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TECHNOLOGY UTILIZATION

NASA Contributions to

ADVANCED VALVE TECHNOLOGY

REVISED AND ENLARGED EDITION

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A SURVEY

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

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NASA Contributions to
ADVANCED VALVE TECHNOLOGY
REVISED AND ENLARGED EDITION

A SURVEY

by

Louis C. Burmeister, John B. Loser
and Eldon C. Sneegas

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Foreword

The exploration of space has necessitated a multitude of innovations and improvements in valves, the controlling elements of both simple and complex fluid-handling systems. In view of the benefits that can be derived from what has been learned in the last few years about the design, performance, and manufacture of these devices, this survey was undertaken. It is one of a series of special publications issued by the National Aeronautics and Space Administration to acquaint nonaerospace industry with knowledge resulting from the space program.

This we had built now w/d was on #8496 J.W.

A first edition of *Advanced Valve Technology* (NASA SP–5019) was published in February 1965. The interest it aroused prompted the Technology Utilization Division of NASA to sponsor preparation of this second edition, which contains additional and more recent information about the results of research and development work done in NASA centers and by NASA contractors.

Other volumes in this series of Technology Utilization Surveys have dealt with NASA contributions to solid lubricants, inorganic coatings, magnetic tape recording, and other subjects of concern to engineers and managers in a great variety of industries.

GEORGE J. HOWICK
Director, Technology Utilization Division
National Aeronautics and Space Administration

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CHAPTER 1

Introduction

Valves have been used for more than 4000 years. The Chinese are said to have used wood-plug valves in bamboo pipelines; artifacts from the depths of the Mediterranean contain fragments of petcocks dating from before the time of Christ. Wooden valves of the Roman Empire bear a striking resemblance to the plug valves on wine casks and beer kegs today.

Nearly every person in the civilized world now comes into daily contact with valves. They are controlling elements in fluid-handling systems, and system control can be no better than the valves being used. Valves serve five primary functions: to start and stop flow, to regulate and throttle flow, to prevent backflow, to regulate pressure, and to relieve excessive pressure. How well these functions are accomplished largely determines the performance of the system.

Management is becoming more and more aware of the importance of valves in industrial plants and processes. In the hydrocarbon and natural gas industries, valves represent about 8 percent of new plant capital expenditures and 10 percent of the maintenance budget for replacement purchases. The price of valve failure and leakage increases as the cost of process fluids rises.

Until the late 1950's valve manufacturers pretty well kept pace with industrial and military demands. The space age then forced them to meet strange and unforeseen specifications. Fluid system control became a major problem in the design and development of missiles, advanced aircraft, hypersonic testing facilities, and space vehicles. Engineers were called on to design valves that could control extremely cold or hot, noxious, highly reactive, intractable, self-igniting fluids; valves that could operate at both high and low temperatures and pressures and high vibration levels, and that could be lightweight and remotely operated. Meteorite penetration, zero-leakage, hard vacuum, radiation tolerance, zero-gravity, and other space-related terms entered the list of specifications for valves. Government-funded programs, many of them originated by the National Aeronautics and Space Administration, produced valves to meet new and strenuous requirements.

NASA-sponsored research programs have advanced valve technology beyond the industrial state of the art. This book was written to disseminate such technology throughout industry. Midwest Research Institute, under contract to NASA, conducted this survey of the open literature and work done in industrial firms and NASA installations.[1] Magazine articles, government publications, engineering indexes, and technical papers published since 1959 were consulted. Specific valve developments were noted and guides for the design, selection, and specification of valves were collected. The book contains valuable guides for valve designers, application engineers, and users.

To identify current valve-problem areas, two principal questions were asked of both industrial manufacturers and personnel at the NASA installations: In what applica-

[1] The following NASA installations were visited: Ames Research Center, Mountain View, Calif.; Flight Research Center, Edwards, Calif.; Jet Propulsion Laboratory, Pasadena, Calif.; Langley Research Center, Hampton, Va.; Lewis Research Center, Cleveland; Manned Spacecraft Center, Houston; and Marshall Space Flight Center, Huntsville, Ala. NASA's Western Operations Office, Santa Monica, Calif., also provided assistance.

tions do commercially available valves fail to meet requirements? For what applications are custom-designed valves necessary?

Commercially available valves are here taken to mean stock items. As might be expected, the first question revealed many instances in which such valves are inadequate. The response to the second question showed that many industrial custom-designed (nonstock) items have been developed at NASA installations.

Specific Problem Areas

Leakage of expensive, toxic, corrosive, or explosive fluids cannot be tolerated. Great effort is being expended to attain the nearly impossible goal of a zero-leakage valve.

The materials selection problems facing the aerospace valve designer are extremely difficult. He must consider many factors:

Large temperature excursions. The annealing effect of temperature cycling can change physical properties. Differential thermal expansion can cause warping and binding.

Vacuum. Many materials outgas in a vacuum environment and undergo property variations. Cold welding of mating metal parts is poorly understood, particularly in connection with such pairs as seats and poppets, plungers and solenoids, and in flanges, crimped tubes, and screw threads.

Extreme temperatures. Materials lose strength at very high temperatures and welding can occur. At low temperatures, many metals become brittle and sealing materials lose plasticity. Extreme temperatures present material selection problems of the greatest magnitude.

Compatibility. Corrosive propellants and other fluids pose major problems in materials selection. Some materials, for example, react with liquid oxygen when shocked. Metal parts may gall and experience excessive wear.

Radiation. Some aerospace and nuclear reactor valves must operate in high-intensity radiation fields. The properties of polymeric materials can be greatly affected. Materials must often be sought that are nonporous, easy to fabricate, readily available, and economical.

Friction and its attendant wear, also are serious problems in space applications. Many lubricants are ineffective in space. Contaminant particles generated by wear can seriously degrade the performance of a valve.

Extremely high or low temperatures require special consideration of differential thermal expansion. In some cases, valves for these extreme conditions may be inoperable at room temperature. The temperature gradients within the separate valve parts may be sufficient to cause trouble even though only one metal is used. Temperature excursions can anneal springs and warp parts, causing permanent sets.

In outer space it is often mandatory that valves operate perfectly for thousands of cycles. Unattended industrial plants have a similar requirement. The factors causing unreliability are legion.

Valve response times of less than 5 milliseconds are required for many aerospace systems.

In systems to release a fuel and an oxidizer simultaneously, valve parts must trace their time-travel curve repeatably in all operations. Extreme precision is required in the fabrication of these valves.

The intense vibration of a rocket imposes a further parameter that cannot be ignored in valve design. Accelerations range from the high values of launch to the zero value of orbiting.

Valve weight and size are very important in aerospace vehicles. Efforts are constantly made to reduce these two parameters.

These problems cover nearly every facet of valve design. The majority, however, occur in three areas: extremely low temperatures, extremely high temperatures, and reliability. The primary symptom of unreliability is leakage. This leakage generally results from contamination in the system.

CHAPTER 2

Leakage[1]

Most manufactured components that retain or exclude fluids undergo some leakage measurement or leak testing. This can be costly because of the time and energy required. The procedure should be as inexpensive as possible and yet show that the component meets the leakage specification. Some aspects of leakage phenomena, theory, and measurement experiments will be presented here to further the understanding of the measurement procedure.

SENSITIVITY CONSIDERATIONS

Leakage measurements are made: (1) to prevent material loss, (2) to prevent contamination, and (3) to detect unreliable components. The allowable loss of propellants from a rocket engine is the difference between the amount initially in the propellant tanks and the amount necessary to complete the mission. This difference, divided by the time the tank remains loaded prior to use, represents the maximum tolerable leakage as determined by material loss considerations. For example, a leakage of 1 cc per day at 20° C of nitrogen tetroxide (N_2O_4) would equal 2.3 pounds of material loss in 2 years. For a lunar mission of a few days' duration, the loss of nitrogen tetroxide at this rate may not be a serious problem, but for an interplanetary mission such a loss rate would be intolerable.

Contamination caused by valve leakage can be severe. Corrosive effects of the leaked substance may degrade nearby equipment and materials. An atmosphere hazardous to personnel may ensue because of the toxic nature of the leaked substance or because of the possibility of fire and explosion. If a rocket-engine valve leaks either fuel or oxidizer into an engine in sufficient quantity, the engine may explode when called upon to deliver thrust. A large volume of leakage is also a nuisance. In many industrial processes that require accurate mixing, a run or batch can be ruined by valve leakage, or the leakage may be so unsightly it demands continual "mopping up."

The maximum tolerable leakage on the basis of contamination can be determined by considering: (1) the amount which will cause serious corrosion damage, (2) the human tolerance for the leaked substance or the amount which will constitute an explosive or flammable situation, and (3) the amount tolerable as a general nuisance.

The third reason for leakage measurement is that it helps determine the reliability of a valve or component. Leakage testing prior to use may indicate such faults as a cracked component, an inadequate joint or seal, or improper component installation in the system.

Any one of the preceding three basic reasons may be a criterion for determination of the maximum tolerable leakage. Knowledge of the consequences of material loss and contamination, together with the basic leakage equations, permits accurate calculation of the allowable leakage. The relationship between allowable leakage and component reliability is nebulous and probably is best derived from experience. The environ-

[1] Based on a paper, "Leakage Phenomena," presented by J. W. Marr, physical chemist, Research and Development Center, General Electric Co., Schenectady, N.Y., at the Valve Technology Seminar, Oct. 21–22, 1965, Midwest Research Institute, Kansas City, Mo.

ment in which a valve is to operate should be specified as completely as possible. The fluid, temperatures, pressures, flow direction, vibration and acceleration conditions, and contamination particle size and concentration, are all factors in valve leakage. The importance of vibration and acceleration conditions is obvious in the case of spring-loaded valve designs.

LEAKAGE UNITS

For gases, the leak or leakage rate is defined as the flow rate of gas through a leak with a specified high pressure on the entering side and a lower pressure on the exit side. The dimensions of leak rate are:

$$\frac{\text{Pressure} \times \text{volume}}{\text{Time}}$$

Leakage rate is occasionally referred to as mass flow, since at constant temperature it is proportional to mass-flow rate. In describing a leak, the nature of the leaking gas and its temperature is usually known. The actual leaked gas mass may be determined by use of the formula

$$PV = gRT/M \qquad (1)$$

where

P = pressure
V = container volume
g = mass of gas in the container
R = universal gas constant
T = absolute temperature
M = molecular weight of the gas

In scientific work, mass flow is usually expressed in units of atm cc/sec or torr liters/sec at 25° C. In engineering work, various leakage units are used. The diversity of units is justified by the convenience and ease with which measurements made in those units can be adapted to individual engineering problems. For example, consider a gas cylinder whose volume is known in units of cubic feet, and a pressure gage calibrated in units of pounds per square inch. If the gage is read daily, it is convenient to express leakage as psi ft^3/day (Δ cylinder pressure × cylinder volume/time increment), where the unit of psi refers to the change in cylinder pressure during a time interval of 1 day. The mass of gas which leaks from the cylinder in 1 day is obtainable from equation (1) ($PV = gRT/M$). Since the only variables are the pressure P and the mass of gas in the cylinder, equation (1) can be recast as

$$\Delta P = RT\,\Delta g/MV \qquad (1a)$$

where ΔP is the difference in pressure and Δg is the mass of gas leaked per day. The leakage in units of psi ft^3/day has been found to have a numerical value of Q, a constant. Thus $\Delta PV/\text{Day} = Q$. Substituting from equation (1a) gives $\Delta g = MQ/RT$, the mass of gas leaked from the cylinder in 1 day. Mass flow rate is a dimension sometimes used instead of pressure × volume/time.

Generally, leakage is first measured with a gas. This is usually easier than with a liquid because diffusion brings the gas to the detector. It is also easier to flush out a tracer gas than to remove a liquid.

The general range of interest for leakage determination is between 10 atm cc/sec and 10^{-10} atm cc/sec. This, however, is not the whole range. New York City is interested in water leakage of the order of thousands of gallons per second. Manufacturers of high-reliability vacuum tubes, on the other hand, are concerned with leakage as low as 10^{-14} atm cc/sec, since the tubes are small and must operate for long time periods. A 10 atm cc/sec leak represents loss of approximately 10 gallons of gas at atmospheric pressure per hour. Such a leak is usually audible. A 10^{-10} atm cc/sec leak represents leakage of approximately 1 cc of gas at atmospheric pressure in 300 years. This slow loss, however, may be detected readily with a mass spectrometer leak detector.

LEAKAGE MEASUREMENT COST

The cost of leak detection increases as the required sensitivity increases. The initial equipment investment cost versus leakage sensitivity is plotted in figure 1. This graph

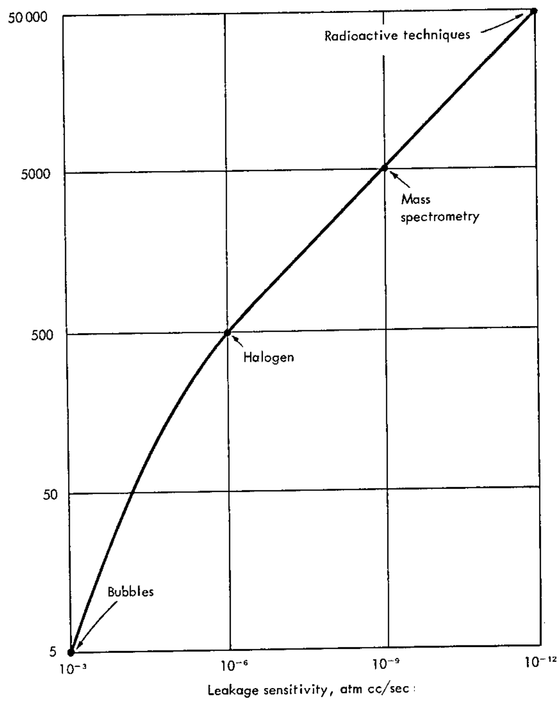

FIGURE 1.—Effect of required sensitivity on leak detection equipment cost.

shows that the investment for a leakage determination to approximately 10^{-3} atm cc/sec is about \$5, while the investment for 10^{-12} atm cc/sec determination is 10 000 times as much.

Even after a technique has been chosen, increasing the sensitivity of the measurement will usually increase measurement cost because more time is required to obtain confidence in the measurements. For example, if a component is tested by immersion in water, the required sensitivity may range from 10^{-1} to 10^{-4} atm cc/sec. To measure the 10^{-1} atm cc/sec leaks, the component may be quickly placed in the water and then removed. The bubbles coming from the pressurized component evolve at such a rapid rate that there is no question of the existence of the leak. In the 10^{-2} to 10^{-3} atm cc/sec range, care must be taken that the component has been immersed long enough for any bubbles emerging from crevices to collect and rise. In the 10^{-4} atm cc/sec range, the component must be completely stripped of attached air bubbles after immersion to detect bubbles formed by the leaking gas. The 10^{-4} atm cc/sec range is near the limit of detectability for this technique. High sensitivity becomes impractical because the time needed for bubble formation permits the leakage rate to approach the rate at which the gas dissolves in the immersion fluid.

This example shows how an increase in the sensitivity of a leakage-measurement technique is accompanied by an increase in testing time and cost. The cost is maximal when the leakage specification reads:

(1) No detectable leakage
(2) No measurable leakage
(3) No leakage
(4) Zero leakage

Such specifications are both expensive and confusing unless the method of leak testing is also prescribed. It is usually cheaper to specify rejection leakage at least a decade above the noise level of the instrument employed in the measurement technique. It is easier to determine whether or not leakage is greater than a given standard than it is to discriminate between leakage and random instrument noise. Zero leakage can then be defined as a measurable quantitative value; any smaller leakage would be insignificant in the component's operation.

MEASUREMENT ACCURACY

In specifying rejection leakages, two factors concerning the standards used need to be considered: the accuracy of the standards and the ability of the leak-detection instrument to discriminate between leakage values. Several comparisons of leakage standards were made by the General Electric Research & Development Center (ref. 1). Helium-calibrated leaks reported by various manufacturers were compared on a precision mass spectrometer. This instrument was constructed to produce a linear variation of ion current with helium leakage from the calibration standards. Figure 2 shows the data obtained. The experimental results are plotted as leakage values stated by the manufacturer versus the mass spectrometer ion

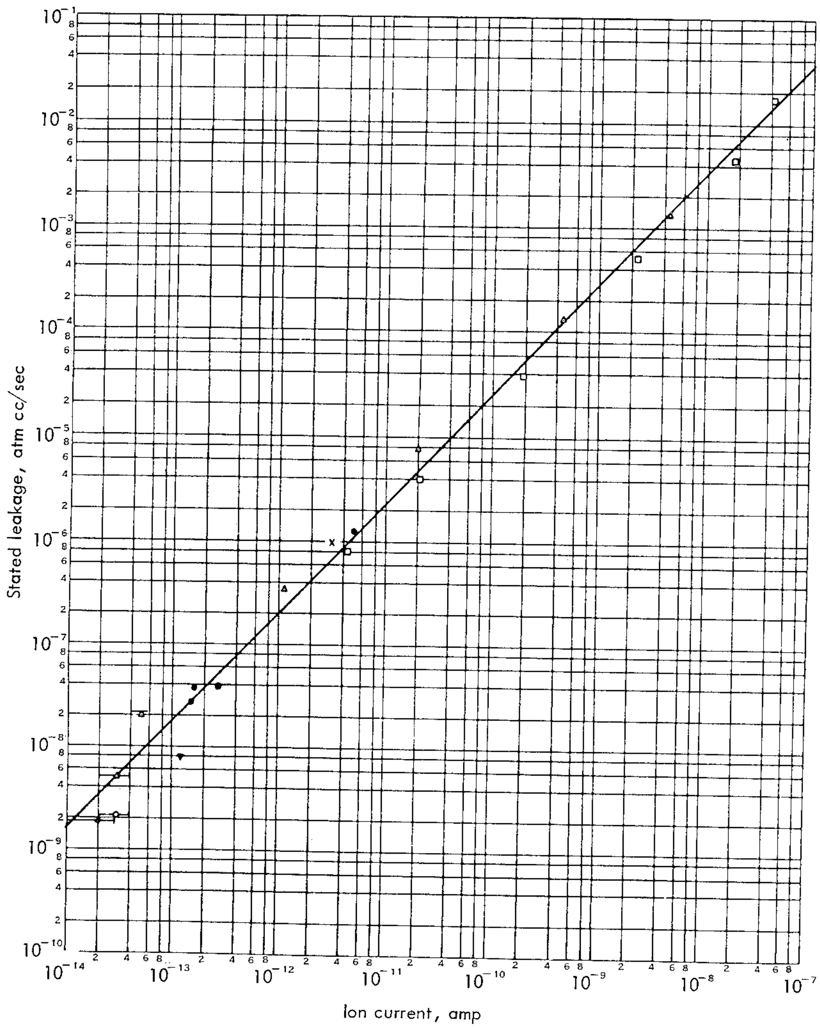

FIGURE 2.—Comparison of leaked values stated by manufacturers.

current reading. The solid line on the graph in figure 2 is the least mean square straight line fit to the data.

The values of the calibration standards vary greatly among manufacturers. These deviations are found in part because no standards exist either for pressure measurements at low pressures or for leakage. The manufacturers are forced to establish their own primary standards from which commercial secondary standards are made. Use of these standards for instrument calibration immediately increases the uncertainty of measurements, as shown by figure 2.

Unfortunately, a greater error is introduced when leaks are compared by various instruments. For the ASTM (ref. 2), numerous "standard" leaks reported by several manufacturers were tested in a roundrobin series by a number of different laboratories using helium mass spectrometer leak detectors. The purpose was to obtain data on instrument response. For any one leak of stated leakage value, considerable variation of responses among the different instruments was found. Moreover, instrument response was not linear with leakage. Figure 3 shows the varieties of instrument response when a leak was compared to a calibrated leak. Hence, even if the calibrated leak is

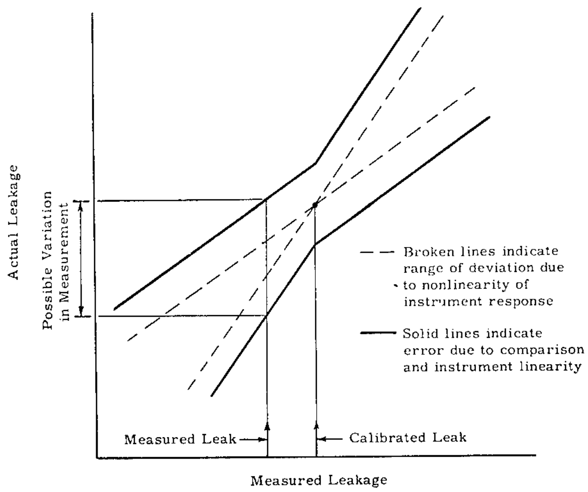

FIGURE 3.—Instrumentation error in leakage measurement.

similar in magnitude to the measured leak, there is some doubt as to the actual value of the latter. And if the measured leak is substantially different in magnitude from the calibrated leak, confidence in the measured value is still lower.

