-coated metal seals.

3.3.1.2 LEAKAGE MEASUREMENT

Seal installation shall satisfy leakage testing requirements as well as functional requirements.

Design activity should include, or be closely coordinated with, the specification of test setups and procedures for determining leakage as well as the specification of allowable leakage limits. When leakage through a specific static or dynamic seal is a critical design factor, design features for convenient and accurate measurement of that leakage should be provided. The use of dual seals with intermediate tapoff ports for leakage measurement is recommended for critical seal applications wherever feasible within design envelope constraints. When direct leakage measurements are not feasible, actuator and operator assembly design should be related closely to the design of adapters and test fixtures so that leakage through individual seals can be isolated and leakage measurement accuracy will be commensurate with specified seal leakage limits. It is recommended that progressively greater seal leakage limits or tolerances be specified for new component tests, for engine or vehicle system tests, and for postservice tests to allow for differences in test equipment and for anticipated normal seal wear.

3.3.1.3 SEEPAGE AT LOW DIFFERENTIAL PRESSURE

Actuator and operator seals shall maintain leakage within allowable limits at the minimum pressure differential applied during pressurized storage and system standby as well as at the maximum pressure differential applied during operation or test.

It is recommended that all seals for applications requiring essentially zero leakage be designed with special consideration of long-term seepage under minimum pressure conditions. Seals and seal installations should be designed to maintain adequate installed load for seepage sealing, with no more than static head pressure applied, throughout the test, storage, and operational life of a component. The details for successful seal design are presented in reference 147.

3.3.1.4 INADVERTENT CONTACT SEALING

Mechanical position stops shall not at any time impede flow of actuation fluid to the full effective area of the piston.

An actuator like that in figure 18 should be designed with generous cutout sections or ports in the piston skirt or in the housing to provide flow passages through or around the contact surfaces of the position stop.

3.3.2 Dynamic Seals

3.3.2.1 SEAL CONTACT SURFACES

Contact surfaces shall not impair the mechanical integrity of dynamic seals.

Specification of surface roughness in the range of 16 to 32 μ in. (0.406 to 0.812 μm) AA is generally recommended for surfaces in dynamic contact with seals. The specified surface finish must be consistent with the requirements of specific seal configurations. Although a single-value specification on a drawing usually is sufficient, a minimum as well as a maximum surface roughness should be specified if the manufacturing process for a specific part can result in a surface that is too smooth.

Surfaces in dynamic contact with seals should be specified by drawing notes as sealing surfaces that must be free of nicks, scratches, and asperities that are acceptable under the ASA specification (ref. 148) for surface texture but still result in unacceptable leakage. The use of abrasive materials or surface coatings such as oxides and carbides should be avoided for surfaces in dynamic contact with seals.

Plastic seals with low coefficients of friction are recommended in preference to elastomeric seals for minimizing adherence problems when exposure to flushing fluids, operating fluids, or long-term storage results in poor or unreliable lubrication.

3.3.2.2 LIP SEALS FOR CRYOGENICS

Lip seals for cryogenics shall accommodate the maximum applied differential pressure without damage or excessive leakage.

The simple design concept illustrated in figure 19 is recommended for applications within its pressure limitations. For nominal operating pressures greater than 500 psi (3.45 MN/m^2), the concept of figure 20(a) is recommended; for nominal operating pressures greater than 1500 psi (10.3 MN/m^2), the concept of figure 20(b) is recommended. When seal configurations or applications differ appreciably from designs with which there is experience and application data, seal material thicknesses and dimensions for seals and seal retaining details should be optimized by iterative design and development effort.

3.3.2.3 WIPER SEALS

A wiper seal shall shield a dynamic seal from direct exposure to contamination.

The use of a wiper seal is recommended when component and system design features, process specifications, and procedural controls do not ensure cleanliness in the seal environment. External environments generally are considered as uncontrolled sources of contamination, as are actuator cavities exposed to unfiltered fluids. Any actuator cavity containing elements with metal-to-metal relative motion that may generate particles should be considered as an uncontrolled source of contamination.

A lip seal with an interference fit on the movable element that contacts a dynamic seal is the recommended wiper-seal configuration. Acceptable seal configurations are illustrated in figures 19 and 20.