INSTRUMENTATION FOR LEAK DETECTION AND MEASUREMENT

The instruments used to detect and measure a leak are quite varied (ref. 3). Methods to be briefly described here may be placed in four categories: (1) Physical observation of leakage; (2) detection of difference in pressure due to leakage; (3) observation of change in gage reading due to tracer gas; and (4) detection of tracer gas.

Physical Observation Methods

Sonic detectors. Leak detectors are available to physically detect the sound made by the outflow of a gas under what are presumably choked flow conditions. The manufacturers' stated sensitivity is approximately 10^{-1} atm cc/sec.

Fluorescent dyes. A penetrating fluorescent dye solution is made which may be painted onto a part and then wiped away. The dye which is not removed by wiping will penetrate into any cracks and fissures, and will fluoresce under ultraviolet light and be visible. Flaws in machining and joining are made evident, but no quantitative estimate can be made as to the sensitivity of this method of leak detection.

Bubble testing by soap solution. The component to be tested is pressurized and painted with a soap solution. Bubble growth is evidence of a leak. The efficiency of this method depends upon the operator's observational skill, the ability to wet the tested component completely, the surface tension of the soap solution, and the component's complexity. Under proper conditions this method has an estimated sensitivity of 10^{-4} atm cc/sec.

Bubble testing by immersion. The component to be tested is immersed in a liquid and bubbles rising to the surface indicate a leak. This method depends upon the pressure capability of the component, the liquid's surface tension, and the component's surface area. It is not suited for large components, but modifications can be made to cover only part of an area by the liquid. This technique is probably simpler than soap solution testing because the observational abilities of the operator are not as important. The sensitivity of this method is between 10^{-4} and 10^{-5} atm cc/sec.

Halide torch leak detector. This detector consists of a propane gas torch with a copper plate in the flame and a rubber tube supplying the flame with air. The component under test is pressurized with a Freon[2] gas and the end of the rubber tube supplying air to the flame is passed over the suspected area. A leak is evident when the flame turns bright green, since a piece of copper in a flame containing halogen atoms gives off a characteristic green glow. Household refrigerator repairmen often use this technique because the Freon refrigerant acts as the tracer gas and the equipment is inexpensive. The sensitivity of this method is reported to be approximately 10^{-5} atm cc/sec.

Difference in Pressure Due to Leakage

Pressure drop in container. When a pressurized container with an attached pres-

[2] Trademark of E. I. du Pont de Nemours & Co., Inc.

sure gage is used to measure leakage rates, the sensitivity is controlled not only by the sensitivity of the pressure gage and by the magnitude of the time increment between gage readings, but also by environmental temperature changes which produce additional pressure changes.

Pressure increase in vacuum system. The leakage rate of a component placed in a vacuum system can be determined by the pressure increase of the vacuum system in a manner similar to the above. The same sensitivity comments apply, with the additional caution that proper correction for outgassing and vacuum system leakage is necessary.

Ultimate pressure obtained in a vacuum system. The tested component is placed in a vacuum system, which is then evacuated by pumping through an orifice of known dimensions. The ultimate pressure obtained in such a system is given by

$$Q = P_u S$$

where

Q = measured leakage
S = pumping speed of the system
P_u = ultimate pressure obtained

The sensitivity of this method of leak detection is dependent upon the magnitude of outgassing and the pumping speed of the system as compared with the leakage rate.

Argon leak detector. This detector consists of a vacuum system connected to the component to be tested and pumped by a getter pump. Because a getter pump is efficient for all but inert gases, argon is used as a tracer gas. When it enters the vacuum system through a leak, the pump does not getter it efficiently and the system pressure rises. This method is not readily applicable to large-volume systems because of the amount of argon which must be removed from the air in the system. Sensitivity of this method is reported to be extremely high (10^{-9} atm cc/sec).

Change in Pressure Gage Readings Due to Tracer Gas

A large number of leak-detection methods involve the use of three types of low pressure (vacuum) gages. The Pirani gage consists of a wire through which electrical current flows. The wire resistance depends upon its temperature, which in turn depends upon the thermal conductivity of the surrounding gas. For low pressures, the thermal conductivity of the gas is proportional to pressure. Thus, resistance of the wire as measured by a Wheatstone-bridge arrangement allows determination of system pressure.

The thermocouple gage uses the same principle for pressure measurement as the Pirani gage. However, in the thermocouple gage the wire current is constant and the wire temperature is measured by a thermocouple welded to it.

The ionization gage is a triode in which an electrical current passes between a cathode and a plate. The electrons encounter residual gas molecules, which become ionized. The current produced by these ions is measured as a grid current. Since this current depends upon the number of gas molecules encountered by the electron stream, and this number is proportional to the system pressure, the grid current is itself proportional to the system pressure.

Leak testing with pressure gages. Leaks into an evacuated system containing one of the three pressure gages described can be detected by observation of the system pressure. When a tracer gas enters the system, the gaseous composition within the system changes. Thus, the thermal conductivity measured by the gage changes. This change can be interpreted as a tracer gas leaking into the system and can be used to determine that there is leakage. Many tracer gases can be used with this technique. If a thermocouple or Pirani gage is used, hydrogen or helium, because of their high thermal conductivities, indicate a large rise in pressure upon entering the system. The indicated pressure rises because of two effects: the actual pressure rise due to the influx of the tracer gas and the indicated increase due to its high thermal conductivity. Alternatively, acetone or alcohol can be used as the tracer fluid by painting the high-pressure side of the suspected area with a brush. A pressure decrease will occur because of clogging of the leak by the liquid. A drop in pressure due to clogging and an indicated pressure drop due to the low thermal conductivity of the entering tracer are produced. With an ionization gage, the ionization efficiency in the tracer gas should be different from that of the air.

Hydrogen sensitive leak detector. This detector consists of an ionization gage connected to the vacuum system by a heated palladium diaphragm. The tracer gas, hydrogen, permeates through the palladium to the ion gage. This system is relatively insensitive to pressure changes. It has a reported sensitivity of about 10^{-8} atm cc/sec.

Halogen leak detector. Platinum at red heat emits positive ions in air. The emission at any given temperature is markedly increased when vapors containing a halogen strike the electrode surface. This phenomenon provides the basis of the halogen leak detector, of which two types are available. One consists of a detector mounted in the vacuum system and the halogen gas, usually a Freon, used as a tracer outside the system. A second type consists of a small gas pump probe which contains the detection apparatus. The system to be leak tested is pressurized with a halogen-containing gas. The sensitivity of this method is approximately 10^{-6} atm cc/sec; special units are as sensitive as 10^{-9} atm cc/sec.

Electron emission leak detector. Electron emission from a tungsten- or oxide-coated filament is greatly diminished by contact with oxygen. This phenomenon has been applied to leak detection, and the device is reported to have a sensitivity of approximately 10^{-8} atm cc/sec when oxygen is used as the tracer gas.

Detection of Tracer Gas

Helium leak detector. A very sensitive leak-detection method involves the use of a mass spectrometer and helium as the tracer gas. The mass spectrometer is attached to the vacuum system, and a probe of helium gas is passed around the suspected leak area. Detection and measurement consists of observation of the ion current intensity, with the mass spectrometer set directly on the helium mass peak. Helium is chosen as the tracer gas because its low molecular weight gives a greater leakage rate through a given leak than any other gas except hydrogen, it occurs in the atmosphere only to the extent of 1 part per 200 000 parts of air, and there is no possibility that an ion of any other gas will give an indication that can be mistaken for helium. The sensitivity of this technique is about 10^{-10} atm cc/sec. A helium leak detector operating on a pressure buildup by getter-ion pumping prior to detection of the peak is commercially available. Its sensitivity is claimed to be 10^{-14} atm cc/sec. Because the vacuum equipment in this apparatus is similar to that of the previously described argon leak detector, this instrument may be used as a binary leak detector. For gross leaks, the pressure buildup may be measured by an argon probe. After all the large leaks have been detected, helium gas may be used to detect small leaks.

Helium probe. The helium leak detector can also be used as follows: the tested component is pressurized with helium, a vacuum probe collects air from outside the supected leak area, and this air is passed into the vacuum system containing the mass spectrometer. This method has a sensitivity of approximately 10^{-6} atm cc/sec.

Leak detection by radioactivity. The basic principle of leak detection by radioactivity involves detection of radioactive krypton 85 which has been allowed to diffuse into the leaking component. The component is placed in an activating tank which is then sealed. Air is evacuated from the tank and krypton 85 is pumped into the tank to a pressure of 7 atmospheres. It diffuses into the leaks in the component. After a prescribed soaking period, the krypton is pumped out and returned to storage for reuse. Air is circulated over the component to remove residual krypton 85 from the outer component surfaces, and the component is removed from the tank and tested with a scintillation counter. Under favorable conditions, leakage rates of the order of 10^{-12} atm cc/sec can be measured.

Methyl orange indicator paper. This has been used to detect oxidizer leaks in the Ranger spacecraft propulsion system (ref. 4). The indicator paper changes color when contacted by an oxidizer. The reported sensitivity is 10^{-5}–10^{-6} atm cc/sec for detection of nitrogen tetroxide (N_2O_4).

Mass Leakage Analyzer

A leakage measurement device developed by TRW Systems (refs. 5 and 6) does not fall conveniently into the four preceding instrumentation categories. It measures, independently of temperature and pressure, leakage of both gases and liquids. The operating principle is illustrated in figure 4 and the device itself is shown in figure 5. A gas injected into the initially evacuated chamber test volume, V_c, causes the chamber density to increase, resulting in an increased buoyancy force, F, on the evacuated sphere of volume, V_b. If the leaking fluid is a liquid, it will vaporize because of the low chamber pressure relative to the vapor pressure of the liquid. Measuring the buoyancy force as a function of time allows the leakage, $\dot{M}$, to be established from the relation $\dot{M} = (V_c/V_b)\dot{F}$. The leakage is given by this relation in units of mass/unit time.

The sensitivity of this device is determined

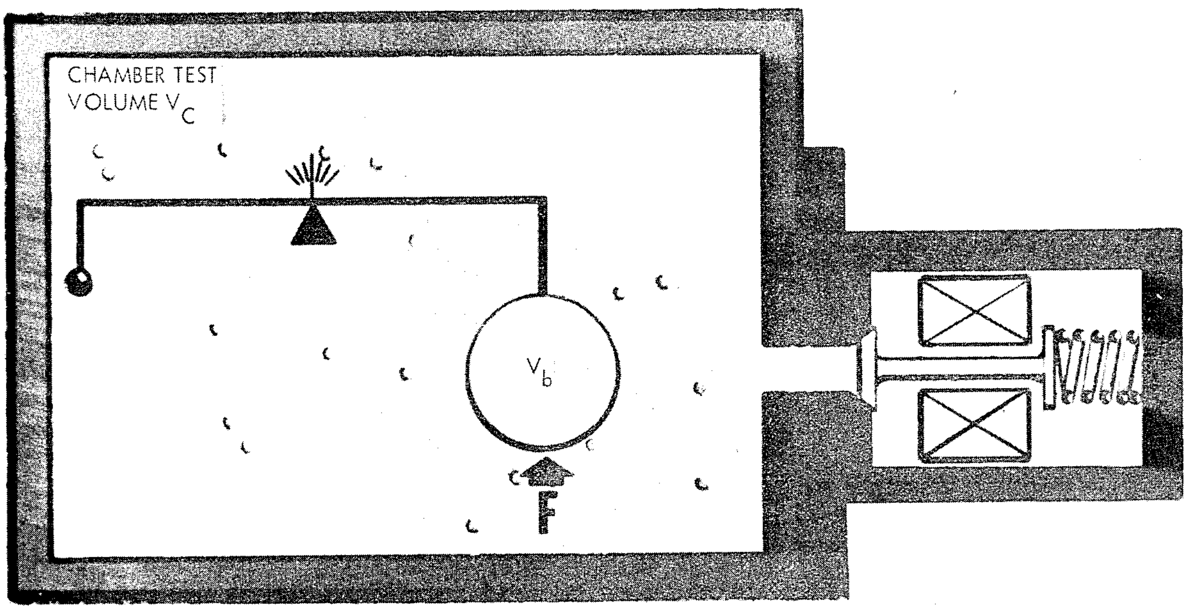

FIGURE 4.—Operating principle of mass leakage analyzer with test valve. (Courtesy of TRW Systems.)

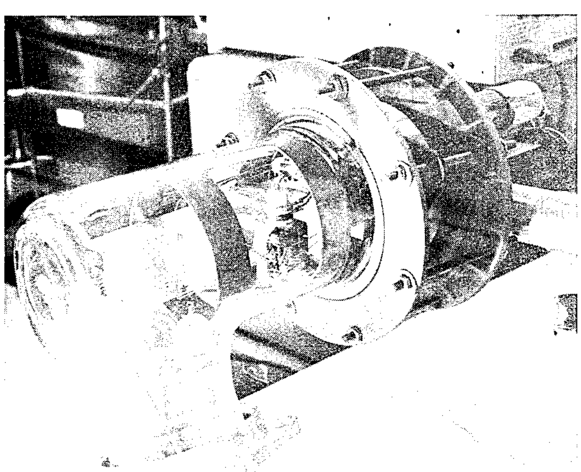

FIGURE 5.—Mass leakage analyzer showing electrobalance glass ball and glass enclosure. (Courtesy of TRW Systems.)

by the balance sensitivity and the volume ratio, V_c/V_b. The smallest increment of mass presently measurable is 10^{-5} gram, which is equivalent to a leakage volume of 5×10^{-2} standard cc of helium. Improved models will have a sensitivity of 10^{-9} gram. This device basically measures leaked mass, whereas a mass spectrometer measures leakage rate. It is claimed that the mass leakage analyzer can conveniently simulate a variety of downstream conditions, such as vacuum, gaseous environment, or planetary atmospheres, within a wide range of temperatures and pressures.

FLOW MODES

Mass transfer attributed to leakage can occur either as permeation or as pneumatic flow. Permeation is passage of a fluid into, through, and out of a material barrier having no holes large enough to permit more than a small fraction of the fluid molecules to pass through any one hole. The process usually involves diffusion through a solid and may involve other phenomena, such as adsorption, disassociation, migration, and disorption. It is significant for two reasons: it may be a source of error in leakage measurement, and it represents a leakage mechanism in critical component applications. Pneumatic flow occurs when leakage is by passage of fluid through finite holes in the component. The general formula for permeation is

$$Q=K_p A \frac{\Delta P}{l} \tag{2}$$

where

Q = volumetric rate of flow
K_p = permeation rate constant
A = area normal to permeation
ΔP = pressure drop along the permeation path
l = length of flow path

ΔP represents the difference in partial pressure of the leaking fluid between the two sides of the barrier.

The permeation rates for some characteristic materials are given in table I. Additional data may be obtained in larger tabulations and the original references mentioned therein (ref. 7).

If a system is to be leaktight, the construction materials must exclude leakage by permeation. As an example, consider the leakage rate of a natural rubber gasket at room temperature 0.1 inch thick, 0.1 inch wide, and 5 inches in diameter with a 1-atmosphere hydrogen pressure differential. As determined by equation (2), with a value

TABLE I.—*Permeation Rates*

[From ref. 7]

Permeation rate constants (cm^3-mm/sec-cm^2-atm) at room temperatures

	Hydrogen	Oxygen	Nitrogen	Helium
Iron	4 to 15 $\times 10^{-9}$		4.2 $\times 10^{-19}$	
Steel (low carbon)	3 $\times 10^{-10}$			
Steel (27 percent chrome)	1 $\times 10^{-12}$			
Aluminum	3.1 $\times 10^{-22}$			
Natural rubber	1.5 to 4.0 $\times 10^{-6}$	1.8 $\times 10^{-6}$	1.0 $\times 10^{-6}$	3.0 $\times 10^{-6}$
Neoprene	1.0 $\times 10^{-7}$	0.3 $\times 10^{-7}$	0.1 $\times 10^{-7}$	0.6 $\times 10^{-7}$

Permeation rate constants (cm^3-mm/sec-cm^2-atm) at various temperatures

	75° F	176° F	350° F
Butyl rubber	0.02 $\times 10^{-6}$	0.32 $\times 10^{-6}$	6.1 $\times 10^{-6}$
Silicone rubber	22.0 $\times 10^{-6}$	45 $\times 10^{-6}$	112 $\times 10^{-6}$
Kel-F		0.8 $\times 10^{-6}$	

of $K_p = 4 \times 10^{-6}$ cm mm/sec atm from table I, the leakage rate is 1.6×10^{-5} cc/sec. In some uses, this permeation might represent an unacceptable leakage rate.

Permeation presents a problem in components in which the construction material has a high permeability to the tracer gas. For example, if a component containing a rubber diaphragm 1-millimeter-thick and of 1-square-inch surface area is leak checked using helium gas, a leakage of approximately 10^{-5} atm cc/sec will be measured. This is due to permeation of helium through the diaphragm rather than to actual holes. It represents the maximum sensitivity of helium leak testing that can be performed on this component. However, if the component is to be used with a fluid to which the membrane is impermeable, the apparent leakage due to permeation shown during the leak testing has little meaning under operational conditions.

Another example of a situation in which a false reading can be taken is leak testing of a rubber O-ring. Depending on material, a rubber O-ring usually represents a permeability of approximately 10^{-6} to 10^{-7} atm cc/sec per linear inch of exposed surface. However, this permeability does not have to be taken into consideration during routine leak checking, since this leakage measurement occurs too rapidly to permit the saturation and diffusion of helium through the O-ring.

To eliminate permeation as a factor in leakage measurement, three procedures may be used. First, the measurement may be taken rapidly so as not to allow the material to be saturated with gas. This is possible only if the material is relatively thick. For example, a rubber diaphragm will saturate rapidly and almost immediately show leakage. On the other hand, the O-rings are relatively thick and will not saturate rapidly enough to give a reading within a reasonable period of time (5 minutes). The second procedure is to account for the permeability by calculating the maximum permeability of all components and the mass transfer it will produce during leak testing. In this way, the permeation value will be known and only leakage above this value will be taken into consideration as pneumatic flow. The last and most difficult method is to measure the leakage quantitatively at various pressure differentials. If the leak being measured is pneumatic (i.e., through a finite hole in the component), it will be proportional to the square of the pressure differential across the leak, if the flow is laminar. However, if the flow is strictly due to permeation, the flow through the leak will be directly proportional to the pressure differential. In this way, the leakage due to holes in the component can be differentiated from that due to permeation.

Occurrence of the Flow Types

Pneumatic gas flow which occurs in a leak may be placed in three categories: turbulent, laminar, and molecular. Table II shows the approximate region of the three types of flow modes: above 10^{-2} atm cc/sec for turbulent, 10^{-1} to 10^{-6} atm cc/sec for laminar, and below 10^{-5} atm cc/sec for molecular types of flow. As can be seen from this table, the predominant flow mode is laminar for leakage rates of primary interest.

Turbulent Flow

Above a critical value of the Reynolds number (about 2100 in the case of circular pipe flow), flow becomes unstable, resulting

TABLE II.—*Properties of the Various Flow Modes*

Type of flow	Turbulent	Laminar	Molecular
Leakage region	$> 10^{-2}$ atm cc/sec	10^{-1} to 10^{-6} atm cc/sec	$< 10^{-5}$ atm cc/sec
Flow-pressure relationship	$Q \alpha \Delta P$	$Q \alpha\ (\Delta P)^2$	$Q \alpha \Delta P$
Flow-gas property relationship	$Q \alpha \sqrt{\frac{1}{M}}$	$Q \alpha \frac{1}{\eta}$	$Q \alpha \sqrt{\frac{1}{M}}$

in innumerable eddies or vortices in the flow. Any particle in turbulent flow follows a very erratic path, whereas in laminar flow the particle travels in a smooth path.

The laws for turbulent flow are quite different from the laws for laminar flow. The equation expressing mass flow rate (Q) in units of pressure×volume/time may be written

$$Q = \pi d^{5/2} \left(\frac{RT \ (P_2^2 - P_1^2)}{16 f M} \right)^{1/2} \qquad (3)$$

where f, the friction factor, is defined by Hunsaker et al. (ref. 8). The friction factor depends on roughness of the channel walls, and can be considered a constant in fully developed turbulent flow.

Turbulent flow, because it requires relatively high velocity, occurs only in leaks larger than 10^{-2} atm cc/sec. This has been demonstrated both mathematically (ref. 9) and experimentally (ref. 10).

Laminar Flow

Laminar flow of a fluid in a tube is defined as a condition in which the velocity distribution of the fluid in the cross section of the tube is parabolic. Laminar flow is one of the two classes of viscous flow; the other class is turbulent flow. Because turbulent flow is rarely encountered in leak detection work, the term "viscous flow" is sometimes incorrectly used solely to describe laminar flow.

The most familiar laminar-flow equation was developed by Poiseuille (ref. 11) for flow of constant temperature gas through a long straight tube of circular cross section:

$$Q = \frac{8}{\pi} \left(\frac{d}{2} \right)^4 \frac{P_a}{\eta l} (P_2 - P_1) \qquad (4)$$

where

Q = leakage rate
d = diameter of the tube
l = length of the tube
P_2 and P_1 = pressures at the two ends
η = viscosity of the gas
P_a = arithmetic mean of P_2 and P_1

In the range where this equation is applicable, it has been substantially verified experimentally. The above equation is applicable where the length and diameter of the flow passage are known. This is not the case for most leaks, but the equation may be rewritten in the form:

$$Q = K \frac{P_a (P_2 - P_1)}{\eta} \qquad (5)$$

where K represents the constants and the two geometric terms, d and l, of equation (4).

The two most important characteristics of laminar leaks are that the flow is proportional to the square of the pressure difference across the leak and that the leakage is inversely proportional to the leaking gas viscosity. As may be seen in table III, the viscosities of most gases are similar. Therefore, a change of gas will not markedly increase the sensitivity of the leak detection method unless this change of gas implies a change of instrument sensitivity. However, as shown in figure 6, increasing the pressure difference across the leak by a factor of a little over 3 will increase the flow rate through this leak by a factor of 10.

TABLE III.—*Viscosity of Gases at 0° C*

[From ref. 12]

Gas	Viscosity, μP	Viscosity of gas divided by viscosity of ether
Acetylene	93.5	1.38
Air	170.8	2.51
Ammonia	91.8	1.35
Argon	209.6	3.09
Carbon dioxide	139.0	2.05
Cyanogen	92.8	1.37
Ethane	84.8	1.25
Ether (diethyl)	67.8	1.00
Helium	186.0	2.74
Hydrogen	83.5	1.23
Krypton	232.7	3.43
Neon	297.3	4.38
Nitrogen	166.0	2.45
Oxygen	189.0	2.79
Xenon	210.1	3.09

Obviously then, when the leaks to be measured are in the laminar flow range, the simplest method of increasing detection sen-

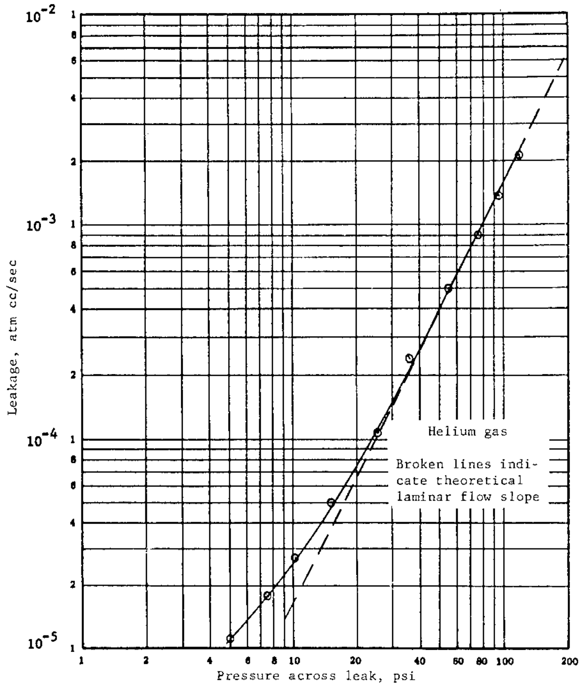

FIGURE 6.—Laminar flow in typical hardware leak.

sitivity is by an increase of pressure across the leak.

Molecular Leaks

Molecular flow occurs when the mean free path of the leaking molecules is greater than the largest dimension of a transverse section of the leak. The mean free path is the average distance that a molecule travels between successive collisions with other molecules of an ensemble and is given by the relation (ref. 13).

$$L=8.589\ \frac{\eta}{P}\frac{T}{M}$$

where

L = mean free path, cm
P = pressure, torr
η = viscosity, poise
T = absolute temperature, °K
M = molecular weight

For air at 20° C, this relation gives

$$L=5\times10^{-3}/P$$

which indicates that leak dimensions for molecular flow are quite small for all but very low pressures.

The original mathematical derivations of molecular flow are attributed to Knudsen (ref. 14). The leakage rate in a long pipe of circular cross section is

$$Q=\sqrt{\frac{2\pi}{6}}\sqrt{\frac{RT}{M}}\frac{d^3}{l}\ (P_2-P_1) \qquad (6)$$

Equation (6) is not directly applicable in most leakage situations because the leak length and diameter are not known. However, most leaks are long compared to their effective diameter so that the indicated dependence of leakage rate on pressure difference and gas properties is of general utility.

Figure 7 is an example of molecular flow and its transition into laminar flow; the slope of the pressure-leakage curve is one in the molecular flow region, and increases to two when laminar flow predominates.

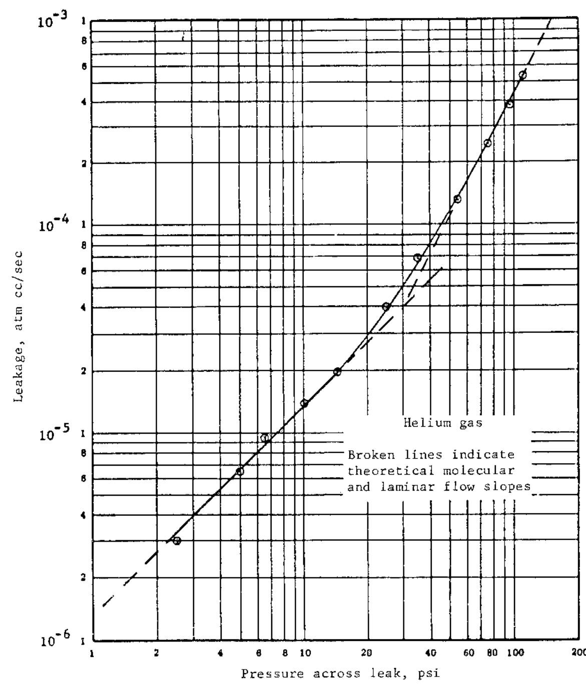

FIGURE 7.—Transition flow in typical hardware leak.

GAS-FLOW CONSIDERATIONS

The laminar-flow mode is the most common one in leaks. The other two modes of flow are encountered less often, although turbulent flow occurs in exceptionally gross leaks. Molecular-flow leaks, on the other hand, are only of interest when extremely precise equipment is being constructed. Both of the latter two types of leaks have the same characteristics; flow is directly proportional to the pressure across the leak and inversely proportional to the square root of the molecular weight of the leaking gas.

Using the information previously discussed, several rules can be stated concerning leakage correlation between a tracer gas and the operating fluid:

(1) In the laminar-flow regime no correction for change of gas is necessary. The leakage is inversely proportional to the viscosity of the gas. However, as seen in table III the ratio of viscosity of most gases is less than the variations usually encountered in leakage measurement.

(2) A correlation for molecular flow can be made using the relationship that the leakage is inversely proportional to molecular weight. This correlation is only applicable to leaks smaller than 10^{-6} atm cc/sec leaking into a vacuum.

(3) Although the pressure-leakage relationships are known, these should not be used for leakage correlation because of pressure-induced distortions of leak geometry (see later discussion of anomalous leakage).

LIQUID LEAKAGE

The leakage correlation of greatest interest is the one that predicts liquid leakage from measured gas leakage. One possible correlation method is to calculate the conductance from the measured gas leakage, and to use this conductance to calculate the liquid leakage. Although no experimental proof of this technique is available, experiments at the General Electric Research & Development Center, supported by NASA Contract NAS 7–102 (ref. 15), are attempting to prove the validity of such a technique.

The Poiseuille equation for liquid leakage is

$$Q_v = \frac{\eta}{8}\left(\frac{d}{2}\right)^4 \frac{P_2 - P_1}{\eta l} \qquad (7)$$

In this equation, Q_v is the leakage of liquid expressed in volume per unit time. This equation differs from that for laminar flow of a gas by the lack of the modification due to gas compressibility.

Both the equations for flow of gases

$$Q = \frac{\pi}{8}\left(\frac{d}{2}\right)^4 \frac{P_a}{\eta l}\ (P_2 - P_1) \qquad (4)$$

and for liquids

$$Q_v = \frac{\pi}{8}\left(\frac{d}{2}\right)^4 \frac{P_2 - P_1}{\eta l} \qquad (7)$$

have the similar geometry factor K where

$$K = \frac{\pi}{8}\left(\frac{2}{d}\right)^4 \frac{1}{l} \qquad (8)$$

Substituting the geometry factor reduces the equations to

$$Q = \frac{K}{\eta} P_a\,(P_2 - P_1) \qquad (5)$$

for gases, and

$$Q_v = \frac{K}{\eta}\,(P_2 - P_1) \qquad (9)$$

for liquids. In use, the equipment leakage is measured and equation (3) is solved for K. Once calculated, K is then used in equation (9) to predict the liquid leakage. Such a procedure will be accurate only in the case of laminar flow leaks. An example of such a procedure is shown in table IV. Should the measured gas leakage be molecular rather than laminar, the error introduced in the calculation will predict a greater liquid leakage than will actually be found. The equation may be used with confidence, since any error will introduce a margin of safety into the results.

If the ratio of liquid to gas leakage through the same leak at the same operating pressures is desired, equations (5) and (9) may be combined to obtain

$$\frac{Q}{Q_v} = \frac{\eta \text{ liquid}}{\eta \text{ gas}} P_a \qquad (10)$$

The ratio of liquid to gas leakage decreases as the operating pressure increases.

In other words, the higher the operating pressure, the smaller the volume of liquid going through a hole will be compared to the amount of gas leakage. The conclusion is that the most liquid-tight system is the one that successfully passes the leak test at the highest pressure.

TABLE IV.—*Calculation of Liquid Methyl Alcohol Leakage From Measured Gaseous Helium Leakage*

Measured values of helium leakage are:

$$Q = 5 \times 10^{-3} \text{ atm cc/sec}$$
$$P_2 = 15 \text{ psi}$$
$$P_1 = 10^{-4} \text{ psi}$$

A 5×10^{-3} leak should be laminar as indicated in table II. Therefore, equation (5) is used for evaluation of the geometric factor, K. Values of $P_a = 7.5$ psi and $\eta = 1.8 \times 10^{-2}$ centipoise from table III for helium viscosity when substituted into equation (5) give

$$Q = K\,P_a(P_2 - P_1)/\eta$$

$$5 \times 10^{-3} \text{ atm cc/sec} = K(7.5 \text{ psi})(15 \text{ psi})/1.8 \times 10^{-2} \text{ centipoise}$$

$$K = 8 \times 10^{-7} \text{ atm cc centipoise/sec (psi)}^2$$

Liquid methyl alcohol leakage at 100-psi pressure differential is calculated from equation (9). Using a value of $\eta = 0.6$ centipoise for methyl alcohol and the previously determined value of K for substitution into equation (9) give

$$Q_v = K(P - P)/\eta$$

$$Q_v = (8 \times 10^{-7} \text{ atm cc centipoise/sec psi}^2)(10^2 \text{ psi})(14.7 \text{ psi/atm})(0.6 \text{ centipoise})^{-1}$$

$Q_v = 1.96 \times 10^{-3}$ cc/sec predicted methyl alcohol leakage

94.7 cc/day

ANOMALOUS LEAKAGE

The previous discussion shows one reason for testing at as high a pressure as possible to obtain a liquid-tight system. A second reason is that the amount of gas leaking by laminar flow increases proportionally to the square of pressure in the system. Figure 7 is a typical curve of leakage into a vacuum versus pressure differential. In such a laminar leak, it is possible to predict leakage at a higher pressure from a leak at any lower pressure simply by extrapolation. However, in the case of a typical leak in an improperly seated gasket (fig. 8), accurate prediction of leakage is not possible. Not only does the leakage increase with pressure with a slope of two as expected in laminar flow, but there is an additional increase caused by two other factors. The size of the leak expands because of pressure increase, and the leak goes through a process of cleaning itself which irreversibly enlarges the hole. Figure 9 shows this self-cleaning process.

This experiment was performed in such a way that hole expansion due to pressure increase could not occur. The first pressure rise cleans the leak and forms a larger hole. However, upon reversal and return of the original pressure, the pressure-leakage relationship remains normal. Because of these phenomena, it is advisable to measure leakage of all components at the expected operating pressure.

To summarize some of the principles discussed, let us consider a leak specification:

This component is to leak no more than 5×10^{-5} atm cc/sec at 300 psi pressure differential.

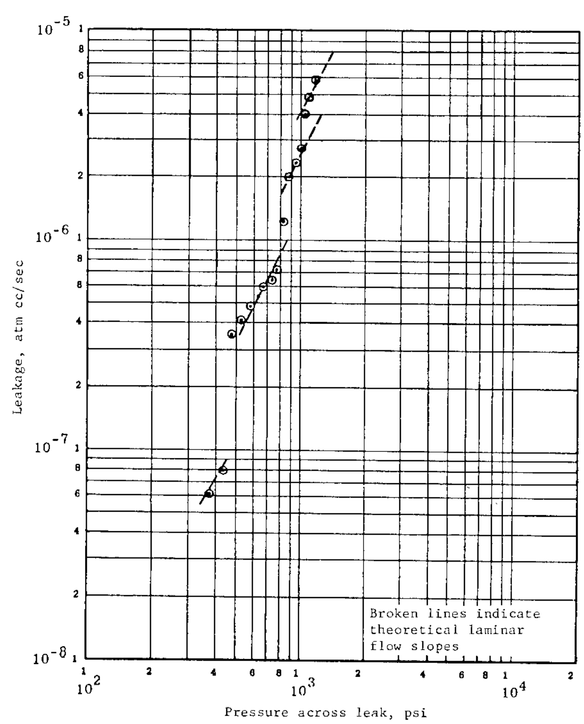

FIGURE 8.—Gasket leak.

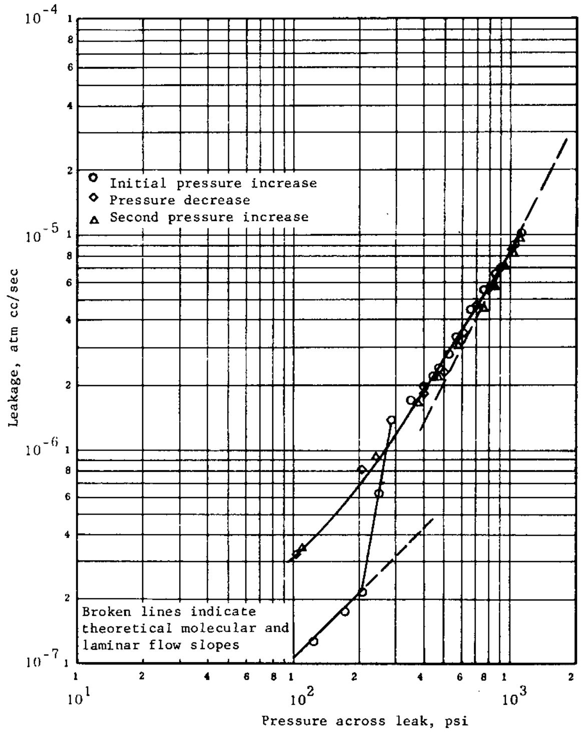

FIGURE 9.—Self-cleaning effect in gasket leak.

The accuracy of this specification is low. The numerator only specifies whether leakage is in the beginning or the middle of the particular decade of leakage. More accuracy would have little meaning because of the inadequacy of the standards and the calibration of the measurement instruments. The test gas is not specified, since a leak of this size would be of laminar nature; therefore, a correction for different test gases is minor. The pressure at which the test is to take place, however, has to be specified, since it is of primary importance in determining the leakage rate. This test pressure should be the maximum operating pressure, since it has been shown that extrapolation to a higher operating pressure is difficult.

Prior to setting this specification, calculations should be made to make sure that the specified leakage is much greater than permeation through the component.

To choose the appropriate leakage measurement method, equipment investment costs should be balanced against time required for testing.

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CHAPTER 3

Materials

A discussion of valve materials could touch upon almost every area of valve design and fill many volumes. This chapter is limited to a brief exposition of some important factors in materials selection. Specific applications of valve materials are covered in other chapters.

TEMPERATURE CONSIDERATIONS

The operating temperatures discussed under this heading are those of the flowing media, and are considered independently of temperature effects created by environments.

Cryogenic temperatures are considered to be in the range from −100° F to absolute zero. This range encompasses a number of gases and liquids used in space vehicles. These fluids are becoming more and more common in industry. They are manufactured and supplied by commercial organizations. The values for boiling points (at 1 atmosphere of pressure) for some of the gases used in space vehicles are:

	°F		°F
Oxygen	−297	Nitrogen	−320
Argon	−302	Hydrogen	−423
Fluorine	−306	Helium	−452

The effects of cryogenic temperatures on valves and valve materials may be summarized as follows: (1) dimensional changes in critical subcomponents, such as seats and seals; (2) greatly increased viscosity in lubricants, with conventional lubricants reaching the solid state; (3) changes in the structural properties of materials, with some properties being enhanced and some being degraded; and (4) contamination resulting from the solidification of gases.

Moderate temperatures are defined as the range from −100° to +400° F. If devices such as gas generators are excluded from consideration, the temperature range of the various flow media encountered on spacecraft using storable propellants may vary from approximately −100° to +400° F.

High temperatures are defined as those from +400° F up. Requirements for valves to operate in this range arise in hot, liquid metal systems and from the use of products of combustion, obtained either from rocket engine exhaust or from gas generators that burn fuels. The effects produced by these hot gases vary with the materials being used and the duration of exposure. The significant effects are:

(1) Metals lose strength as the temperature is raised. Additional material must therefore be provided for operation at high temperature, thus incurring a weight penalty.

(2) Extreme changes in temperature can cause relatively large changes in dimensions, which may be critical for a given part. If severe temperature gradients exist across a valve, seizure of movable parts can occur because of differential expansion.

(3) Rocket exhaust and gases may contain large quantities of particle contaminants. These contaminants, in conjunction with gases that may be corrosive in nature, can produce severe erosion and corrosion of surfaces on which they impinge.

The *strength of materials* as a function of temperature must be given prime consideration in valve application and selection. Figure 10 illustrates the large change in tensile and yield strengths of several metals with temperature variation.

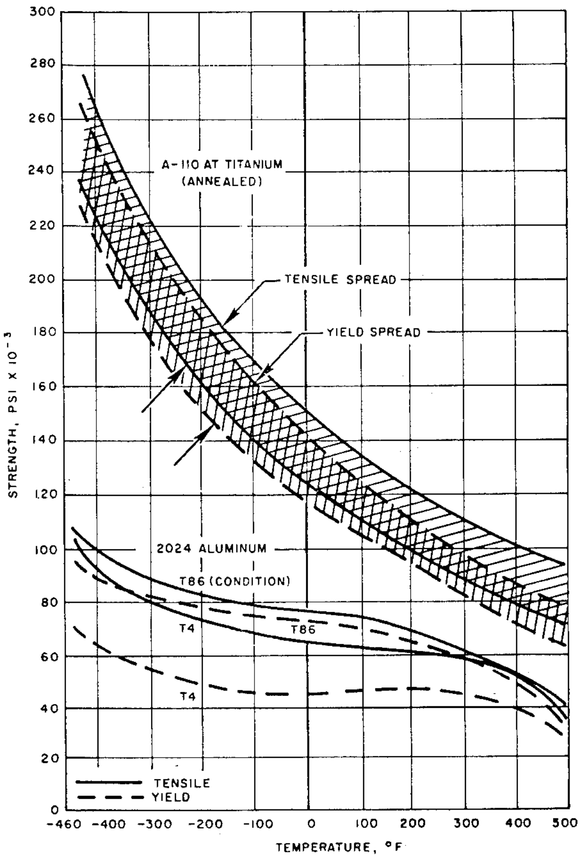

FIGURE 10.—Strength as a function of temperature.

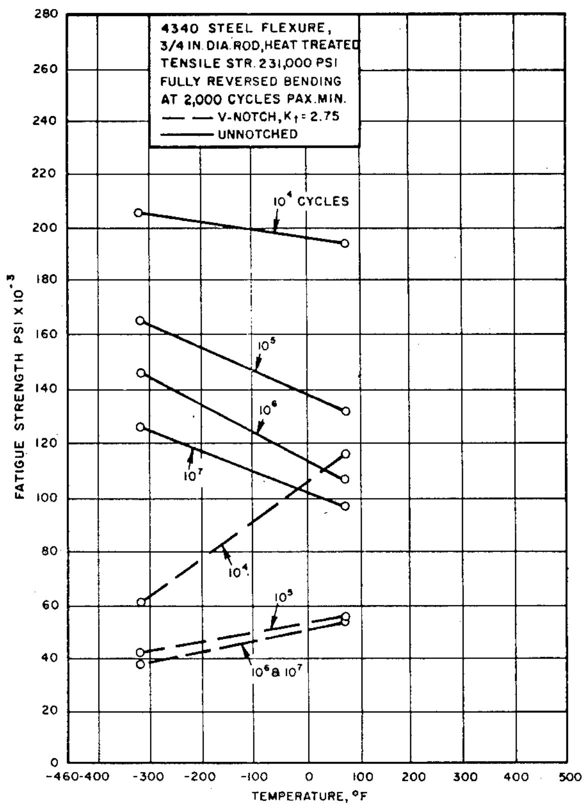

FIGURE 11.—Fatigue strength as a function of temperature.