3.3.3 Static Seals

3.3.3.1 SEAL COMPRESSION FORCES

Seal compression forces shall be sufficiently uniform to eliminate unacceptable leakage caused by distortion of seals or seal-retaining surfaces.

Satisfaction of this criterion requires careful design of seal-retaining parts on the basis of their rigidity. Assembly procedures specific to the individual design must be controlled to minimize distortion resulting from uneven distribution of load as fasteners are tightened. The number and size of fasteners must be adequate for the required seal loads, due allowance being made for potential inequalities in fastener loads during installation.

3.3.3.2 SEAL CONTACT SURFACES

Surface roughness of parts in contact with static seals shall not degrade sealing capabilities.

It is recommended that drawings identify all seal contact surfaces. Machining-roughness symbols for sealing surfaces should be accompanied by a note specifying that those surfaces be free of nicks, scratches, or asperities. Drawings should also specify the use of protective covers for parts with sealing surfaces to minimize potential handling damage.

3.3.4 Seal Welds

3.3.4.1 PROVISIONS FOR PRELIMINARY TESTING

The design of an actuator or operator with seal-welded joint seams or with welded plumbing connections shall provide for convenient preliminary testing prior to welding.

It is recommended that actuators and operators requiring seal welding be designed for the use of temporary plumbing connections and temporary static seals for convenient preliminary testing. The use of tubing stub-outs as integral parts of housings is recommended; plumbing weld joints thus will be kept at a sufficient distance from the housing to eliminate problems resulting from heat transfer to an assembled unit during welding. Dimensions, tolerances, and surface finish for the outside diameter of each stub-out should be selected and specified so that standard 0-rings can be used in slip-on fittings for temporary plumbing connections. All seal-welded external joints should be designed to permit the use of standard 0-rings or plastic static seals for preliminary testing. If static seals are left in place during seal welding, they must be resistant to welding heat and compatible with the fluids to be used.

3.3.4.2 PROVISIONS FOR DISASSEMBLY AND REASSEMBLY

Design of a seal-welded joint shall provide for removal and reuse in disassembly and reassembly.

It is recommended that seal-welded assembly joints be designed so that structural loads, including differential expansion or contraction loads, are mechanically supported and restrained so that weld seams with minimum penetration can be used. In addition, seal-welded joints should be designed to provide sufficient material, accessible for weld-seam removal by machining, to permit disassembly and reassembly at least several times. Figure 30 illustrates a seal-welded diaphragm actuator. Structural loads are supported by a screw thread, a light seal weld is specified, the joint is accessible for machining, and enough material is provided for approximately five weld-removal-and-reweld operations.

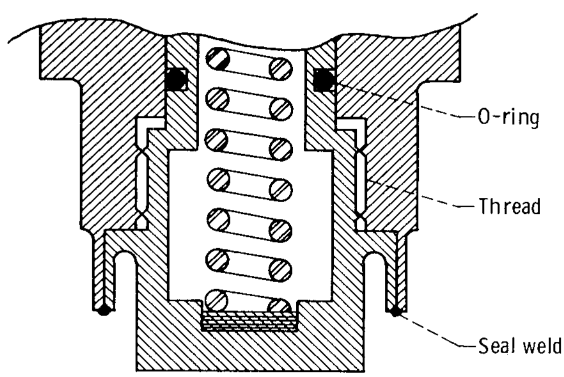


Figure 30. - Sketch of seal-welded joint in a diaphragm actuator.

3.4 MECHANICAL TRANSMISSION

3.4.1 Common Problems

3.4.1.1 MOISTURE IN VENTED CAVITIES

Adverse environmental conditions shall not degrade functioning of relief valves for actuator vented cavities.

Previously qualified vent port relief valves with demonstrated capabilities for the intended conditions should be used in new actuator designs. Vent port relief valves should be installed in an actuator assembly in a manner that minimizes the potential for moisture accumulation near the relief valve. Reference 150 provides specific design details. The vent port relief valve in an actuator assembly should be located where exposure to accidental collision damage is a minimum.

3.4.1.2 FASTENER RETENTION

Fasteners in a mechanism for the transmission of actuator power shall not loosen or disengage.

Self-locking nuts and self-locking tapped bolthole inserts that rely on mechanical interference of distorted threads are the recommended fastener-locking devices for power-transmission elements. The torque required for developing the nut or bolt clamping force should be specified on an assembly drawing as a torque value to be applied in addition to the torque required for rotation against the thread interference (e.g., "Secure nut with 50 in.-lb. (5.65 N-m) torque in excess of the required running torque.") The use of shoulder bolts with self-locking nuts is recommended in preference to linkage pins with snap-ring retainers. The use of scrubbing-contact setscrews is strongly discouraged. Staking of fasteners is unacceptable. Wherever feasible, parts surrounding fasteners in a completed assembly should be designed to limit disengagement of a fastener.

3.4.1.3 GUIDES

3.4.1.3.1 Materials and Configurations

3.4.1.3.1.1 Nonmetal Guides

Friction forces at sliding contact surfaces in an assembly for the transmission of actuator linear motion shall remain within acceptable limits.

It is recommended that elastic or plastic seals and bushings be used wherever feasible to minimize metal-to-metal sliding contact. Rod or shaft materials that are heat treated for hardness are recommended; these materials minimize surface damage should metal-to-metal contact occur and provide maximum strength to withstand loads under fast acceleration or impact.

3.4.1.3.1.2 Metal-to-Metal Guides

Metal-to-metal relative motion in an actuator assembly shall not damage the surfaces involved.

It is recommended that actuators and their transmission elements be designed so that surfaces exposed to metal-to-metal relative motion will not enter into contact with dynamic seals. Transmission elements and their guides must be made from materials that are compatible with the anticipated environments and service fluids. In addition, the selected material combinations must be resistant to wear, abrasion, and galling. Linkage elements should be made from materials that are heat treated for hardness or are coated with a hard material. Guides should be made from material with good bearing properties (e.g., aluminum-bronze) if it is compatible with the service environment and fluid.

3.4.1.3.1.3 Flexure Guides

> *Flexure guides in an assembly for the transmission of actuator linear motion shall have sufficient lateral and angular stiffness and axial flexibility to minimize the effects of misalignment or sliding contact.*

The use of metal flexure guides or supports is recommended for minimizing lateral forces and maintaining alignment of moving elements in short-stroke actuator applications when metal-to-metal relative motion, friction forces, vibration, or misalignment are design problems. The use of flexure guides as short-stroke antirotation devices also is recommended. Metal flexure guides should be designed for minimum mechanical spring rates in the axial direction; this practice will minimize the guide contribution to the axial-direction force balance and, consequently, will reduce its potential for impeding actuation or deactuation. Metal flexure guides should be designed for lateral and angular mechanical spring rates that are great enough to oppose significant radial or rotational displacement under operational conditions in acceleration and vibration environments.

3.4.1.3.2 Lengths and Clearances

> *Linear motion actuator guides shall not bind, score, or chatter under any operating condition.*

To ensure constancy of the intended guide length under all operational conditions, general design practices for guide length-to-diameter ratios must be supplemented by analysis of all dimensions and dimensional tolerances affecting angularity at sliding contact surfaces in a complete assembly. When guiding occurs on nonmetal dynamic seals or guide inserts, the design analysis must allow for deflection, distortion, or wear of nonmetal parts in determining the potential for metal-to-metal sliding contact. If metal-to-metal contact can occur, contacting metallic materials should be selected for maximum resistance to scoring or galling, and leading edges should be rounded or chamfered to eliminate cutting edges. The number of parts and dimensions affecting clearances, normalities, and eccentricities should be minimized. Transient and steady-state temperature distributions under all test conditions and all operating conditions must be accommodated.

3.5 INSTRUMENTATION FOR ACTUATORS AND OPERATORS

3.5.1 General Requirements

3.5.1.1 ISOLATION OF FUNCTIONAL INSTRUMENTATION

Failure of monitoring instrumentation shall not interfere with system operation.

The functional and the monitoring instrumentation subsystems and all corresponding components within each subsystem should be completely independent of each other. In the case of actuator position-indicating switches, two position-sensing electrical switches with independent power supplies, independent wiring, and independent output circuits should be used. The two switches can be in a single subassembly with a common position-sensing input member, provided that a mechanical malfunction of the monitoring switch does not impede operation of the functional switch.