The *fatigue strength* of a material as a function of temperature is illustrated in figure 11. The absence of an oxidizing atmosphere and the loss of the initial oxide films on the surface of structural parts may influence the fatigue strength of the parts. Investigators have found that the fatigue life of many metals increased substantially when the metals were tested in a vacuum, as compared to tests performed in air.

Elongation and *reduction of area* as functions of temperature are shown in figures 12 and 13. These characteristics may be of some significance in the design of rupture or breakaway types of valves.

Figure 14 illustrates the relationship between hardness and temperature. Caution should be exercised in designing to the increased hardness which occurs at cryogenic temperatures unless a valve or valve parts

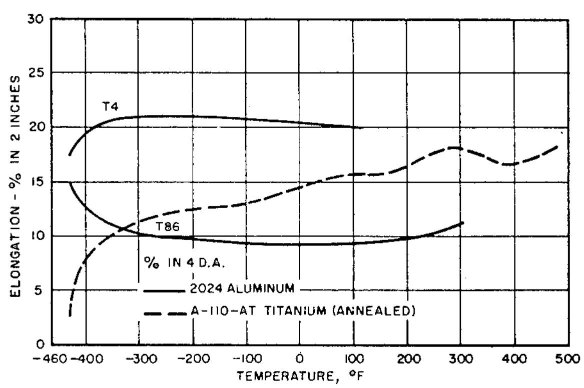

FIGURE 12.—Elongation as a function of temperature.

will be exposed to these conditions indefinitely. If the initial hardness level of a material was obtained through heat treatment, an annealing effect will occur with temperature cycling. This effect is particularly troublesome with springs. It can cause

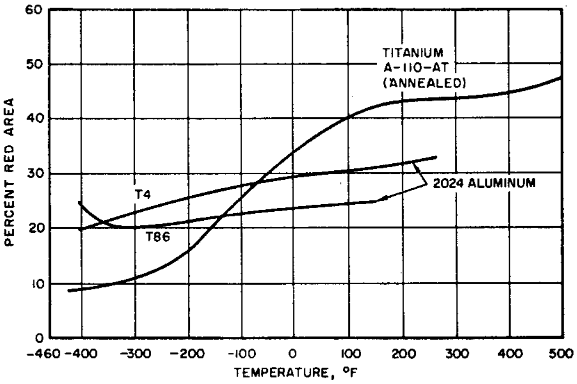

FIGURE 13.—Reduction in area as a function of temperature.

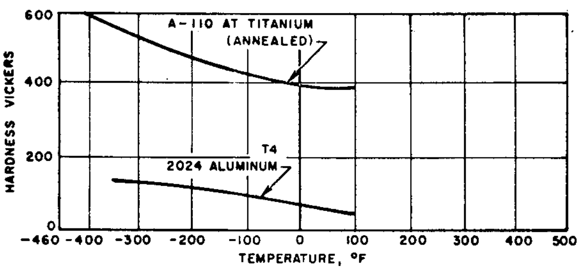

FIGURE 14.—Hardness as a function of temperature.

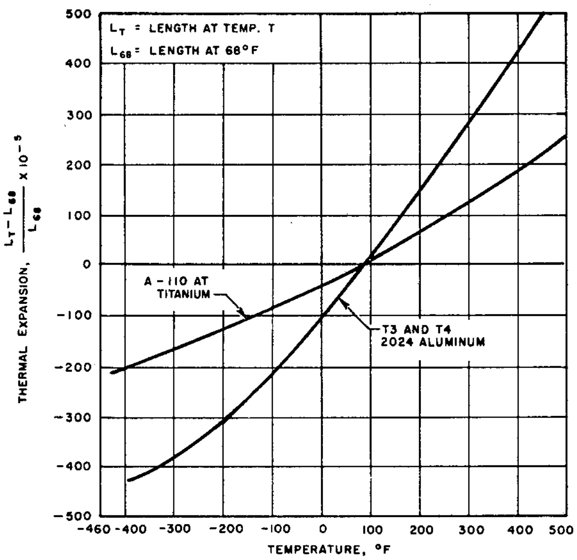

FIGURE 15.—Thermal expansion as a function of temperature.

springs and metal diaphragms to take a permanent set. A problem at Marshall Space Flight Center involved springs used in valve position indicators which suffered from this effect.

Furthermore, temperature cycling of valves will relieve the stresses left in valve parts during their manufacture. This results in warpage and out-of-tolerance parts. Sliding parts may bind and fail to operate; valve seats can develop serious warpage problems. Before valves are flight qualified for use in the Saturn V program, it is common practice to anneal all valve parts, reassemble, and test each valve.

Test programs are presently underway at Marshall Space Flight Center to investigate the use of new materials for springs. At present, Inconel X appears to be a superior spring material for cryogenic use.

The *thermal expansion* of metals should be thoroughly investigated for either cryogenic or hot valves. Very critical working clearances and tolerances exist for most valves operating in these extreme temperature ranges. The valve designer must recognize the differences in thermal expansion of various materials as illustrated in figure 15. Personnel at several NASA installations have stated that valves which are completely satisfactory at extremely high or low temperature may exhibit extremely poor qualities or not function at all at room temperature.

Various ball-and-seat closure configurations are in use to achieve essentially zero-leak seals. The ball-type poppet appears to be a good answer to problems involving thermal expansion, warpage, and annealing of built-in stresses. The present state of the art is excellent for manufacturing and producing extremely good surface finishes on spheres. Geometrically, a sphere is less susceptible than other shapes to physical distortion due to temperature changes. Further, hollow spheres exhibit better dimensional stability properties than solid spheres. In extremely critical applications, ceramic poppet balls have been substituted for both solid and hollow metal balls because the ceramic balls retain the precision spherical geometry better at low temperatures. Ceramic balls outlasted stainless-steel balls in one application by a 5:1 ratio.

The temperature versus *vapor-pressure* relationship for a number of materials is shown in figure 16. The effects of hard vacuum on both industrial and space systems that must be considered include sublimation, evaporation, and vapor pressure of materials.

PRESSURE CONSIDERATIONS

Obviously, a valve must withstand the maximum operating pressures to which it will be subjected. Overpressure from testing and water-hammer phenomena should be considered. In cases where critical tolerances exist, care must be exercised to assure that pressure-induced strains will not adversely affect operation. If a valve is subjected to many pressurizing cycles, fatigue failure is a possibility. The internal pressure will govern such aspects of design as the types of seals to be used. When this pressure is high enough to require significant actuating forces, a balanced poppet design may be required. For saturated vapors such as steam, temperature and pressure are related. Hence, specification of internal pressure can influence the choice of

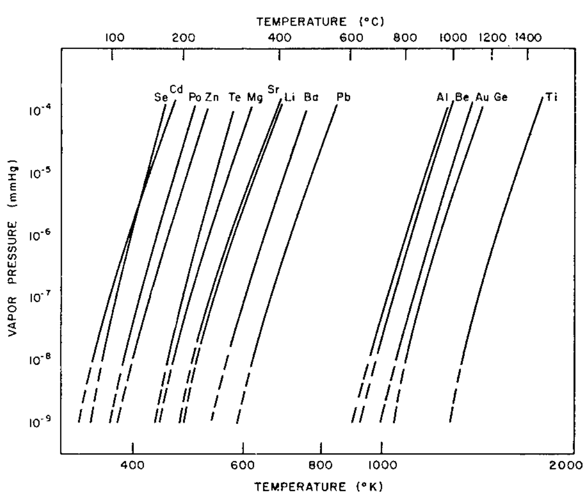

FIGURE 16.—Vapor pressure as a function of temperature.

materials through their temperature-strength relations.

In addition to internal pressure, external or ambient pressure must be considered if a valve or system is to operate in atmospheres such as those of Venus or Jupiter. The atmosphere of Venus is estimated to be 235 psi, and that of Jupiter several thousand psi. In such atmospheres, the initial system pressure would have to be higher by the increment of the planetary ambient pressure to permit overboard discharge of a gas, as is done with an altitude-control valve. If system pressure were to be depleted prior to entry into a high-pressure planetary atmosphere, the structural design of the tanks, valves, tubing, etc., would have to be such that components would not collapse because of the high external ambient pressure. Care must be taken to insure that seals used on the various components are effective in both directions of pressure application if seepage of planetary atmosphere into any part of the system would be objectionable. Valves for undersea service also operate in environments of greater than 1 atmosphere of pressure.

Very low pressures are encountered in vacuum applications and pose a series of problems. These will be dealt with in a separate section partly because of the confusion in terminology; i.e., pressures below atmospheric are referred to as both vacuum and very low pressures.

VACUUM CONSIDERATIONS

Whether valves are being designed for interplanetary space travel in ultra-low-density gas mixtures consisting primarily of hydrogen and helium or for an industrial process requiring high vacuums, definite effects and phenomena exist which must be recognized and overcome.

Cold welding and outgassing present very serious problems. As lower and lower pressures are encountered, new effects are discovered. Of recognized concern are vacuum problems involving friction, wear, and strength of materials.

Sublimation and Evaporation of Materials

The sublimation rates of metals in a vacuum can be calculated from the Langmuir equation, which assumes that none of the molecules leaving the surface returns to it. The Langmuir equation is

$$G = \frac{P}{17.4}\left(\frac{M}{T}\right)^{1/2}$$

where

G = evaporation rate in grams/cm sec
M = molecular weight
T = absolute temperature in °K
P = vapor pressure at temperature T in millimeters of mercury

The higher the vapor pressure of the material, the higher the rate of sublimation. Cadmium and zinc, which are often used for plating, are poor materials for use in high vacuum; alloys containing lead and zinc as significant constituents should also be avoided. Metals that sublimate or evaporate from a warm surface can condense and collect on a cooler surface; this may cause electrical short circuiting, change surface emissivities, or change optical properties of mirrors and lenses.

Vacuum Effects on Organic Materials

The Space Technology Laboratories investigated the weight loss exhibited by various organic materials in high vacuum. Their experiments showed that the rate of weight loss of materials in high vacuum is, for some initial period, relatively high. It was assumed that the products of this initial loss are, in general, surface contaminants such as moisture, absorbed gases, and low-molecular-weight products of the formulation. Following this initial rapid rate of loss, the weight-loss rate began to follow a characteristic path for the material, suggestive of its general character, formulation, and cure. In some instances, such as for unmodified epoxies, the rate of weight loss became exceedingly small and might even have been undetectable. Materials formulated with completely unreactive additives, with additives which do not completely crosslink, or those materials polymerized by catalysts, exhibited varying rates of weight loss. It is doubtful whether any of the materials tested to date exhibited actual depolymerization or chemical breakdown of the polymers because of the relatively moderate test temperature of 200° F. Exceptions to this statement may possibly be the polysulfide Pro-Seal 727, and the polysulfide-polyamide adhesive Pro-Seal 501. These materials, because of their poor elevated temperature stability, may have suffered some thermal decomposition during testing. However, the chemical composition of each of the vaporized products formed in these experiments was not determined.

The conclusions from this program were:

(1) High-molecular-weight polymers such as Teflon apparently do not evaporate or sublime in vacuum.

(2) The thermal stability of these polymers should be at least as good in vacuum as in air.

(3) The weight loss exhibited by engineering plastics in vacuum is the result of the evaporation of relatively lower molecular weight fractions, unreacted additives, contaminants, etc.

(4) Weight-loss rate and amount of weight loss are greatest early in the test period when the materials at or near the surface evaporate; these loss factors decrease subsequently to a rate determined principally by diffusion rate through the polymer to the surface.

For high-vacuum uses, investigators at various laboratories have concluded that the following organic material recommendations for high-vacuum usage are valid:

(1) Rigid plastics, in general, are preferred over flexible, elastomeric materials.

(2) Materials with a minimum number and quantity of additives and modifiers are preferred.

(3) Complete cure of the plastics must be obtained by extended time and/or elevated temperature postcuring to insure the elimination of unreacted, low-molecular fractions in the products.

(4) Those materials exhibiting high loss rates but considered necessary for use because of special desirable properties should be preconditioned in vacuum at elevated temperature to reduce, as much as possible, the potential loss of the material in actual operation.

Cold Welding

Cold welding appears to be a serious problem area in space applications, and has received considerable attention. It may be defined as the joining of two metal specimens without the presence of a molten-melt phase at the interface.

After prolonged exposure in a vacuum, the surface of a metal loses the layer of adsorbed gas that is normally present on a metal surface on Earth. This loss leads to an increase in the coefficient of friction, and also may lead to partial or complete welding of surfaces in contact.

A clean metallic surface, free of oxides and adsorbed gases, is more subject to cold welding than a contaminated one. For this reason, cold-welding studies are customarily performed under vacuum conditions. Of course, the time an initially clean surface remains clean depends upon the number of gas molecules remaining in the vacuum system. The number of gas molecules is in turn dependent upon the vacuum system's pressure. Variation of clean surface time with pressure is shown in figure 17, in which it is assumed that all incident gas molecules are adsorbed by the metallic surface. If cold welding does require a clean surface for its most extreme effects to be made evident, tests under vacuum conditions are necessary. The information in figure 17 further suggests that effects observed at 10^{-6} torr may be significantly different from those observed in the range of 10^{-11} torr.

The effect of cold welding can be illustrated by an example: A copper tensile test specimen, surrounded by the vacuum environment of space, is pulled apart by thousands of pounds of force. When the broken pieces are pushed together with 100 or 200 pounds of force, the fracture mends itself

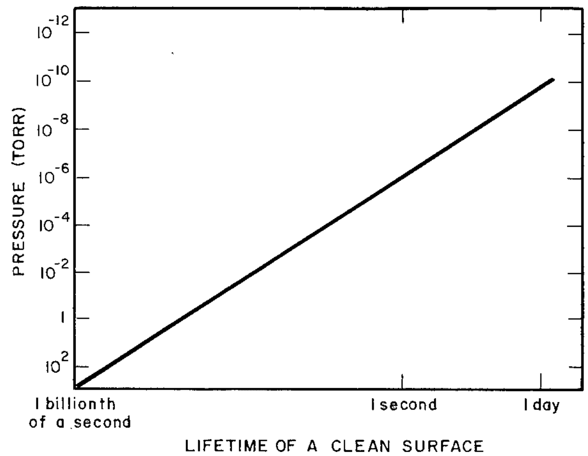

FIGURE 17.—Variation of clean surface time with pressure. (Courtesy of Science and General Electric Co.)

and regains 96 percent of its original strength. This same effect has been observed with materials other than copper. Experiments with 1018 steel in a vacuum show that cohesion, or cold molecular welding, takes place. The studies were made by breaking a specimen in an atmosphere having a pressure of 10^{-6} torr, and then placing the broken faces in contact again with a force equal to the yield strength of the material. Some of the results are given in table V.

TABLE V.—*Cold Welding of 1018 Steel*

Temperature		Maximum cohesion, percent
°C	°F	
500	932	96
150	302	35.9
25	77	18.5

The maximum cohesion shown in the column on the right is the ratio to the initial breaking force of the force required to break the specimen a second time. Several factors such as temperature, time in contact, and force influence the cohesion, but the primary variable is temperature. Apparently, there is some temperature below which cold welding will not take place. Although some metals do not exhibit this phenomenon of cold molecular welding, most of the engineering materials such as steel, aluminum, nickel, and copper will weld together under vacuum conditions.

Cold welding has been a problem in several valve applications. For example, a stainless-steel valve used by D. V. Keller in his work with high vacuum at Syracuse University cold welded shut after baking out at 450° F for 2 hours at 10^{-8} mm Hg. Spaceflight experimental tests are planned to determine the degree of cold welding of valve-seat materials. These will be conducted by TRW Systems for the Air Force Rocket Propulsion Laboratory under contract AF 04(611)–9883. Four valve experiments are planned. Each valve seat and poppet couple will consist of different material combinations of 440° C, tungsten carbide, and 17–4 PH. The valves are normally closed poppet-type solenoid valves and are expected to complete a minimum of 50 000 cycles within a 6-month period.

Subliming camphor vapor will flow through all valves during the first-month cyclic test. The valves will continue to cycle dry for the remaining 5 months. The subliming camphor is intended to be a contamination source, possibly preventing adhesion in the same way that normal propellant leakage might. Seat-bearing stresses under a static poppet load are about 500 psi, with dynamic seating stresses being somewhat higher. Adhesion forces will be determined by monitoring the valve opening time. Cyclic tests conducted by the National Research Corp. (NRC) at 10^{-13} mm Hg under Air Force contract AF 04(611)–9717 on the four valves to be mounted in the spacecraft showed no evidence of cold welding. However, independent tests conducted by NRC showed some adhesion developed on identical material couples tested under dynamic (rotary) test conditions.

On the same research satellite, additional cold-welding tests include supplemental contactors consisting of eight pairs of ⅛-inch-diameter balls, each pair of balls experiencing point contact in a poppet-type cyclic test. The contactors are to cycle a minimum of 400 000 times within a 6-month period. The materials to be tested include 440C, 2014–0 aluminum, OFHC copper, tungsten carbide, and 17–4PH. The maximum contact force between each ball is 10 grams. Adhesion forces are related to contactor opening time.

An important consequence of cold welding is that some maintenance operations, such as drilling, cutting, and unscrewing threaded parts, may be impossible in the space vacuum, particularly since standard lubricants volatilize off the working parts.

OTHER CONSIDERATIONS

The design of valve seats and packing glands becomes critical for pressures in the range of 1500 to 6000 psi. The usual practice of tightening the packing gland when it begins to leak may cause damage to the gland and the valve stem, and may result in even more leakage. For these high pressures, both packing glands and seats are usually made of Teflon, nylon, or polyurethane material. These materials not only minimize the forces required to operate the valve, but also require little or no lubrication.

Many industrial valves can adequately close off pressures up to 4000 psi, but are not bubbletight to a pressure beyond 1000 psi if metal-to-metal seats are used. In metal-to-metal seat design, surface finish becomes the determining factor for leakage control; imperfections in the surface finish form passages for leakage. Higher closing torques will produce frictional wear and cause even further problems. Erosion of hard valve seats is a particular problem with throttling valves. In metal-to-metal seats, it is important to avoid the use of materials which will acquire magnetization during temperature cycling, thereby increasing vulnerability to seat damage by system-generated contamination.

Thermal shocking of the valve seats can cause warpage, cracks, and other problems. As high-pressure gases are released, the sudden expansion normally causes cooling that results in an uneven temperature gradient across the valve.

Soft resilient valve seats have been a common solution to leakage, even in high-pressure applications. At Langley Research Center, an experimental polyurethane valve seat has operated satisfactorily in a liquid nitrogen system. Consideration is being given to replacing Teflon and Kel-F seats with the solid polyurethane material.

Under certain temperature-pressure combinations, helium gas will experience a rise in temperature upon throttling (the reverse Joule-Thompson-law effect). If even the most minute valve-seat surface scratch exists, localized heating will further soften the soft material and cause very rapid erosion. It has been determined that, when this reverse Joule-Thompson-law effect is encountered, the surface finish of the valve seat must be extremely smooth (preferably less than an rms value of 16) to eliminate minor leaks and subsequent throttling heat problems.

O-Ring Assembly Under Zero Lubrication Requirements

A number of aerospace applications require valves with surfaces 20 microinches clean, with no permanent lubricant. Seals and O-rings must be assembled under an essentially dry condition. At Marshall Space Flight Center, Freon was used in an attempt to lubricate the seals and rings temporarily for easy assembly, but the Freon evaporated too fast for proper assembly. Alcohol is now used on K-seals and O-rings to provide a temporary lubricant with a slow evaporation rate, permitting easy assembly of these parts. After assembly, the use of a vacuum oven hastens drying and prevents, to some extent, residual contamination.

Cold Flow of Soft Seats

Cold flow of soft seats has been reported in numerous applications. A number of good design practices have been developed to solve this problem at the various NASA centers. The majority of the solutions consist of surrounding the soft seat material on three sides with a metal backup and then closing the poppet against the fourth surface. Another solution was to machine a groove on the poppet at the mating seat area. Teflon material was then molded in the groove and machined almost flush with the metal-poppet surface. As the poppet closes on the metal seat, the Teflon is compressed into the poppet.

Porosity and Plating

Problems occurred on the poppet of valves used to control hydrogen peroxide in the Project Mercury program. Steel materials were not compatible with the hydrogen peroxide; the steel valve materials were porous. Nickel plating overcame the porosity problem and, polished smooth, was found to be compatible with hydrogen peroxide. In some other aerospace programs, however, plating has been eliminated from valve parts because of the porosity of the plating material.

In solenoid valves, if a material is selected for its magnetic properties, it is seldom compatible with the fluid being controlled. When compromises are made to select a material that is sufficiently magnetic, and yet somewhat incompatible with the fluid, new problems often occur with galling, binding, design clearances, and tolerances. A magnetic material covered with a fluid-compatible plating has seldom proved successful because of the porosity problem.