3.5.1.2 SAFE FAILURE MODES

System functioning shall be independent of structural or operational failure of an instrument.

When a pressure-sensing device such as an operator-supply-pressure transducer is installed in the operator assembly, the device should be provided with an orifice that will limit fluid outflow in the event of an internal structural failure, e.g., rupture of a pressure-sensing diaphragm. A position-indicating device such as an actuator position indicator should be designed to include a shear pin or an over-travel mechanism so that actuator motion will not be impeded by an operational failure of the device.

3.5.1.3 ELECTRICAL CONNECTORS

Electrical connectors for instrumentation shall satisfy structural requirements and system interface requirements.

Except for the use of flanged joints in place of welded seams, recommended practices of section 3.1.4.4.1 for solenoid-actuator electrical connectors are applicable to instrumentation connectors. Instrumentation electrical connectors for position-indicating switches in an actuator assembly should be attached as sealed closures for a vented cavity within the assembly. Vent port relief valves should be used to protect the vented cavity from entry of moisture from the actuator external environment and from inadvertent pressurization.

3.5.1.4 ACCESSIBILITY

Test as well as flight instrumentation shall be accessible for installation, calibration, and servicing.

It is recommended that the needs for checkout and test instrumentation as well as for flight instrumentation be investigated thoroughly during preliminary design activities. It is essential that instrumentation be accessible at the system level as well as at the component level.

3.5.2 Pressure Measurement

3.5.2.1 PRESSURE-SENSING LOCATIONS

The locations for measurement of actuator internal pressures shall not result in erroneous data.

Avoid indirect measurements. Locate pressure taps in the actuator housing for direct measurement of internal pressures. If actuator pressure taps for attachment of test instrumentation are deemed undesirable in flight hardware, the hardware should be designed to permit inclusion of pressure taps in units selected for development testing or for performance-analysis testing.

3.5.2.2 TEST-INSTRUMENTATION EFFECTS

Pressure-sensing instrumentation for ground test shall not affect the critical time constants of an operator/actuator design.

Test instrumentation should be selected, installed, and primed if necessary, so that the increase in volume or the increase in elasticity at the sensed-pressure location does not result in system performance under test or checkout conditions that is significantly different from system performance under flight operating conditions. For dynamic tests, it may be necessary to remove or deactivate instrumentation (e.g., pressure gauges) installed for measurements under static-test conditions. This criterion is satisfied best by formal specification and control of test setups, instrumentation, and procedures based on analysis of test-instrumentation effects.

3.5.2.3 FLIGHT-INSTRUMENTATION EFFECTS

Actuators and operators for systems that include pressure-measuring flight instrumentation shall eliminate or accommodate instrumentation effects that tend to degrade dynamic performance.

Pressure-measuring flight instrumentation should be close-coupled to the volume in which pressure is to be sensed and should be designed for negligible sensing-element displacement. If a pressure-sensing instrument must be located at a distance from an actuator and operator functional system, time lags in instrument response must be accommodated. In an hydraulic system, time lags resulting from compressibility of trapped gases in a sensing line can be minimized or eliminated by a circulation flow that automatically primes the sensing line.

3.5.2.4 PRESSURE-TRANSIENT EFFECTS

Operation of pressure-sensing instrumentation shall remain within accuracy tolerances throughout the range of rate of change of applied pressure.

This criterion is satisfied best by close-coupling the pressure-sensing instruments to the sensed-pressure locations. In hydraulic systems, pressure-sensing flight instrumentation can be mounted remotely from sensed-pressure locations if the signal lines are self-priming.

3.5.3 Position Indication

3.5.3.1 BACKLASH AND HYSTERESIS

A position-indicating device for closed-loop control, and its attachment to the actuator, shall be free from excessive deadband or hysteresis.

The recommended practice is to attach the position-indicating device directly to the actuator piston or rotor or to a linkage member that is attached to the piston or rotor with no mechanical backlash or flexibility in the linkage. Backlash as a result of mechanical clearance in the attachment should be minimized or eliminated. Hysteresis as a result of attachment flexibility that may be required for alignment should be minimized or eliminated. If the position-indicating device must be attached to an actuator linkage member with mechanical backlash or flexibility between that member and the actuator piston or rotor, the linkage backlash and flexibility should be minimized, and must be considered as attachment clearance and flexibility. This criterion is satisfied only if the sum of the mechanical clearances in the attachment method and the deadband within the indicating device and the sum of attachment hysteresis and indicating-device hysteresis are both within the limits allowed for the combination of actuator and position indicator.

3.5.3.2 ATTACHMENT MECHANICAL INTEGRITY

The mechanism attaching a position-indicating device to an actuator or actuator linkage shall maintain a controlled relation between the indicator and the actuator under all operating conditions.

The attachment mechanism should be designed to maintain its original configuration under the extremes of actuator acceleration, shock, vibration, and pressure transients. Recoverable deflections within elastic limits are evaluated with respect to the design criterion of section 3.5.3.1; this criterion applies to nonrecoverable deformations, loosening of fasteners, and failure of force-bias springs in preventing backlash. It is recommended that the design of each attachment mechanism be based on an analysis of the applied transient forces, taking into account the inertia forces at maximum actuator acceleration and at maximum shock and vibration inputs (including shock and vibration amplification at the indicator), the momentum forces when the actuator piston reaches a mechanical stop at maximum velocity, pressure forces resulting from dashpot effects at maximum velocity, and the forces resulting from static and dynamic friction.

3.5.3.3 POSITION-INDICATOR RELIABILITY

Position-indicating devices and instrumentation systems shall satisfy system reliability requirements as well as actuator functional requirements.

When actuator position indication, on-off or continuous, is a functional requirement for system operation, reliability considerations must be given equal priority with functional considerations. Reliability should not be compromised by overly conservative specification of functional requirements or by arbitrary limitations on envelope and weight. Alternate concepts for position indicators should be evaluated in selecting the most reliable concept that satisfies realistic functional requirements.

When actuator position indication is required for checkout or test purposes only, it is recommended that actuation systems be designed for convenient attachment and removal of test instrumentation. The reliability of flight hardware assemblies should not be compromised by the inclusion of instrumentation or instrumentation attachment features not required for flight operation.

3.5.4 Temperature Measurement

Temperature-monitoring instrumentation shall measure actual temperature distributions in the hardware.

It is recommended that all component development testing to obtain thermal-effect information be performed with thermocouples in intimate contact with the test specimens. For actuator assemblies with large metal masses, exploratory tests with thermocouples attached to internal parts are useful in determining the time duration, under controlled test conditions, for temperature stabilization. For production unit thermal-environment tests, specific locations and methods of attachment for test specimen thermocouples should be specified for obtaining hardware temperature data. As an alternate, test setups, test conditions, and a minimum elapsed time for temperature stabilization should be specified, specifications being based on prior development test experience.

3.5.5 Vibration-Test Measurement

Each actuator and operator shall accommodate the attachment of accelerometers.

The design configuration should permit the simple temporary attachment of accelerometers at locations where maximum vibration amplitudes are anticipated. Otherwise, mounting pads should be provided to permit drilling and tapping for attaching accelerometers to units selected for vibration testing. Accelerometer locations and attachment methods should be specified and controlled to preclude their altering the vibration-response characteristics of systems with small spring mass.

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GLOSSARY

Term	Definition
actuator	any device that converts hydraulic, pneumatic, electrical, or potential energy into mechanical motion
anodize	produce a protective oxide film on a metal (usually Al) by electrochemical means
articulated	segmented or jointed
Belleville spring	truncated, conical, metal spring washer that can be designed to provide a negative mechanical spring rate
bellows	thin-wall, circumferentially corrugated cylinder that can be elongated or compressed longitudinally
chatter	rapid uncontrolled seating and unseating
closed loop	an electrical or mechanical system in which the output is compared with the input command signal, and any discrepancy between the two results in corrective action by the system elements
coil clash	contact of adjacent coils as helical spring is compressed
Coulomb damping	dry-friction damping; friction force is constant in magnitude but always directed opposite to the velocity
cryogenic	fluids or conditions at low temperatures, usually at or below -150°C (123 K)
diaphragm	thin dividing membrane that can be used as a seal to prevent fluid leakage and as an actuator to transform an applied pressure to linear force
digital	operating in discrete increments
elastomer	polymeric material that at room temperature can be stretched to at least twice its original length and upon release of stress will return quickly to approximately its original length
ferromagnetic	of or relating to materials that readily provide a conductive path for magnetic flux and that can be attracted by a magnet