In applications using nitrogen tetroxide (N_2O_4), NASA strictly avoids plating of materials. Standard specifications for nitrogen tetroxide allow one-half percent of moisture in the gas. Excess moisture will produce nitric acid, which is corrosive to most materials normally used in valves. The attempted use of plating to overcome the compatibility problem introduced insurmountable porosity problems.

Hot Valve Design

The handling of liquid metals at high temperatures has pinpointed problems related to strength of materials and corrosion in valves.

At Lewis Research Center, stainless steel is used up to 1500° F. Haynes 25 is used from 1500° to 1750° F. A 99-percent columbium and 1-percent zirconium alloy looks promising from 1750° to 2000° F.

At this NASA center, liquid-metal systems are in operation using mercury to 800° F, sodium from 200° to 1500° F, and NaK (sodium potassium mixture) in the range of 200° to 1200° F. Valves for these liquid metals depend upon thicker walls to overcome the high-temperature strength problem, but corrosion problems are extremely critical. Hot liquid sodium has a very high affinity for oxygen, and sodium oxide is extremely corrosive.

These corrosion problems require special fabrication techniques. For example, deep penetration welds are required and no cracks or crevices can exist. The welds are very carefully inspected by X-ray, radiography, and dye-penetration techniques.

Packings and bellows present very critical design problems. To gain strength for high-temperature valve usage, it is not always possible to utilize thicker walls. Packings and bellows must remain thin for flexibility.

Another problem with high-temperature valves is that, when high seating forces are encountered, plugs can weld to the seat and stems can weld to guides. Any metal-to-metal-contact surface subjected to both high temperature and high pressure should be critically analyzed in view of the welding problem. In several applications, solutions have been found to the welding problem at high temperatures and pressures by the use of two difference grades of Stellite with different hardnesses.

New materials and processes for improving mechanical properties may soon be applied to valves. Dispersion strengthening and fiber reinforcement of metals, for example, have aroused interest because of their ability to increase both the high-temperature strength and stability of selected alloys. Nodular (ductile) iron, developed in the mid-1940's, is now used in valves; its adoption illustrates the utility to be expected from new material developments.

CHAPTER 4

Compatibility With Liquid Propellants

In missile and space programs, cryogenic and noncryogenic propellants have caused serious compatibility problems. Similar problems arise in many branches of industry where propellants and cryogenic fluids are manufactured, stored, and transported. Compatibility problems also crop up in the handling of many chemicals and process fluids.

This section reports valve materials compatibility with liquid propellants. The information is largely from "Advanced Valve Technology for Spacecraft Engines," by Salvinski et al. (ref. 1), which reviews the literature. Further information on materials for various fluid component elements that show a good probability of being resistant to chemical attack under usual conditions is given in the "Aerospace Fluid Component Designers' Handbook" (ref. 2). Many of the propellants listed are hypergolic in nature. Some, such as liquid fluorine, hydrogen, and oxygen, are cryogenic fluids.

From the standpoint of the valve designer, the available data do not provide the information necessary for rapid and automatic selection of material. For example, the data presently available do not normally distinguish between static and dynamic compatibility either with respect to the test methods employed or, where a material's use is approved, with respect to that material's potential application. The mechanism for potential failure of a particular material may be augmented by the mechanical action of adjoining component materials or the erosive effects of the flowing propellant itself. Specifically, a material appropriate for use as a static packing within a given valve assembly may be completely incompatible when used as a sliding seal or an impact seal in the identical propellant service. Data regarding these conditional aspects of compatibility are not documented at present. Similarly, the effects of temperature, pressure, and propellant phase variations have seldom been treated in establishing compatibility.

A second distinct information void, with respect to material compatibility, exists in accumulation of historical data. The primary source of the data presented is the laboratory test. This is seldom supplemented by field information, which should be extensive.

Table VI shows the compatibility ratings for valve components with each propellant. Values were assigned according to the following definitions:

Rating	*Definition*
1	A value of 1 was assigned to those combinations with which a serious problem exists; i.e., one for which there is no satisfactory solution.
2	A value of 2 was assigned to those combinations with which a problem exists, but for which a remedy may be available. That is, the combination of parameter is deemed to be acceptable, with qualifications.
3	A value of 3 was assigned to those combinations which were deemed to be satisfactory, i.e., within the present state of the art.
U	A designation of U was made where the necessary information upon which to base a judgment was unavailable.
NA	Where a parameter was not applicable, the letters NA were used.

TABLE VI.—*Ratings of Compatibility Between Value Components and Liquid Propellant Ingredients*

Propellant ingredient	Compatibility							Propellant characteristics							Valve types					Probable rating in 10 years
	Metals	Ceramics	Organic polymers	Wet lubes	Dry lubes	Soft seats	Hard seats	Shock sensitivity	Lubricity	Viscosity	Radiation tolerance	Effects of leakage	Control of flow	Sterilization	Regulators	Shutoff	Flow metering	Vent (zero-g)	Disconnect	
Fuels:																				
Hydrazine	3	3	3	1–2	2	3	3	3	1	2	2	1–2	3	1	3	3	3	NA	3	3
Monomethylhydrazine	3	3	2	1	1	U	U	3	U	3	2	1	3	1	3	3	3	NA	3	3
U-Dimethylhydrazine	3	3	2	1–2	1–2	[a]3	3	3	1	3	2	1–2	3	1	3	3	3	NA	3	3
Aerozine-50	3	3	2	1–2	1–2	[a]3	3	3	1	2	2	1–2	3	1	3	3	3	NA	3	3
Pentaborane	3	2	2–3	2	3	3	2	3	U	3	3	1	3	1	3	3	3	NA	3	3
Oxidizers:																				
Nitrogen tetroxide	3	3	1–2	1	2	1–2	3	3	3	3	3	1	3	1	3	3	3	NA	3	3
Oxygen difluoride	3	2	1–2	1	1	1	3	3	U	3	U	1–2	3	U	3	[b]2	3	2	3	2
Chlorine trifluoride	2–3	2	1–2	1	1	1	3	3	U	3	3	1	3	U	3	[b]2	3	2	3	2–3
Perchloryl fluoride	3	2	2	1–2	1	2	3	3	U	2–3	U	1–2	3	U	3	[b]2	3	3	3	2–3
Cryogenics:																				
Liquid hydrogen	3	3	1–1	1	1	[a]2–3	3	3	2	3	3	2	3	2	3	[b]1–2	3	1	2	2–3
Liquid oxygen	3	3	2	1	1	[a]2–3	3	3	2	3	3	2	3	2	3	[b]2	3	1	2	3
	3	2	1–2	1	1	1–2	2–3	3	U	3	3	1	3	U	3	[b]1	1–2	1	2	2
Gels:																				
Liquid gels	2	2	2	2	2	[a]3	3	3	2	2	2	2	2	1	U	U	U	NA	U	2
Metallized gels	2	2	2	2	2	2	2	3	1	2	2	1	1	1	U	U	U	NA	U	2

[a] These ratings, unlike those under "Organic polymers," were based on the use of a specific polymeric material, in most cases Teflon, for soft seats.

[b] Ratings based on leakage control.

Note: 1=serious problem of compatibility; 2=problem which may have remedy; 3=satisfactory combinations; U=unavailable; NA=not applicable.

The more serious problem areas (ratings 1 or 2), as related to each propellant type, are discussed in the text that follows.

HYDRAZINE

Physical Properties

Specific gravity	1.00 (60° F)
Molecular weight	32.048
Freezing temperature, °F	35.1
Normal boiling point, °F	236.3
Critical temperature, °F	716
Critical pressure, psia	2131
Heat of vaporization, Btu/lb_m	540

Hydrazine (N_2H_4) is a clear liquid used as a high-energy propellant because it is insensitive to mechanical shock or friction and exhibits excellent thermal stability at ambient temperatures. It is hazardous, because of its toxicity, reactivity, and flammability. Since it is thermodynamically unstable (i.e., a monopropellant), hydrazine will decompose under elevated temperatures when catalyzed by graphite or a metal oxide such as iron oxide or copper oxide, and will release considerable energy resulting in a possible explosion or fire. In addition, liquid hydrazine exerts sufficient vapor pressure above 100° F to form flammable mixtures with air. Its freezing point is +34° F. Because hydrazine is hygroscopic and readily forms flammable mixtures in air, a nitrogen blanket is required.

In assessing the compatibility of a material with hydrazine, the specific application must be considered. Materials satisfactorily used with hydrazine where air oxidation of the surface can be prevented may not be satisfactory for service where prolonged exposure to air cannot be avoided. Factors to consider when selecting materials to use with hydrazine for any given exposure condition are: (1) corrosiveness of the material in contact with hydrazine, and (2) the effect of the material and/or its corrosion products formed on the rate of decomposition of hydrazine. These factors are particularly important for carbon steel, low-alloy steels, copper alloys, and molybdenum. From the corrosion standpoint these metals are satisfactory for use in hydrazine, but these metals and/or their oxides may catalyze hydrazine decomposition at elevated temperatures and explosions may result.

Table VII lists those materials considered to be compatible with hydrazine for long-term application.

TABLE VII.—*Compatibility of Materials Tested With Hydrazine* (N_2H_4) *for Long-Term Application*

Material	Test temperature, °F
Aluminum alloys:	
1100	140
2014	80
2017	160
2024	70
3003	80
4043	160
5052	80
5456	140
6061	160
6066	80
716	140
356	160
40E	75
Tens 50	----------
Stainless steel:	
410	80
416	[1]200
430	68
440C	80
302	80
304	140
316	200
317	80
321	140
347	200
17–4PH	140
17–7PH	75
AM 350	160
AM 355	160
Miscellaneous metals:	
Brass	80
Chromel A	80
Chromium plating	70
Copper	68
Gold	75
Hastelloy C	125
Inconel	200
Inconel-X	80
K-Monel	140
Lead	68
Molybdenum	75
Monel	80
Nichrome	80
Silver	80

TABLE VII.—Continued

Material	Test temperature °F
Miscellaneous metals—Con.	
Silver solder	75
Stellite 21	75
Tantalum	80
Tin	80
Titanium, A110AT	140
Titanium, 6A1–4V	160
Zirconium	75
90 Pb–10 Sn	75
Nonmetals:	
Butyl rubber compound 805–70	140
Butyl rubber compound 823–70 (Parco)	----------
Butyl rubber compound B 480 (Parker)	----------
Butyl rubber compound 9257 (Precision)	----------
Ethylene propylene (Stillman 822–70 and Parker O rings)	----------
Kel-F	80
Polyethylene	80
Teflon	140
Teflon coatings	----------
Asbestos	80
Delanium	140
Glass	80
Polypropylene	----------
SBR	75
Silicone grease DC–11	----------

[1] Pits.

NOTE.—Metals listed above are rated compatible based on a corrosion rate of less than 1 mil per year and in the case where the material does not cause decomposition and is free from impact sensitivity. Nonmetals are rated for satisfactory service for general use.

Materials Used in Valve Parts

Valve bodies. Stainless steels 304, 304L, 316, 321, 347; aluminum alloys 6061, 3003, 4043, 2024, 356T6, Tens 50; titanium 6A1–4V, B120VCA.

Springs. Stainless steels 301, 321, 347, AM 350, AM 355, 17–4PH, 17–7PH; alloy steel A–286; Inconel, Inconel-X.

Stems. Stainless steels 321, 347, 403, 410, AM 350, AM 355, 17–4PH, 17–7PH; alloy steel 8630.

Bellows. Stainless steels 303, 321, 347; Inconel, Inconel-X.

Bearings. Stainless steels 301, 301N, 403, 410, 440C.

Valving units (seats and poppets). Stainless steels 303, 321, 347, 440C, AM 350, 17–4PH; Teflon; aluminum 1100; Stellite 21; nylon; Kynar.

Seals. Teflon; aluminum 1100; butyl rubber compounds 805–70 (Parco), 613–75 (Stillman), 823–70 (Parco), B–480–7 (Parker), 60–61 (Bell), 9257 (Precision); propylene; polyethylene; Hypalon; Cis-4 polybutadiene; Buna N; Neoprene; natural rubber; Kel-F (subject to stress cracking), SBR.

TABLE VII.—Concluded

Packing. Teflon, Kel-F.

Lubricants. Teflon coatings and carbon graphite; DC–11.

Bolts, nuts, and screws. Stainless steels 303, 321, 347, 17–4PH, 17–7PH; Inconel-X.

Thread sealants and antiseize compounds. Unsintered Teflon; Redel UDMH Sealant, LOX Safe (exterior use only).

Coatings. Chrome plate, anodize (aluminum and magnesium), nickel plate.

Diaphragms. Stainless steels 304, 321, 347; Teflon; butyl rubber; SBR.

Hydrazine (N_2H_4) Problem Areas

Wet lubes [*1–2*].[1] No completely satisfactory lubricant has been developed. For specific and/or limited use, some silicone lubricants and "Q-Seal" (Quigley Co.) are being used with fair results.

Lubricity [*1*]. Unsatisfactory lubricating performance was found for hydrazine in a series of low-load short-duration ball bearing and gear tests at 24 600 rpm. The poor lubricity, resulting in degradation of the component metals surveyed, was attributed to its active solvent and reducing properties.

Viscosity [*2*]. The viscosity of hydrazine presents a problem only at low temperatures. The freezing point of commercial hydrazine is approximately 30° F, which is relatively high for operation under cold climatic conditions. Three methods of preventing N_2H_4 from freezing have been employed: (1) the addition of freezing-point depressant, (2) insulation of container and components, and (3) insulation plus tracing with heat elements, steam, or hot water.

Radiation tolerance [*2*]. Hydrazine, UDMH, and Aerozine-50, subjected to an irradiation dose of 1×10^9 erg/g, which is the maximum space radiation dose likely to be incurred in 2 years in the Van Allen belts, showed that the composition of the

[1] Bracketed numbers refer to compatibility ratings of valve components with propellants which are defined in table V.

propellants was not significantly affected. However, a pressure increase resulted, accompanied by the evolution of insoluble gases, due to slight decomposition. It is therefore recommended that tanks be designed to minimize this problem.

Effects of leakage [1–2]. Because anhydrous hydrazine is a flammable liquid, leakage may initiate a fire. It is hypergolic with most oxidizing agents and decomposes explosively on contact with catalytic materials including iron rust. Vapors of hydrazine can be detonated by an electric spark or an open flame.

Since this propellant is a strong chemical reducing agent, leakage may result in malfunction of other elements in the system and physical injury. Prolonged exposure to this toxic material produces damage to the liver and kidneys and, to a lesser extent, anemia and lowering of blood-sugar concentrations. The threshold value adopted by the American Conference of Governmental Industrial Hygienists is 1 ppm.

Leakage rates in space compared to Earth will be greater, since they are dependent upon the pressure differential between the system and the external pressure. The leakage potential will be increased by approximately 14.7 lb/in.^{2} in space vacuum over that found at the Earth's surface. Leakage of the propellant to space atmosphere will result in much more rapid volatilization and dispersion of the material than occurs on Earth, resulting in a lessening of many hazards associated with propellant contact. However, leakage in a confined system in space will have the same detrimental effects as those previously discussed as Earth hazards; i.e., toxicity, component malfunction, and fire and explosion hazards.

MONOMETHYLHYDRAZINE

Physical Properties

Molecular weight	46.075
Freezing temperature, °F	−63
Normal boiling point, °F	189
Critical temperature, °F	609
Critical pressure, psia	1195
Heat of vaporization, Btu/lb$_m$	377

Characteristics

Monomethylhydrazine (CH_3NHNH_2) (MMH) is a clear, water-white liquid with a strong amine odor detectable in concentrations of 1 to 3 ppm. It is a fairly volatile liquid; the vapor-pressure, 0.96 psia at 77° F, is higher than that of hydrazine. Hence it is more hazardous to health than hydrazine. The maximum allowable concentration of MMH vapor in air has not been established, but it has been suggested that it be set within 0.1 and 0.5 ppm.

Liquid MMH is not sensitive to impact or friction and is more stable than hydrazine under conditions of mild heating. The flammability characteristics of MMH with air are close to those of hydrazine and UDMH; consequently, it should be maintained under a nitrogen blanket at all times. MMH must be stored away from any oxidizers and from possible sources of ignition. All metallic equipment employed for storage and handling of MMH requires grounding to prevent an accumulation of static charge.

Monomethylhydrazine, like hydrazine, decomposes explosively on contact with a decomposition catalyst. Iron oxide, for example, is known to cause spontaneous ignition of these propellants. Copper, lead, and zinc are not compatible with MMH.

An extensive compilation of compatibility data for metals and nonmetals is not available. Due to the similarity in catalytic oxidation and chemical activity for MMH and hydrazine, those metals found satisfactory for hydrazine should suffice for MMH. Only short-term studies on a selected few plastics have been reported; no long-term compatibility tests have been done.

Table VIII lists those materials recommended for storage and handling.

Monomethylhydrazine Problem Areas

Organic polymers [2]. In general, MMH attacks organic materials more readily than does hydrazine. Materials satisfactory for limited use include Teflon, some silicone rubbers, and high density polyethylene.

TABLE VIII.—*Compatibility of Materials Tested With Monomethylhydrazine for Short-Term Use* [a]

Material	Test temperatures, °F
Aluminum alloys:	
3003	Satisfactory below 160.
5052	
5154	
6061–S	
Steels:	
303 Stainless steels	Satisfactory below 160.
304 Stainless steels	
321 Stainless steels	
347 Stainless steels	
17–7PH Stainless steels	
4130[b]	
Plastics and elastomers:	
Natural rubber	Intermediate[c] below 160, 1 week.
Neoprene	Intermediate[c] below 160, 1 week.
Polyethylene (high density)	Good[d] below 160, 1 week.
Silicone rubber	Intermediate[a] below 160, 1 week.
Teflon	Intermediate[a] below 160, 1 week.

[a]2 to 4 weeks.
[b]Very susceptible to rusting.
[c]Moderate weight change (0.5–2.5 percent); change in elasticity of 25–40 percent.
[d]Negligible weight change (0.5 percent); no change in elasticity.

Materials Used in Valve Parts

Valve bodies. Stainless steels 304, 304L, 321, 347, 17–7PH; aluminum alloys 3003, 5052, 6061, Tens 50, 356T6.

Springs. Stainless steels 301, 321, 347, 17–7PH.

Stems. Stainless steels 321, 347, 403, 410, AM 350, AM 355, 17–4PH, 17–7PH.

Bellows. Stainless steels 303, 321, 347; Inconel, Inconel–X.

Bearings. Stainless steels 301, 301N, 403, 410, 440.

Valving units (seats and poppets). Stainless steels 303, 321, 347; Teflon; polypropylene; nylon.

Seals. Teflon, polypropylene, nylon, polyethylene, Neoprene, silicone, natural rubber.

Packing. Teflon, Kel-F.

Lubricants. Teflon coatings, Dow-Corning 11

TABLE VIII.—Concluded

Compound (silicone), Fluorolube GR–470, Kel-F grease.

Bolts, nuts, and screws. Stainless steels 303, 321, 347, 17–4PH, 17–7PH; Inconel-X.

Thread sealants and antiseize compounds. Unsintered Teflon tape.

Coatings. Chrome plate.

Diaphragms. Stainless steels 304, 321, 347; Teflon.

Braze alloys. Permabraze 130 (82 percent), Au, (18 percent Ni).

Wet and dry lubes [*1*]. Because MMH is a better solvent than UDMH and N_2H_4 it also has attendant undesirable lubricant and "washout" effects.

Radiation tolerance [*2*]. This rating was made by analogy to hydrazine and UDMH, which have very similar chemical structure and physical and chemical properties.

Effects of leakage [*1*]. MMH is somewhat more toxic than UDMH or hydrazine; its flammability characteristics are similar. MMH vapors detonate on ignition by static sparking. Iron rust promotes the catalytic decomposition, resulting in spontaneous ignition. For effects of space vacuum, see the discussed effects of leakage for hydrazine.

UNSYMMETRICAL DIMETHYLHYDRAZINE

Physical Properties

Specific gravity	0.785 (60° F)
Molecular weight	60.078
Freezing temperature, °F	−72
Normal boiling point, °F	146
Critical temperature, °F	480
Critical pressure, psia	880
Heat of vaporization, Btu/lb_m	250.7

Unsymmetrical dimethylhydrazine ($(CH_3)_2NNH_2$) (UDMH) is a derivative of hydrazine and is a moderately toxic, shock-insensitive, storable liquid propellant. It exhibits excellent thermal stability and resistance to catalytic breakdown. It has a lower freezing point and higher boiling point than hydrazine.

Because of its extremely wide flammable range in air and the possibility that explosive vapor-air mixtures may be found above the liquid, UDMH should not be exposed to open air. It should be stored in a closed container under a nitrogen blanket.

Test results imply that lubricants which are compatible for use with UDMH are still in the development stage. Lubricants such as APS C–407, Parkerlube 6PB, Molykote, and Peraline 12-4 may cause decomposition, while petroleum and silicone greases do not react but are dissolved by the UDMH.

Table IX lists those materials which are considered to be compatible with UDMH for long-term application.

TABLE IX.—*Compatibility of Materials Tested With Unsymmetrical Dimethylhydrazine (UDMH) for Long-Term Application*

Material	Test temperature, °F
Aluminum alloys:	
1100	160
1100–H14	145
1260–H14	145
2014	140
2017	75
2024	75
2024–T3	145
3003	86
3003–H14	145
3004–H34	145
5052	160
5052–H34	145
5086	85
5086–H34	145
5154–H34	145
5456	140
6061	160
6061–T6	145
6063–T6	145
7075	160
7075–T6	145
356	160
356–T6	85
43	145
Stainless steels:	
302	160
303	160
304	160
316	140
321	160
347	160
410	160
416	250
422	145
17–7PH	160
PHI5–7 Mo	85

TABLE IX.—Continued

Material	Test temperature, °F
Stainless steels—Continued	
Carpenter 20	140
A286	85
AM350	----------
AM355 CRT	100
17–4PH, condition H925	----------
17–4PH, condition H1151	----------
Miscellaneous metals:	
Brass	75
Copper	145
Hastelloy (B, C, X, F)	145
Haynes alloy 25	140
Inconel	140
Magnesium alloy AZ–31B–O	85
Magnesium alloy AZ–31B, AZ–61A	130
Magnesium alloy AZ–92–F	85
Mild steel	140
Monel	140
Nickel	140
Tantalum	140
Titanium A–55 (commercially pure)	145
Titanium alloy B–120VCA	145
Titanium alloy C–120AV	160
4130 Steel	85
Plastics and elastomers:	
Butyl rubber	140
Kel-F (unplasticized)	140
Nylon	130
Polyethylene	80
Teflon (FEP)	160
Teflon (TFE)	160
Miscellaneous materials:	
Delanium	75
Glass Pyrex	160
Graphitar No. 2	140
Graphite	75

NOTE: Materials listed above are rated compatible based on a corrosion rate of less than 1 mil per year and in the case where the material does not cause decomposition, and is free from impact sensitivity. Nonmetals are rated for satisfactory service for general use.

Materials Used in Valve Parts

Valve bodies. Stainless steels 303, 304, 316, 321, 347; aluminum alloys 6061, 3003, 356T6, 2024; brass; titanium A–556, 6A1–4V, B–120VCA; magnesium AZ–92–F, AZ–31B–0.

Springs. Stainless steels 301, 321, 347, 17–4PH, 17–7PH; alloy steel A–286; Inconel; Monel.

TABLE IX.—Concluded

Stems. Stainless steels 321, 347, 403, 410, AM 350, AM 355, 17-4PH, 17-7PH; alloy steel 8630.

Bellows. Stainless steels 303, 321, 347; Inconel-X.

Bearings. Stainless steels 301, 301N, 403, 410, 440C; alloy steel 4130.

Valving units (seats and poppets). Stainless steels 303, 347, 410, AM 350, 17-4PH; Teflon; aluminum 1100; polypropylene; copper; polyethylene; nylon; Kel-F.

Seals. Teflon; aluminum 1100; butyl rubber compounds 823-70 (Parco), B480-7 (Parker), ethylene propylene rubber E515-8 (Parker), 61375 (Stillman), 9257 (Precision); polypropylene; polyethylene Cis-4-polybutadiene; Kel-F.

Packing. Teflon, Kel-F.

Lubricants. Teflon coatings, carbon graphite, Apiezon L, Reddy Lube 200.

Bolts, nuts, and screws. Stainless steels 303, 321, 347, AM 350, AM 355, 17-4PH, 17-7PH; Inconel-X.

Thread sealants and antiseize compounds. Unsintered Teflon; Redel UDMH Sealant, LOX Safe (exterior use only).

Coatings. Chrome plate.

Diaphragms. Stainless steels 304, 321, 347; Teflon TFE and FEP. Mylar unsuitable.

Unsymmetrical Dimethylhydrazine (UDMH) Problem Areas

Organic polymers [*2*]. In view of the varied services in which organic polymers might be employed, it is difficult to be specific regarding their performance. Because of the excellent solvent properties of UDMH, most polymeric materials are unsatisfactory. Numerous plastics and rubbers are rated "good" (suitable for repeated short-term exposure). The references cited indicate that Teflon was not attacked by UDMH and rated it as suitable for long-term storage or exposure.

Wet and dry lubes [*1–2*]. Because of the solvent properties of UDMH, no completely satisfactory lubricant has been found. Several lubricants, litharge, glycerine paste, X-Pando, and Q-Seal, are compatible and have been employed for pipe threads and similar applications involving minimum contact with UDMH. Petroleum and silicone greases do not react; however, all tend to "washout" under dynamic conditions.

Lubricity [*1*]. UDMH was found to have unsatisfactory lubricating properties when subjected to bearing and gear tests. The performance was thought to be related to the solvent and reducing properties of UDMH.

Radiation tolerance [*2*]. See hydrazine.

Effects of leakage [*2*]. Being chemically similar to hydrazine, the same hazards are encountered. The maximum allowable concentration is somewhat less (0.5 ppm) than hydrazine. UDMH is flammable in air and hypergolic in the presence of oxidizers. Its excellent solvent properties may cause malfunction of components constructed of incompatible organic plastics. For space conditions, see the discussion on effects of leakage for hydrazine.

AEROZINE-50

Physical Properties

Molecular weight	41.797
Freezing temperature, °F	18
Normal boiling point, °F	170
Critical temperature, °F	634
Critical pressure, psia	1696
Heat of vaporization, Btu/lb$_m$	425.8

The miscible blend of a nominal 50 percent by weight of hydrazine (N_2H_4) and 50 percent by weight of unsymmetrical dimethylhydrazine (UDMH) is a hygroscopic liquid (readily capable of absorbing moisture) which is insensitive to mechanical shock but flammable in both liquid and vapor states. Since the vapor phase is predominantly UDMH at 72°, the flammability hazards of the mixture are the same as for UDMH. Explosion hazards can be minimized, however, by maintaining the fuel in closed systems.

Most common metals which might be used for valve construction, with the exception of the magnesium and copper alloys, are compatible with the 50/50 fuel blend if they are clean. Care should be exercised when using ferrous alloys because of the possible catalytic decomposition of the fuel blend due to rust. Under certain conditions (e.g., high temperature), rust may catalyze decomposition of N_2H_4.

Table X lists those materials which are considered to be compatible with Aerozine-50 for long-term application.

TABLE X.—*Compatibility of Materials Tested With Aerozine-50 for Long-term Application*

Material	Test temperature, °F
Aluminum alloys:	
1100	55–60
2014–T4	55–60
[a]2014–T6	160
[a]2024–T6	160
2219–T81	55–60
3003–H14	150
5086–H36	160
5254–F	160
5456–H24	55–60
5456–H321	160
6061–T6	160
6066	160
[a]7075–T6	160
356	160
Tens 50	160
Stainless steel:	
303	160
304L	
[a]316	160
321	160
347	160
PH15–7 Mo (condition A)	160
[a]17–4 PH	160
17–4 PH (condition A)	160
[a]AM355 (condition H)	160
[a]AM350 SCT	160
[a]410H and T	160
440C	160
Other metals:	
[a]Anodize coatings on aluminum	160
Berlyco 25	160
Cadmium plate	60
Nonporous chromium plating	55–60
Gold plate	160
Monel	80
Nickel	160
Nonporous electrolytic nickel plating	55–60
[a]Electroless nickel plating	160
Silver	55–60
Silver solder	60
Stellite 6K	160
[a]Stellite 21	160
Stellite 25	160
A286 Steel (rust free)	55–60
[a]1020 Steel (rust free)	55–60
4130 Steel (rust free)	55–60
Ti alloy A110–AT	160
Ti alloy C/20 AV	160
Ti alloy B/20 VCA	55–60

TABLE X.—Continued

Material	Test temperature, °F
Other metals—Continued	
Tin	55–60
Titanium carbide (Ni binder)	160
Tungsten carbide	160
Plastics and elastomers:	
Enjay 035	80
Fluorobestos filled with asbestos	55–60
Fluorogreen	55–60
Hadbar SB800–71 rubber	160
Kel-F 300 (15 percent glass filled)	75
Kel-F 300 (unfilled)	75
Parker B496–7 rubber	160
Low-density polyethylene	60
Polypropylene	160
Precision rubber 9257, 940, X559	80
Teflon (FEP)	70–80
Teflon (TFE)	70–80
Teflon filled with graphite	55–60
Teflon filled with molydisulfide	55–60
Teflon filled with asbestos	55–60
Zytel 31 nylon	70–80
Zytel 101 nylon	60
Lubricants and graphite:	
Flake graphite	80
Graphitar 39	70–80
Graphitar 84	70–80
Graphitar 86	160
Microseal 100–1 (dry lube)	80
National carbon CCP–72	160
Purebon P3N	160
Reddy lube 100	80
Reddy lube 200	80
Silicone DC II	80
Water glass graphite	80
Ceramics:	
Rockflux	75
Sauereisen P–1	60
Sauereisen 31	60
Temporall 1500	60
Adhesives:	
Epon 422	80

[a]Disagreement exists among authorities as to acceptability.

Materials Used in Valve Parts

Valve bodies. Stainless steels 303, 304L, 316, 321, 347; aluminum alloys 2219, 6061, 3003, 5456, 7075, 2024; titanium alloys B120–VCA, A110–AT.

Springs. Stainless steels 301, 321, 347, 17–4PH, 17–7PH; alloy steel A–286; Ni Span C; Inconel–X.

TABLE X.—Concluded

Stems. Stainless steels 321, 347, 410, 403, AM 355, 17–4PH, 17–7PH; alloy steel 8630; Haynes Stellite 25.

Bellows. Stainless steels 303, 321, 347; Inconel-X; Berylco 25.

Bearings. Stainless steels 301, 301N, 403, 410, 440C.

Valving units (seats and poppets). Stainless steels 303, 347; aluminum 1100; Teflon; Zytel 101; 31 nylon, butyl rubber compounds 823–70 (Parco), B480–7 (Parker), 9257 (Precision); polypropylene; Haynes Stellite 25, 6K, 21; titanium carbide, tungsten carbide.

Seals. Aluminum 1100; Teflon; butyl rubber compounds 823–70 (Parco), 805–70 (Parco), 1357 (Goshen), B480–7 (Parker), B496–7 (Parker), 9257 (Precision); polyethylene; Kel–F; ethylene propylene rubber.

Packing. Teflon, Kel-F.

Lubricants. Teflon coatings and carbon graphite; UDMH Lube; LOX Safe, Microseal 100–1. Fluorinated lubricants unsatisfactory.

Bolts, nuts, and screws. Stainless steels 303, 321, 347, AM 355, AM 350, 17–4PH, 17–7PH.

Thread sealants and antiseize compounds. Unsintered Teflon; Redel UDMH Sealant, LOX Safe; Reddy Lube 100, 200; Drilube 822.

Coatings. Chrome plate, nickel, anodize.

Diaphragms. Teflon, butyl rubber, Berylco 25. Mylar satisfactory for vapor exposure but unsuitable for liquid.

Aerozine-50 Problem Areas

Organic polymers [2]. Most organic polymers either dissolve or deteriorate in the presence of Aerozine-50. Teflon has proved most successful for some uses; butyl rubbers are being used for most dynamic seals.

Wet lubes [1–2]. Certain silicone-base greases, polyglycol oils, and powdered Teflon have been used; however, no lubricant has been found immune to "washout." The halogenated lubricants react; many dissolve.

Dry lubes [1–2]. Graphite has been used, but found susceptible to washing out under dynamic conditions.

Lubricity [1]. Being a mixture of hydrazine and UDMH, Aerozine-50 has properties similar to the two components.

Viscosity [2]. Because of the high freezing temperature (18° F) of Aerozine-50 and the rapid increase in viscosity as this temperature is approached, fuel temperature must be maintained well above this point to insure proper flow characteristics.

Radiation tolerance [2]. See hydrazine.

Effects of leakage [1–2]. Leakage effects are similar to those of the components discussed below.

PENTABORANE

Physical Properties

Molecular weight	63.17
Freezing temperature, °F	−52
Normal boiling point, °F	136
Critical temperature, °F	435
Critical pressure, psia	557
Heat of vaporization, Btu/lb_m	219

Characteristics

Pentaborane (B_5H_9) is an extremely hazardous, high-energy rocket fuel, insensitive to mechanical shock and in an inert atmosphere exhibits satisfactory thermal stability. It is considered a hazardous propellant due to its toxicity, high reactivity, and erratic hypergolic characteristics. Pure pentaborane will usually ignite spontaneously upon contact with air at atmospheric pressure. It is also hypergolic with high-energy oxidizers such as chlorine trifluoride at atmospheric pressure. In oxidation-reduction reactions, pentaborane behaves as a very strong reducing agent.

Any substance which will function as a potential oxidizer will react with pentaborane. Materials such as water, air, metal oxides, and reducible organic compounds are in this category. For this reason, considerable care should be exercised in the selection of materials to be used with pentaborane. Use of any organic compounds containing a reducible functional group should be avoided. Teflon, Viton, Kel-F, and Fluorosilica rubber are among the polymers compatible with pentaborane.

Gaskets, lubricants, and seals must be chemically inert to pentaborane. High-porosity castings and gaskets should be avoided. To date, no metals are known to be incompatible with pentaborane at ordinary room temperatures and atmospheric pressure.

Table XI lists those materials which are considered to be compatible with pentaborane for long-term applications.

TABLE XI.—*Compatibility of Materials Tested With Pentaborane for Long-Term Application*

Material	Test temperature, °F
Aluminum Alloys:	
2024–T3	75
3003–HI4	75
5052–S	75
6061–T6	75
7075–T6	75
356–T6	75
Cadmium-coated aluminum	75
Chromated aluminum	75
Stainless steels:	
302	75
304	75
321	75
347	75
Other metals:	
Brass	75
Cadmium-plated steel	75
Copper	75
Hastelloy No. X–1258	75
Iron	75
K-Monel	75
Magnesium alloy, AZ63A	
Magnesium alloy, AZ318	
Monel, soft, M–8330–B	
Nichrome "V"	75
Steel	75
Titanium alloy C–110M	75
Titanium alloy C–130AM	75
Nonmetals:	
Fluoroflex T	75
Fluorosilicone rubber	75
Graphitar No. 39	75
Graphite-impregnated asbestos	75
Kel-F No. 5500	75
Kel-F and glass cloth	75
Kel-F and glass yarn	75
Molybdenum disulfide	75
Pure carbons	75
Rockwell Nordstrom Lube No. 921	75
Teflon	75
Viton	75
Viton A	75

NOTE: Materials listed above are rated compatible based on corrosion rate of less than 1 mil per year and in the case where the material does not cause

TABLE XI.—Concluded

decomposition, and is free from impact sensitivity. Nonmetals are rated for satisfactory service for general use.

Materials Used in Valve Parts

Valve bodies. Stainless steels 304, 316, 321, 347; alloy steel 4130; titanium alloys; magnesium alloys; aluminum alloys 2024, 6061, 356T6, 7075T6.

Springs. Stainless steels 302, 304, 321, 347, 17–7 PH; K-Monel.

Stems. Alloy steel 17–7PH; K-Monel.

Bellows. Stainless steels 304, 321, 347; Monel, K-Monel; Inconel.

Bearings. Alloy steel 4130.

Valving units (seats and poppets). Teflon, Kel-F, copper.

Seals. Teflon, Kel-F, Viton A, Viton B, polyethylene, polypropylene, Kel-F elastomer, fluorosilicone.

Packing. Teflon, Kel-F.

Lubricants. Rockwell Lubricant 921, Graphitax No. 39, Gold Harmony 0.1 Nos. 44 and 69.

Bolts, nuts, and screws. Stainless steels 304, 321, 347, 17–7PH; alloy steel 4130; Monel.

Thread sealants and antiseize compounds. Teflon tape.

Coatings. Cadmium; anodize (aluminum and magnesium).

Diaphragms. Stainless steels 304, 321, 347; Teflon. Mylar unsuitable.

Pentaborane (B_5H_9) Problem Areas

Ceramics [*2*]. Because of its strong reducing properties, pentaborane will reduce some metal oxides and also precipitate some heavy metals from solutions of their salts.

Wet lubes [*2*]. A number of lubricants are satisfactory for short-term service with pentaborane; however, being soluble in these lubricants, pentaborane presents problems of cleaning, disposal, and "washout" of lubricant from components. Generally, nonlubricated valve designs acceptable for use with other toxic and corrosive liquids may be used successfully with pentaborane.

Effects of leakage [*1*]. The toxicity of pentaborane, with a tentative threshold limit of 0.005 ppm, would constitute a severe hazard to humans in the event of leakage. The pyrophoricity of pentaborane in air has been a controversial subject; however, it is generally agreed that pentaborane must be treated as if it were spontaneously flammable in air under most conditions. For leakage in space, see the discussed effects of leakage for hydrazine.

Hard seats [2]. Because of the necessity of eliminating all leakage, soft-seat materials compatible with pentaborane (i.e., Teflon or Kel-F) are recommended. Particle migration occurring where valve parts rub on plastic seals may cause problems of seal life: plugging or opening and fouling up close tolerants fits. Hard seats of compatible metals would suffice, provided leakage could be eliminated.

NITROGEN TETROXIDE

Physical Properties

Specific gravity	1.49 (60° F)
Molecular weight	92.016
Freezing temperature, °F	11.8
Normal boiling point, °F	70
Critical temperature, °F	316
Critical pressure, psia	1470
Heat of vaporization, Btu/lb$_m$	178.2

Nitrogen tetroxide (N_2O_4) is a highly reactive, toxic oxidizer, insensitive to all types of mechanical shock and impact. It is a dense brown liquid and, although nonflammable itself, it will support combustion, and upon contact with certain high-energy fuels, such as the hydrazines, it will react hypergolically. Nitrogen tetroxide can cause spontaneous ignition with common materials such as leather and wood. The fumes are extremely toxic. Nitrogen tetroxide is a storable propellant oxidizer and is used in the Titan II missile and many spacecraft propulsion systems.

Dry nitrogen tetroxide is compatible with many metals and alloys used in space vehicles. However, water contamination in nitrogen tetroxide causes formation of nitric acid, which is corrosive to many metals; therefore, materials selected for use with nitrogen tetroxide should be compatible with nitric acid as well. Gold and a few types of stainless steel have been satisfactory in nitric acid storage containers. No bearing materials have been found satisfactory for use in nitric acid for long durations.

In general, aluminum alloys and stainless steels are most suitable for use as materials in contact with dry nitrogen tetroxide. The resistance to corrosion exhibited by the various aluminum alloys is a function of water content in the nitrogen tetroxide and the aluminum content of the alloy in question. As the water content in the nitrogen tetroxide exceeds 0.3 percent, highly alloyed aluminum (e.g., 7075 aluminum alloy) shows a sharp increase in corrosion rates. By contrast, the purer aluminum alloys (e.g., 1100 aluminum alloy) show much less increase in corrosion rates. For stainless steel, however, the corrosion rate in nitrogen tetroxide varies directly with water content.

Titanium propellant tank failures were experienced by one contractor. Other users experienced no problems. No conclusion can be reached on the use of titanium with nitrogen tetroxide on the basis of present preliminary engineering data. Copper, magnesium, and nickel alloys are not recommended for use because of their poor corrosive resistance to nitric acid.

Most nonmetallic materials show poor resistance to nitrogen tetroxide and are considered unsatisfactory for use. Reaction of nitrogen tetroxide with nonmetals can result in decomposition of the materials, causing degradation or complete destruction; or it can alter the physical properties, such as volume and/or hardness of the material. The propellant's physical characteristics may also be affected. Of all the plastics available for use, Teflon and Teflon products exhibit the best resistance to nitrogen tetroxide; however, nitrogen tetroxide permeates and is absorbed by Teflon. Results from permeability tests show that the permeability rate for Teflon TFE is three times greater than for Teflon FEP.

Most lubricants in contact with nitrogen tetroxide are either dissolved and washed off or undergo a substantial change in hardness. Dry lubricants Molykote Z, Drilube 703, and Electrofilm 66–C have been rated as compatible. Microseal 100–1 is rated as compatible and does not undergo any physical changes.

The formation of a gelatinous material

has been reported during high velocity flow of nitrogen tetroxide through small clearances. Additional information is presented by Salvinski et al. (ref. 1).

Table XII lists materials considered to be compatible with nitrogen tetroxide for long-term application. Temperatures, temperature ranges, and percent of water contamination are parameters of conducted tests and are not necessarily temperature limits or moisture content limits.