Term	Definition
free height	height of a spring in the free (unloaded) condition
gamma	unit of magnetic induction; 1 gamma = 10^{-5} gauss
hermetic	pneumatically leak tight
hi-pot test	dielectric-strength test performed at a high electric potential
hydraulic	a system or device using a liquid as the operating fluid
iterative	proceeding in a step-by-step repetitive manner
modulating	the controlled variable (flow, pressure, or position) is proportional to a sensed parameter and is infinitely variable within the regulated range
negative spring rate	change of mechanical spring force per unit of deflection in a flexure element with the characteristic that an increase in deflection is accompanied by a decrease in flexure force opposing deflection
nonmodulating	the controlled variable (flow, pressure, etc.) cycles between limits
normal positions	positions assumed by the elements in a component when no operating forces are applied
on-off	a system or device in which full-stroke actuation or deactuation occurs in response to input signals
open loop	an electrical or mechanical system in which the response of the output to the input is scheduled or preset; there is no feedback of the output for comparison and corrective adjustment
operator	any device that causes an actuator to function
overstroke	displacement of a component that exceeds the maximum allowable
permeance	ratio of magnetic flux to magnetomotive force in a flux path
pilot valve	an on-off operator that amplifies a low-power control signal to cause full-stroke displacement of an actuator
plastic	high-molecular-weight material that, although hard or firm in its finished state, at some stage in its manufacture is soft enough to be formed through application of heat or pressure or both

Term	Definition
plenum	enclosed volume in a pneumatic system or device for the purpose of providing gas storage capacity
pneumatic	a system or device using a gas as the operating fluid
primed	condition wherein an hydraulic system or device is completely filled with fluid
response	capability of a valve to achieve 63 percent of a signaled position with minimum time delay
restrictor	discrete flow resistance in a fluid flow passage; usually an orifice
safe operating pressure	maximum operating pressure without shield for personnel safety
seal weld	weld seam with the primary function of sealing against fluid leakage
servovalve	modulating operator that amplifies a low-power control signal for variable-displacement, closed-loop control of actuator position
solenoid	helically wound coil of insulated wire that when conducting electricity excites a magnetic field that actuates a movable core
solid height	height of a spring compressed to the point where no further compression is possible
spring rate	change of mechanical spring force per unit of deflection in a flexure element
torque motor	electromagnetic actuator with an angular displacement output

Symbol	Definition
A	area
AA	arithmetic average
A1, A2	any two separate but related areas
CA	effective flow area
cmf	circular mil foot
F	force

Symbol	Definition
g	acceleration due to gravity
k	ratio of gas specific heat at constant pressure to specific heat at constant volume
P	pressure
PA, PB, PC	primary, secondary, and tertiary actuation pressure
PD	pressure at pressure switch
PR	return pressure
PS	supply pressure; also, sensed pressure
P_{amb}	ambient pressure
P_{app}	applied pressure
P_{sig}	signal pressure
P1, P2	any two separate but related pressures
R	specific gas constant; also, electrical resistance
R_T	electrical resistance at a specified temperature
rms	root mean square
S_{sonic}	gas flow parameter (figs. 26 and 27)
T	temperature
TIR	total indicated runout
V	volume
V1, V2	any two separate but related volumes
X	linear displacement
α	temperature coefficient of resistance
τ	time constant

Material[1] (designation in monograph)	Identification
A286	heat-treatable, high-strength austenitic steel
AM-350 AM-355 PH13-8Mo PH14-8Mo PH15-7Mo 15-5PH 17-4PH 17-7PH	semi-austenitic or martensitic precipitation and transformation hardening stainless steels
40E	cast aluminum alloy with zinc as the principal alloying element
195 B195	cast aluminum alloys with copper as the principal alloying element
220	cast aluminum alloy with magnesium as principal alloying element
302 304L 316L 321 347	austenitic nickel-chromium-iron steels
430	high-chromium, low-carbon stainless steel with good formability, weldability, and corrosion resistance
430F	magnet-quality 430 steel
440C	martensitic chromium steel
446	high-permeability, magnet-quality steel
2014 2024 2219	wrought aluminum alloys with copper as principal alloying element
7001 7075 7079 7178	wrought aluminum alloys with zinc as principal alloying element