TABLE XII.—*Compatibility of Materials Tested With Nitrogen Tetroxide for Long-Term Application*

[Moisture content less than 0.2 percent unless otherwise noted]

Material	Test temperature, °F	Percent moisture content
Aluminum alloys:		
1060	80	0.2–1.0
1100	60	0.3
1100–0	60	0.2–1.0
2014–T6	60	0.2–1.0
2014–T6 (Hardas anodize)	60	
2014–T6 (H_2SO_4 anodize)	65	
2014–T6 (Iridite)	60	
2014–T6 (welded)		
2024	140	0.2–1.0
2024–0	150	
2219–T6	60	
2219–T81	60	0.2–1.0
3003–H14	150	0.6
4043	80	
5052	130	0.2–1.0
5086–H34	165	
5086–H36	65	
5254–F	65	
5456	60	
5456 H–24	60	0.2–1.0
5426 H–24 (Iridite)	60	
5456 H–321	65	0.2–1.0
6061	130	
6061–T6	130	0.2–1.0
6061–T6 (welded)	65	
6066	65	
7075	60	
7075–0	150	
7075–T6	150	
7075–T6	60	0.2–1.0
356	80	
356–T6	80	0.2–1.0
Tens 50	65	
Stainless steels:		
410	150	
416	65	
440C	100	
302	100	
303	80	0.2–1.0
304	140	0.2–1.0
304L	165	3.2
316	65	
321 (including welded)	60	3.0
347 (including welded)	100	10.0
17–4PH (condition A)	65	
17–4PH (H1000)	100	0.3
17–4PH	65	3.0
17–7PH (TH950)	100	
17–7PH (RH950)	100	
AM–350 annealed	100	10.0
AM–355 (condition H)	100	
Miscellaneous metals:		
Chromium plate	60	
Hayness Stellite 1	100	
Hayness Stellite 12	100	
Hayness Stellite 6K	65	
Hayness Stellite 21	65	
Hayness Stellite 25		
Hayness Stellite 93	100	
Gold	75	
Gold plate	60	
Cast iron	80	
Carbon steel	80	
Mild steel	140	
1020 steel	130	
A–285 (grade C)	165	3.2
8630 steel	140	
A286 (annealed) steel	100	
A286 (aged)	60	
PH15–7 Mo (condition A)	165	3.2
Magnesium, 100A	60	
Magnesium AZ31C	60	
Magnesium, HM21A–T8	60	
A-Nickel	65	
Nickel electroplate	60	
Electroless nickel plate	100	
Inconel	65	
Monel	65	

TABLE XII.—Continued

Material	Test temperature, °F	Percent moisture content
Miscellaneous metals—Con.		
Ni-Span-C	60	
Inconel X	75	
Hastelloy alloys	75	
Platinum	75	
718 Braze 6061–T6A1	65	
Pure tin solder on 303SS	65	
Easy Flo braze on 347SS	65	
Tantalum	75	
Tin	80	
Nonmetals:		
Microseal 100–1 on 2014–T6 A1	100	
Teflon TFE	75	
Teflon FEP	160	0.2–1.0
Teflon graphite	75	
Teflon MoS_2	75	
Teflon asbestos	75	
Teflon glass filled	80	
Armalon 7700	75	
Armalon 7700B	75	
Fluorobestos	60	
Fluogreen	60	
Genetron GCX–3B	80	
Genetron XE–2B	65	
Kynar	80	
Tedlar	67	
Kel-F (unplasticized)	160	0.2–1.0
Lubricants:		
XC 150	65	
Molykote Z	60	
Microseal 100–1	67	
LOX Safe	80	
Flake graphite	80	
Graphitar 2, 14, 39, 50, 86	67	
CCP–72		
Fluorolube MG6DO	80	
Fluoroethane G	80	

Materials Used in Valve Parts

Valve bodies. Stainless steels 302, 304, 316, 321, 347. Aluminum alloys (anhydrous only, attached by dilute nitric acid formed by combining NTO with water) 6061, 356T6, Tens 50, 3003, 2024. Titanium alloys (should be used with caution if high-impact loads could occur).

Springs. Stainless steels 301, 304, 321, 347, AM 350, AM 355, 17–4PH, 17–7PH; alloy steels 8630, A–286; Inconel, Inconel-X; Ni-Span-C.

TABLE XII.—Concluded

Stems. Stainless steels 321, 347, 403, 410; alloy steels A–286, 8630; René 41.

Bellows. Stainless steels 303, 321, 347; Inconel-X.

Bearings. Stainless steels 301, 301N, 410, 430, 440C.

Valving units (seats and poppets). Stainless steels 303, 347, 403, 410; Teflon; Haynes Stellite 25; aluminum 1100; vinylidene fluoride; polyvinyl fluoride. Nylon unsuitable.

Seals. Teflon; Kel-F 300, aluminum 1100, irradiated polyethylene, vinylidene fluoride, polyvinyl fluoride.

Packing. Teflon, Kel-F 300.

Lubricants. Teflon coatings, flake graphite, molybdenum disulfide, Kel-F 90, Microseal 100–1, silicone greases.

Bolts, nuts, and screws. Stainless steels 303, 321, 347, AM 350, AM 355, 17–4PH, 17–7PH; alloy steel A–286; Inconel-X.

Thread sealants and antiseize compounds. Unsintered Teflon; Redel N_2O_4 thread sealant; LOX Safe; Reddy Lube 100, 200; Drilube 822.

Coatings. Chrome plate (free of pinholes), gold. Avoid cadmium.

Diaphragms. Stainless steels 304, 321, 347; alloy steel 17–7PH. Mylar satisfactory for vapor exposure but unsuitable for liquid.

Braze alloys. Permabraze 130 (82 percent Au, 18 percent Ni).

Nitrogen Tetroxide (N_2O_4) Problem Areas

Organic polymers [*1–2*]. No completely satisfactory nonmetallic material has yet been found for use as valve seats. Most organic polymers have some disadvantages (principally that of swelling) when exposed to nitrogen tetroxide.

Wet lubes [*1*]. Most lubricants absorb nitrogen tetroxide, rendering them useless. Certain silicone greases have limited use because they tend to absorb nitrogen tetroxide slowly. However, these generally wash out under dynamic conditions.

Dry lubes [*2*]. A number of dry lubricants including graphite and molybdenum disulfide show good compatibility, but tend to wash out under flow conditions.

Effects of leakage [*1*]. Nitrogen tetroxide is highly toxic and is classified as a poison in ICC regulations. The maximum allowable concentration of 5 ppm in air is accepted by the American Conference of Governmental Hygienists. Because of its corrosive nature and reaction with, or absorption in organic materials, adjacent components may easily be affected by leakage. Its hypergolic nature with many fuels constitutes a potential fire problem. Effects of leakage in space are discussed under hydrazine.

Soft seats [*1–2*]. A limited number of nonmetallic materials are satisfactory for short-term use, but it is recommended that polymeric seals be avoided whenever possible. Teflon and Kel-F are the most resistant plastic materials. Teflon, while compatible with nitrogen tetroxide, absorbs the vapors slowly. The resultant swelling may result in component malfunction.

OXYGEN DIFLUORIDE

Physical Properties

Specific gravity	1.496 ($-288.8°$ F)
Molecular weight	54.00
Freezing temperature, °F	-370.8
Normal boiling point, °F	-228.6
Critical temperature, °F	-72.8
Critical pressure, psia	719

Oxygen difluoride (OF_2) is a colorless gas at room temperature and atmospheric pressure, condensing to a yellow liquid at $-229°$ F. It has a foul odor; 0.1 ppm in air may be detectable and 0.5 ppm in air is easily detected. It is regarded as a highly toxic gas.

A powerful oxidizing agent, similar to fluorine and the halogen fluorides, oxygen difluoride is generally considered to be much less reactive than fluorine. It can react with a majority of inorganic and organic compounds, provided sufficient activation energy is available. High exothermic heats of reaction are common; many are sufficiently energetic to cause ignition. Reports on hypergolicity of fuels such as hydrazine, ammonia, and monomethylhydrazine give varying conclusions; hence, the material should be treated as potentially hypergolic at all times. Oxygen difluoride and diborane are unquestionably hypergolic.

Oxygen difluoride is a relatively stable compound in that it does not detonate by sparking and was found to be insensitive to shock at $-320°$ F. It begins to decompose thermally, however, at approximately 480° F.

The materials compatibility data are quite limited, but show that no major problems are to be expected with metals. Laboratory experience has shown that rubber and certain plastics can be used with oxygen difluoride in moderate flow and temperature conditions. Service experience with valves containing fluorinated polymers such as Teflon, Aclar, and Halon in contact with the liquid phase is insufficient, but results are reported to be encouraging. Oxygen difluoride can be handled readily in most common metals and glass, with the choice dependent upon the service requirement. Metals such as stainless steel, copper, aluminum, Monel, and nickel may be used for gas and liquid service from cryogenic temperatures to approximately 400° F.

Table XIII lists the materials considered compatible with oxygen difluoride. Tests were conducted with liquid oxygen difluoride at $-320°$ F and gaseous oxygen difluoride at ambient temperature.

Oxygen Difluoride (OF_2) Problem Areas

Ceramics [*2*]. Oxygen difluoride has a strong oxidizing power similar to that of fluorine, and thus reacts with the vast majority of materials. Extreme care should be taken to select materials compatible with the propellant under the temperature extremes of its environment.

Organic polymers [*1–2*]. Fluorinated polymers such as Teflon, Aclar, and Halon have been used for limited service at moderate temperatures. At low temperatures, the loss of elasticity makes them impractical. Some data indicate that under certain conditions such as high flow, contamination, or impact, even the fluorinated polymers will react with oxygen difluoride.

Wet and dry lubes [*1*]. There is no known lubricant for use in contact with liquid or gaseous oxygen difluoride. Because of its high reactivity with organic materials,

TABLE XIII.—*Compatibility of Materials Tested With Oxygen Difluoride for Short-Term Use*

Material	Remarks
Aluminum alloys:	
2024–T3	([a]).
2024–T3 (Alclad)	([a]).
Stainless steel:	
301	([a]).
304	
Maraging AM 355	(No change for gas at ambient temperature.)[a]
Maraging AM 367	([a]).
Other metals:	
Beryllium copper (2 percent)	([a]).
Inconel X750	
Magnesium AZ 31B, H24	([a]).
Monel 400	([a]).
Monel K500	([a]).
Titanium alloy 5Al–2.5 Sn	
Nonmetals: Glass	Above 390° F, glass is attacked by OF_2.

[a]Fluoride coatings formed, but weight gain was considered insignificant.

Materials Used in Valve Parts

Valve bodies. Stainless steels 304, 304ELC, 321, 347; Monel, K-Monel; aluminum alloys 356T6, M517, 359T6, 6061, 5052, 3001, Tens 50.

Springs. Stainless steels 304ELC, 321, 347; Inconel, Inconel-X, Inconel-W; K-Monel.

Stems. Stainless steels 321, 347, 403, 410, 422; K-Monel; René 41.

Bellows. Stainless steels 304ELC, 321, 347; Monel, K-Monel; Inconel-X.

Bearings. Stainless steels 301, 301N; aluminum 6061; hard anodized copper.

Valving units (seats and poppets). Stainless steels 321, 347, 403, 410, 422; Monel; copper; aluminum 1100; Kel-F 81.

Seals. Beryllium-copper, copper, aluminum, brass, lead, 50–50 tin-indium alloy and tin, Kel-F 81, Teflon (avoid impact), vinyl silicone.

Packing. Copper, pure tin, Teflon, Kel-F.

Lubricants. Molybdenum disulfide.

Bolts, nuts, and screws. Stainless steels 304, 321, 347; Inconel-X; Monel, K-Monel.

Thread sealants and antiseize compounds. Unsintered Teflon and Permatex Nos. 2 and 3 applied to all but the first two threads of the male fitting.

Coatings. Hard nickel plate, chrome plate, anodize (aluminum).

TABLE XIII.—Concluded

Diaphragms. Stainless steels 304ELC, 321, 347; Monel, K-Monel; beryllium copper. Mylar is unsatisfactory.

conventional lubricants should definitely be avoided. Even the most likely candidates, the normally unreactive perfluorinated hydrocarbon lubricants, are degraded. All valves should be designed to eliminate the need of a lubricant in intimate contact with this oxidizer.

Soft seats [*1*]. Soft-seat materials, at the low temperatures of liquid oxygen difluoride, lose elasticity, become brittle, and are thus not suitable. Valves useful with this oxidizer should be of the all-metal seat construction.

Effects of leakage [*1*]. Oxygen fluoride displays properties similar to those of liquid fluorine, but is generally considered to be less reactive and easier to handle. It is relatively stable in that it does not detonate by sparking, but it begins to decompose thermally at elevated temperatures (about 480° F). Tests of reactivity to fuels such as hydrazine, ammonia, and monomethylhydrazine have given varied results as to its hypergolic nature. It should be assumed that the oxidizer is hypergolic with all fuels because of its very strong oxidizing power. The data on toxicity are quite limited, but oxygen difluoride must be regarded as highly toxic. Space leakage problems are similar to those discussed for hydrazine.

CHLORINE TRIFLUORIDE

Physical Properties

Specific gravity	1.83 (60° F)
Molecular weight	92.46
Freezing temperature, °F	−105
Normal boiling point, °F	53
Critical temperature, °F	345
Critical pressure, psia	838
Heat of vaporization, Btu/lb_m	128

Chlorine trifluoride (ClF_3), like fluorine, is among the most active chemicals known. It reacts vigorously with most oxidizable substances at room temperature and with most common metals at elevated temperatures. Under ordinary conditions, chlorine trifluoride reacts violently with water or ice. It is, however, insensitive to mechanical shock, nonflammable in dry air, and shows good thermal stability at ambient temperatures.

The corrosion resistance of all materials of construction used with chlorine trifluoride depends on the formation of a passive metal-fluid film to protect the metal from further attack. Some metals such as Monel, copper, nickel, stainless steel, etc., form such a metal-fluoride film. Among the metals mentioned, Monel and nickel are preferred because of their resistance to hydrogen fluoride and hydrazine chloride, which are formed by the reaction of chlorine trifluoride with water. Aluminum alloys, 18–8 stainless steels, and K-Monel have been used for valve bodies; type 321 and 347 stainless steels and Monel have been used for bellow materials. Gaskets have been made from sterling silver and lead indium alloys. Tin, indium carbon, and boron carbide have been used for rotating seals.

Table XIV lists materials which are considered to be compatible with chlorine trifluoride under most conditions for long-term application. These materials must be thoroughly cleaned and passivated (in the case of metal), however, to insure a contamination-free surface. All chlorine trifluoride systems must also be dry and leakproof.

TABLE XIV.—*Compatibility of Materials Tested With Chlorine Trifluoride* (ClF_3) *for Long-Term Application*

Material	Test temperature, °F
Aluminum alloys:	
1060	85
1100	85
2024	85
3003	85
5052	85
6061 welded	85
6063	----------
6066	----------
356	----------
Tens 50	----------
Stainless steels:	
301	----------
302	----------
303	85
304	85
316	85
321	----------
347	85
Other metals:	
A-Nickel	85
Copper	85
Incoloy	85
Inconel	85
Indium	----------
Lead-indium alloy	----------
Tin-indium alloy	----------
Magnesium AZ–31B	85
Magnesium HM–21A	85
Magnesium HK–31A	85
Monel	85
K-Monel	----------
Nitralloy	----------
Silver solder	----------
Sterling silver	----------
Tin	----------
Nonmetals:	
Boron carbide	----------
Carbon	----------
Kel-F (under static conditions only)	----------
Teflon (under static conditions only)	----------

NOTE: Materials listed above are rated compatible based on a corrosion rate of less than 1 mil per year and in the case where the material does not cause decomposition, and is free from impact sensitivity. Nonmetals are rated for satisfactory service for general use.

Materials Used in Valve Parts

Valve bodies. Stainless steels 304, 304ELC, 321, 347, AM 350; Monel, K-Monel; aluminum alloys 356T6, M517, 6061, 5052, 3001, 2024, 7075, Tens 50; magnesium alloy AZ31B. Titanium unsatisfactory.

Springs. Stainless steels 302, 304ELC, 321, 347, AM 350; alloy steel A–286; Inconel, Inconel-X, Inconel-W, K-Monel.

Stems. Stainless steels 321, 347, 410, 403, 422, AM 350; alloy steel A–286; K-Monel; René 41.

Bellows. Stainless steels 304ELC, 321, 347; Monel, K-Monel.

Bearings. Stainless steels 301, 301N; aluminum 6061; hard anodized copper.

Valving units (seats and poppets). Stainless steels 321, 347, 410, 403, 422; Monel; copper; aluminum 1100; titanium carbide.

TABLE XIV.—Concluded

Seals. Beryllium-copper, aluminum 1100, brass, copper, lead, 50–50 tin-indium alloy and tin, Teflon (nonflow), Kel-F (nonflow).

Packing. Copper, pure tin, Teflon.

Lubricants. Molybdenum disulfide.

Bolts, nuts, and screws. Stainless steels 304, 321, 347, AM 350; alloy steel A–286; Monel, K-Monel; Inconel-X.

Thread sealants and antiseize compounds. Unsintered Teflon and Permatex Nos. 2 and 3 applied to all but the first two threads of the male fitting.

Coatings. Hard nickel plate, chrome plate, anodized (aluminum).

Diaphragms. Stainless steels 304ELC, 321, 347; Monel, K-Monel; beryllium copper.

Chlorine Trifluoride (ClF_3) Problem Areas

Organic polymers [*1–2*]. Most organic polymers undergo spontaneous ignition and/or absorption of chlorine trifluoride to form detonable mixtures. Hence use of components incorporating plastic materials is not recommended. Teflon and Kel-F have been found acceptable under static nonflow conditions, but they may ignite when heated.

Wet and dry lubes [*1*]. No completely satisfactory lubricant is known. Most lubricants ignite spontaneously and/or form detonable mixtures with chlorine trifluoride.

Effects of leakage [*1*]. Leakage cannot be tolerated in valves used for chlorine trifluoride. Although it is nonflammable in air and exhibits excellent thermal stability at ambient temperatures, it is an extremely hazardous propellant due to its toxicity and extreme reactivity with the vast majority of organic and inorganic compounds. At elevated temperatures it will react vigorously with most common metals; the propellant readily ignites organic materials such as solvents and lubricants. Space leakage is discussed under hydrazine.

Soft seats [*2*]. Soft seats made of plastic are unsuitable for service with chlorine trifluoride.

Hard seats [*2*]. A very limited number of soft metals (principally aluminum 1100 and copper) have been found satisfactory for valve-seat use. The seats should be thoroughly cleaned and the propellant passivated prior to installation.

PERCHLORYL FLUORIDE

Physical Properties

Specific gravity	1.69 (−52.2° F)
Molecular weight	102.457
Freezing temperature, °F	−231
Normal boiling point, °F	−52.2
Critical temperature, °F	202
Critical pressure, psia	779
Heat of vaporization, Btu/lb_m	84.0

Perchloryl fluoride ($FClO_3$ or ClO_3F) is a colorless gas at ambient conditions with a characteristic sweet odor. Under pressure or low temperature it is storable in liquid form. The toxic action of perchloryl fluoride is derived from its pronounced oxidizing properties and results in respiratory irritation, oxidation of hemoglobin, and absorption of fluoride into the body. A threshold limit value has not been established, but an interim working value of 3 ppm is suggested.

Perchloryl fluoride is thermally stable up to 849° F in absence of air. It is nonflammable, but being a strong oxidizing agent, readily supports combustion of many organic materials. Although not shock sensitive itself, in combination with porous organic or inorganic materials it can produce a potentially shock-sensitive mixture. Combinations of perchloryl fluoride and most rocket fuels are explosion hazards.

The corrosion resistance of metals of construction depends largely on the quantity of moisture present. Hence selection must be governed by the moisture content. Under moist conditions, types 304, 310, and 314 stainless steels have shown good resistance at room temperature.

Organic plastics should be avoided entirely with the exception of the fluorinated plastics. Under mild conditions of heat shock, Teflon, Kel-F, and Kynar can be used, but may undergo structural changes when moderate amounts of perchloryl fluoride are absorbed. These plastics should not be used under dynamic flow conditions such as would occur in valves, since the resultant swelling of seal materials would present problems of physical interference.

Table XV lists materials suitable for use with perchloryl fluoride.

TABLE XV.—*Compatibility of Materials Tested With Perchloryl Fluoride for Long-Term Application*

Material	Test temperature, °F
Aluminum alloys:	
1060	85
1100	85
2024	85
3003	85
5052	85
6061	85
7079	85
Stainless steels:	
304	85
316	85[a]
347	85
Carpenter No. 20–Cb	85[a]
PH 15–7 Mo (condition RH 950)	85
PH 15–7 Mo (condition TH 1050)	85
AM 350 (welded)	85
Copper alloys:	
Aluminum bronze, 8 percent, Ampco 8	85
Beryllium copper, 2 percent	85
Nickel silver, 18 percent, Alloy A	85
Phosphor bronze, 5 percent Grade A	85
Rule brass	85
Yellow brass	85
Magnesium alloys:	
AZ–31B	85
HK–31A	85
HM–21A	85
Nickel alloys:	
"A" Nickel	85
Inconel	85[a]
Incoloy	85[a]
Monel	85
Low carbon steels:	
1010	85
1010 (coated w/Fosbond 40)	85
1010 coated w/Fosbond 27)	85
Miscellaneous metals:	
Gold	85[a]
Platinum	85[a]
Silver	85[a]

[a]Moisture content to 1 percent.