[1]Additional information on metallic materials herein can be found in the 1972 SAE Handbook, SAE, Two Pennsylvania Plaza, New York, NY; and in MIL-HDBK-5B, Metallic Materials and Elements for Aerospace Vehicle Structures, Dept. of Defense, Washington, DC, Sept. 1971.

Material[1] (designation in monograph)	Identification
beryllium copper	copper-beryllium-cobalt alloy with relatively high strength and hardness
constantan	copper-nickel alloy with high electrical resistivity and low temperature coefficient of resistance
CRES	corrosion resistant steel
Elgiloy	trade name of the Elgin National Watch Co. for a cobalt-nickel alloy used as a high-strength spring material
H-11	a Cr-Mo air-hardening, hot-work tool-and-die steel with a high degree of toughness in the temperature range of 400 to 1000°F (478 to 811 K)
Inconel 718	trade name of International Nickel Co. for precipitation-hardening nickel-based alloy
Inconel X (Inconel X-750)	trade name of International Nickel Co. for age-hardenable nickel-based alloy
Kel-F	trade name of the 3M Corp. for a polymer of chlorotrifluoroethylene
low-alloy steel	steel with low carbon content
maraging steel	martensitic age-hardening nickel-iron alloy
Mylar	trade name of the E. I. duPont Co. for polyethylene terephthalate film
Ni-Span-C	trade name of International Nickel Co. for an iron-nickel-chromium alloy that can be heat treated to produce an essentially constant modulus over a wide termperature range
radio alloys	copper-nickel alloys with electrical resistivity and temperature coefficient of resistance as functions of the nickel content
Refractaloy 26	trade name of the Westinghouse Electric Co. for an austenitic nickel-base alloy
Rene 41	trade name of the General Electric Co. for an austenitic nickel-base alloy
Teflon	trade name of E. I. duPont Co. for a polymer of tetrafluoroethylene

[1] Additional information on metallic materials herein can be found in the 1972 SAE Handbook, SAE, Two Pennsylvania Plaza, New York, NY; and in MIL-HDBK-5B, Metallic Materials and Elements for Aerospace Vehicle Structures, Dept. of Defense, Washington, DC, Sept. 1971.

Material[1] (designation in monograph)	Identification
Ternalloy 7	a precipitation-hardening aluminum alloy having good castability, excellent machinability, and high corrosion resistance
Vascojet 1000	trade name of the Vasco Division of Teledyne Corp. for a modified hot-work die steel

Abbreviations	Identification
AFBMD	Air Force Ballistic Missile Division
AFBSD	Air Force Ballistic Systems Division
AFFDL	Air Force Flight Dynamics Laboratory
AFRPL	Air Force Rocket Propulsion Laboratory
AIEE	American Institute of Electrical Engineers (now IEEE)
ASA	American Standards Association
ASME	American Society of Mechanical Engineers
IEEE	Institute of Electrical and Electronic Engineers
ISA	Instrument Society of America
NACA	National Advisory Committee for Aeronautics
SAE	Society of Automotive Engineers
WADC	Wright Air Development Center

[1] Additional information on metallic materials herein can be found in the 1972 SAE Handbook, SAE, Two Pennsylvania Plaza, New York, NY; and in MIL-HDBK-5B, Metallic Materials and Elements for Aerospace Vehicle Structures, Dept. of Defense, Washington, DC, Sept. 1971.

NASA SPACE VEHICLE DESIGN CRITERIA MONOGRAPHS ISSUED TO DATE

ENVIRONMENT

SP-8005	Solar Electromagnetic Radiation, Revised May 1971
SP-8010	Models of Mars Atmosphere (1967), May 1968
SP-8011	Models of Venus Atmosphere (1972), Revised September 1972
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