TABLE XV.—Concluded

NOTE: Metals listed above are rated compatible based on a corrosion rate of less than 1 mil per year and in the case where the material does not cause decomposition, and is free from impact sensitivity. Nonmetals are rated for satisfactory service for general use.

Materials Used in Valve Parts

Valve bodies. Stainless steels 304, 310, 314, 316, 321; Hastelloy B, Hastelloy C; Monel; Durimet 20.

Springs. Stainless steels 304, 321.

Stems. Stainless steel 321.

Bellows. Stainless steels 304, 321; Monel.

Bearings. None.

Valving units (seats and poppets). Stainless steels 304, 321; Kel-F; Teflon.

Seals. Viton B, Teflon, Kel-F.

Packing. Teflon, Kel-F.

Lubricants. Fluorolubes.

Bolts, nuts, and screws. Stainless steels 304, 321; Monel.

Thread sealants and antiseize compounds. Teflon tape.

Coatings. None.

Diaphragms. Stainless steels 304, 321; Teflon.

Perchloryl Fluoride ($FClO_3$) Problem Areas

Metals [2–3]. Anhydrous perchloryl fluoride is less reactive and corrosive than other halogen-containing oxidants at ordinary temperatures. It is not reactive to most common metals, and selection of a metal for use with this propellant should be governed by the moisture content. In the presence of water, perchloryl fluoride is corrosive to most metals. The readily oxidized metals will burn in perchloryl fluoride under severe conditions. Therefore extreme care must be taken in selecting metals and thorough knowledge is needed of propellant purity, compatibility, and potential environmental extremes.

Organic polymers [2]. Many organic materials do not react with perchloryl fluoride at ambient temperature, but if ignited, will burn violently. Some are hypergolic. Hence, materials compatibility is extremely important in the selection of a polymeric material. Teflon and Kel-F appear to be resistant to attack, but tend to absorb the fuel. Some phenolic and epoxy resins have found limited use.

Wet and dry lubes [1–2]. Fluorocarbons are the only suitable lubricants. Perchloryl fluoride must not be brought into contact with any other conventional valve grease or oil.

Effects of leakage [1–2]. Perchloryl fluoride leakage represents an acute toxic hazard. Although a threshold limit value has not been established, an interim working value of 3 ppm has been suggested. Because of its strong oxidizing effect, it readily supports combustion with oxidizable materials such as organic compounds; it is hypergolic with some fuels such as hydrazine, and in combination with most other fuels may detonate. Space effects of leakage are discussed under hydrazine.

Soft seats [2]. Teflon or Kel-F seats are recommended for limited use; however, consideration must be given to their ability to absorb perchloryl fluoride.

LIQUID HYDROGEN

Physical Properties

Liquid hydrogen (H_2) is colorless and odorless. It normally does not present an explosive hazard when it evaporates and mixes with air in an unconfined space. An unconfined mixture of hydrogen gas and air will burn, however, if exposed to an ignition source such as a spark. Liquid hydrogen is not in itself explosive, but reacts violently with strong oxidizers. If contaminated with oxygen, it becomes unstable and an explosion is liable to occur. Reaction with fluorine and chlorine trifluoride is spontaneous. Two species of liquid hydrogen exist: ortho-hydrogen and para-hydrogen.

At the temperature ($-423°$ F) at which hydrogen is a liquid, corrosive attack on materials is not considered an important factor in selecting materials. A more important factor in selecting the materials for use with liquid hydrogen is the embrittlement of some materials by the low temperature of the liquid. This necessitates selection of materials on the basis of their structural properties; i.e., yield strength, tensile strength, ductility, impact strength, and notch sensitivity. The materials must also be metallurgically stable so that phase changes in the crystalline structure will not result from time or temperature cycling. Body-centered metals such as low-alloy steels undergo a transition from a ductile to brittle behavior at low temperatures and therefore are not generally suitable for structural applications at cryogenic temperatures. Face-centered metals such as the austenitic stainless steels normally do not show a transition from a ductile to a brittle behavior at low temperatures. For this reason, these types of materials are desirable for use in cryogenic applications, but care should be exercised in selection of face-centered metals. Low-temperature toughness is not a characteristic of all face-centered metals, nor of all conditions of a specific metal. For example, severely cold-worked or sensitized (carbide precipitation at grain boundaries) austenitic stainless steels can become embrittled at low temperatures.

Table XVI lists materials considered compatible with liquid hydrogen for long-term application.

TABLE XVI.—*Compatibility of Materials Tested With Liquid Hydrogen* (H_2) *for Long-Term Application*

Aluminum alloys:	Other metals:
1100	Molybdenum
1100T	Nickel
2024T	Monel
4043	Inconel
5052	Low carbon steel
Stainless steel:	High nickel steel
301	Titanium
302	Nonmetals:
303	Nitryl rubber
304	Silicone rubber
304L	Teflon
316	Garlock packing
321	Bakelite
347	Micarta
410	Lucite
Haynes 21	Graphite

NOTE: The materials were rated compatible primarily for their embrittlement properties at cryogenic temperatures. Nonmetals shown as being compatible should be restricted for "warm" joint application or equivalent.

TABLE XVI.—Concluded

Materials Used in Valve Parts

Valve bodies. Stainless steels 301, 302, 304, 310, 316, 321, 347; K-Monel; Hastalloy B; aluminum alloys 2014T6, 6061T6, 5456H–24, 5052, 2024, 5154, 5086; titanium; alloy steel N–155.

Springs. Stainless steels 301, 321, 347; alloy steel A–286; K-Monel; Inconel, Inconel-X.

Stems. Stainless steels 321, 347; alloy steel A–286; Haynes No. 25; K-Monel; Inconel-X.

Bellows. Stainless steels 321, 347; K-Monel; Inconel-X.

Bearings. Stainless steels 440C, 52100, 410.

Valving units (seats and poppets). Stainless steels 321, 347; Teflon; Kel-F; copper; aluminum 1100; Monel; Stellite 21; nylon.

Seals. Stainless steels 321, 347; Teflon; Kel-F; silicone rubber (static seals); aluminum 1100.

Packing. Teflon; Kel-F.

Lubricants. Teflon coatings and molybdenum disulfide. Halogenated oils may be used for installation only.

Bolts, nuts, and screws. Stainless steels 304, 321, 347; alloy steel A–286; Inconel-X.

Thread sealants and antiseize compounds. LOX Safe.

Diaphragms. Mylar, Teflon.

Liquid Hydrogen (H_2) Problem Areas

Organic polymers [*1–2*]. The ability of polymeric materials to maintain satisfactory physical properties and to withstand thermal stress caused by large temperature changes is of prime importance. Organic plastics are thus limited to service where the embrittling effect at the low temperatures of liquid hydrogen is minimized. Lip seals constructed of Teflon and Kel-F, static seals of some rubbers, and diaphragm seals from Mylar have proved serviceable. Studies at the National Bureau of Standards have shown that some elastomeric seals under compression retain their sealing capabilities effectively well below the normal brittle point of the polymer without compression.

Wet and dry lubes [*1*]. Lubricants are generally not practical in the presence of liquid hydrogen, since they solidify and become brittle.

Lubricity [*2*]. Satisfactory low-load, short-life bearing and gear operation was found when liquid hydrogen was used as a lubricant. The 22 metals employed in this study were rated as to their suitability as a gear or bearing material by the degree of degradation resulting during the tests (ref. 3).

Effects of leakage [*2*]. Leakage of hydrogen gas does not appear to present unusual problems, since the composition limits for combustion are wide and the gas tends to dissipate rapidly. Proper ventilation and elimination of sources of ignition reduce danger of detonation. An explosion hazard exists when the hydrogen-air mixture is completely or partially confined. Effects of leakage in space are discussed in the section on hydrazine.

Disconnects [*2*]. See comments under "Liquid Oxygen."

LIQUID OXYGEN (LOX)

Physical Properties

Liquid oxygen (LOX) is a nontoxic, nonflammable, and nonexplosive oxidizing agent having much lower reactivity than gaseous oxygen. Mixing of liquid oxygen with a hydrocarbon fuel will cause the latter to solidify, the resulting mixture being extremely shock sensitive.

Most metals are not corroded by liquid oxygen; however, the low temperature of liquid oxygen ($-300°$ F) does not cause embrittlement of some metals, especially in the body-centered ferrous alloys. The alloys most commonly used in liquid-oxygen-handling equipment are nickel, Monel, copper, aluminum, the 300-series of stainless steels, brass, and silver solder. Several violent reactions of titanium and liquid oxygen, presumably due to impact, have been reported. On the other hand, titanium alloys have been successfully used for helium pressure bottles in contact with liquid oxygen in the Titan missile, and for the liquid oxygen pressure bottles in the X–15 rocket planes. It has been postulated that a gas phase is needed to initiate the reaction, but this hypothesis is still under investigation. Apparently the surface conditions of cleanliness and smoothness are of major importance in reducing the danger of detonation.

Thus, the use of titanium in contact with liquid oxygen must be carefully investigated with respect to specific conditions. Impact studies have also shown some reactivity of liquid oxygen with zirconium and aluminum.

All valves require insulation to avoid evaporation losses. Drainage of condensed water on valves should be provided to avoid water entrapment, which may cause corrosion.

Materials compatible with liquid oxygen are given in table XVII.

TABLE XVII.—*Compatibility of Materials Tested With Liquid Oxygen (LOX) for Long-Term Application*

Materials Used in Valve Parts

Valve bodies. Stainless steels 304, 310, 316, 321, 347; K-Monel; Hastelloy B; aluminum alloys 2014T6, 6061T6, 5456H–24, 5154, 5052, 5086, 356T6, 6061; alloy steel N–155.

Springs. Stainless steels 321, 347; alloy steel A–286; K-Monel; Inconel, Inconel-X.

Stems. Stainless steels 321, 347; alloy steel A–286; Haynes No. 25; Inconel-X.

Bellows. Stainless steels 304, 321, 347; K-Monel; Inconel-X.

Bearings. Stainless steels 440C, 52100.

Valving units (seats and poppets). Stainless steels 321, 347; Teflon; Kel-F; aluminum 1100; Monel.

Seals. Stainless steels 321, 347; Teflon; Kel-F; aluminum 1100.

Packing. Teflon; Kel-F.

Lubricants. Teflon coatings and molybdenum disulfide. Halogenated oils may be used for installation only.

Bolts, nuts, and screws. Stainless steels 321, 347; alloy steel A–286; Inconel-X.

Thread sealants and antiseize compounds. LOX Safe.

Coatings. Chromium, nickel, anodize (aluminum).

Diaphragms. Stainless steels 321, 347; Teflon; beryllium copper; Mylar.

Liquid Oxygen (O_2) Problem Areas

Organic polymers [*2*]. Organic materials should be avoided whenever possible with both liquid and gaseous oxygen because of the possibilities of explosion. No tests have given a reliable compatibility rating for organic materials in liquid oxygen. Although there are several lists of organic materials rated as suitable for use with liquid oxygen, specific conditions, compatibility, impact sensitivity, and embrittlement at cryogenic temperatures should be thoroughly and carefully studied before any organic material is utilized. Sources of energy may also be from operation of mechanical parts, such as (1) heat produced by friction of metal surfaces, (2) heat from shearing of liquids, (3) shock waves, and (4) heat generated by the catalytic breakdown of an organic material in contact with the metal surface, etc.

The most reliable organic materials for liquid oxygen applications are the fluorinated organic compounds (the more highly fluorinated the compound, the more stable[2] to attack by liquid oxygen). In special applications many other organic compounds probably can be used. However, investigations with satisfactory testing procedures are needed before organic materials can be used with liquid oxygen with any great assurance of success.

Currently, no single test or group of tests reported gives a reliable compatibility rating for materials suitable for use with liquid oxygen because of difficulty in determining impact sensitivity. Much of the data available were based on a 70-ft-lb acceptance level for impact sensitivity. Apparently, this was an arbitrary requirement based on the impact threshold of a particular lubricant which, at the time, was considered to be the only safe available lubricant. Because of the lack of a technical basis for establishment of 70 ft-lb as an acceptance test parameter, and because the size and shape of the sample and the design of the testing machine affect the detonation results, little value can be assigned to published compatibility tables. For this reason, none is included in this report.

Wet and dry lubes [*1*]. No completely compatible lubricants have been found. Selected perfluorinated hydrocarbon and dry

[2] The word "stable" here is more applicable than "resistant" because it deals with impact sensitivity and not resistance to corrosion.

lubricants have found limited use; however, the possibility of detonation still exists.

Lubricity [2]. Satisfactory low-load, short-life bearing and gear operation was possible where liquid oxygen was used as the lubricant. While some of the metals of construction proved satisfactory for use with liquid oxygen, others, under stress, tended to become brittle.

Effects of leakage [2]. Liquid or gaseous oxygen does not display hazards similar to those encountered with most other oxidizers. Pure oxygen is incapable of burning or detonating, but mixed with a material that will burn, it intensifies the resulting fire. Gaseous mixtures of oxygen and fuels form a potentially dangerous mixture which can be ignited by any form of spark. Liquid oxygen, when in contact or mixed with any form of combustible material such as fuels, wood, plastics, oil, lubricants, or paper, forms an explosive mixture. Even when frozen these mixtures may be detonated by a spark, static electricity, mechanical shock, or similar energy source. Oxygen is nontoxic, but contact with liquid or gaseous oxygen close to its boiling temperature will cause severe frostbite. Space conditions and leakage are discussed in the section for hydrazine.

Soft seats [2]. Because of the composition of soft-seat materials, they should be avoided in liquid oxygen service. Although some fluorinated plastics have been found reliable, doubt will remain about compatibility until satisfactory testing procedures are available.

Disconnects [2]. Pneumatically operated disconnect valve missiles have presented problems due to icing from condensed water vapor in the atmosphere, requiring extensive effort to disengage them. In space transfer of the propellant, "freezeup" should not be a problem due to the absence of water vapor in space. Disconnects of the pullaway type during liftoff caused few problems, except for minor leakage.

LIQUID FLUORINE

Physical Properties

Specific gravity	1.69 (−52.2° F)
Molecular weight	102.457
Freezing temperature, °F	−231
Normal boiling point, °F	−52.2
Critical temperature, °F	202
Critical pressure, psia	779
Heat of vaporization, Btu/lb_m	84.0

Fluorine (LF_2) is the most powerful chemical oxidizing agent known. It reacts with practically all substances, the few exceptions being the inert gases, some metal fluorides and a few uncontaminated fluorinated organic compounds. It exhibits excellent thermal stability and resistance to catalytic breakdown, thereby presenting little or no problem in these areas. Compatibility ratings, therefore, are based primarily on the reaction of the fluorine with the various materials used.

Although fluorine is the most chemically active of all elements, many of the common metals can be considered for use in liquid fluorine systems.

Fluorine is a liquid at atmospheric pressure only in the short temperature range of −306° to −363° F, therefore requiring insulation of all valves. At these low temperatures chemical reactions in general tend to take place rather slowly. Another factor responsible for the low rate of attack by liquid fluorine on the common metals is that protective films of fluoride compounds tend to form on metal surfaces and act as a barrier to further reaction. The effectiveness of the protective film on the metals is based on the solubility of the various metal fluorides that form in the film in fluorine. It is believed that, as a protective film builds up and the rate of reaction slows down, an equilibrium between reactive rate and solubility of the film will be reached and a relatively steady corrosion rate will result. Service data indicate that the fluorides of nickel, copper, chromium, and iron are relatively insoluble in liquid fluorine. Metals such as Monel, nickel, and stainless steels exhibit satisfactory performance in liquid fluorine and indications are that much lower rates of corrosion can be expected for long-term exposure where equilibrium rates are reached than for short-term laboratory exposure. Several lightweight metals such as the alloys of aluminum, titanium, and magnesium are also known to produce protective films in liquid fluorine. Of these, titanium probably exhibits the lowest rate of corrosion; however, tests have shown it to be impact sensitive in fluorine. Actual service data and extrapolation of existing data are needed because of the lack of solubility data for fluorine compounds and corrosion rates for long periods of exposure.

Other factors to consider in selecting materials for use in a liquid fluorine system are: (1) flow rates, (2) water contamination, and (3) mechanical properties of the material at the low temperatures experienced with liquid fluorine. The rate of flow of the liquid fluorine in a valve and through an orifice is considered an important factor in maintaining the protective film on the materials being attacked. Fluoride coatings on some metals that are less resistant to fluorine, such as low-alloy steels, are sometimes very brittle or porous and powdery. High flow rates tend to remove these coatings and thus increase corrosive action. In restricted flow applications, "flaking" of the coating may result in contamination of the propellant, thus creating leakage problems at the valve seat.

Teflon has withstood exposure to liquid fluorine in a static condition. However, Teflon tends to react with fluorine to break down the polymers and form unsaturated low-molecular-weight fluorocarbons which do not adhere to the surface. Any flow of the propellant or movement of material over the surface of Teflon will remove these fluorocarbons, thus leaving them valueless as a protective film. Impurities introduced in the manufacturing process appear to be responsible for accelerated decomposition of Teflon in liquid fluorine service.

Fluorine will react with any water present in the system to form hydrofluoric acid. This acid tends to attack some materials that are normally resistant to uncontaminated fluorine. Of all the metals showing resistivity to fluorine attack, Monel is generally preferred because of its inherent resistivity to the hydrofluoric acid.

In selecting materials for fluorine systems, consideration should also be given to the effects of low-temperature environment on the mechanical properties of the materials. Some metals, such as the martensitic stainless steels, become brittle at these low temperatures.

Table XVIII lists materials considered to be compatible for service with liquid fluorine. However, as previously stated, insufficient information on prolonged usage of these materials in liquid fluorine restricts any rating for long-term application. Also, before using any material with fluorine, ex-

TABLE XVIII.—*Compatibility of Materials Tested With Liquid Fluorine*

Material	Temperature, °F	Class
Aluminum alloys:		
1100	−320	I
2017	−320	II
5052	−320	II
6061	−320	I
7079	−320	I
Stainless steels:		
304	−320	I
316	−320	1
347	−320	I
410	−320	I
420	−320	I
PH 15-7 Mo	−320	I
AM 350-C, Cx, D, DX	−320	I
Carpenter 20	−320	I
Other metals:		
A-nickel	−320	I
Brass (yellow)	−320	I
Brass (casting)	−320	II
Copper	−320	I
Copper—10 percent nickel	−320	I
Copper—30 percent nickel	−320	I
Everdur 1010	−320	I
Magnesium alloy AZ-31	−320	I
Magnesium alloy HK-31	−320	I
Monel	−320	I

TABLE XVIII. (Continued)

NOTE: Materials listed above are rated compatible based on corrosion resistance and shock sensitivity; they do not include effects of cryogenic temperatures on mechanical properties.

Class I—Metals listed above are rated compatible based on a corrosion rate of less than 1 mil per year and in the case where the material does not cause decomposition, and is free from impact sensitivity. Nonmetals are rated for satisfactory service for general use.

Class II—Materials which exhibit corrosion rates as great as 5 mils per year.

Materials Used in Valve Parts

Valve bodies. Stainless steels 304, 304ELC, 316, 321, 347; Monel, K-Monel; bronze; aluminum alloys 356T6, M517, 359T6, 2024, 7075, 6061, 5052, 3001, Tens 50; magnesium alloys HK31, A31.

Springs. Stainless steels 301, 304ELC, 321, 347; Inconel, Inconel-X, Inconel-W; K-Monel.

Stems. Stainless steels 321, 347, 403, 410, 422; K-Monel; René 41, PH15–7 Mo.

Bellows. Stainless steels 304ELC, 321, 347; Monel, K-Monel; Inconel-X.

Bearings. Stainless steels 301, 301N; aluminum 6061; hard anodized copper.

Valving units (seats and poppets). Stainless steels 321, 347, 403, 410, 422; Monel; copper; aluminum 1100; brass.

Seals. Beryllium-copper, aluminum 1100, brass, copper, lead, 50–50 tin-indium alloy and tin.

Packing. Copper, pure tin.

Lubricants. Molybdenum disulfide.

Bolts, nuts, and screws. Stainless steels 304, 321, 347; Monel, K-Monel; Inconel-X.

Thread sealants and antiseize compounds. Unsintered Teflon tape and Permatex Nos. 2 and 3 applied to all but the first two threads of the male fitting.

Coatings. Hard nickel plate, chrome plate, anodize (aluminum).

Diaphragms. Stainless steels 304ELC, 321, 347; Monel, K-Monel; beryllium-copper.

treme care should be exercised in cleaning the material thoroughly to remove all possible contamination. It is also recommended that cleaning be followed by a pretreatment or conditioning treatment, which exposes the material at room temperature to pure fluorine gas or a mixture of fluorine diluted with an inert gas. This is believed to initiate the formation of a relatively inert fluoride film. The use of a diluted gas reduces the violence of the reaction that may take place with any traces of contamination remaining after cleaning. This treatment will help the material withstand attack by full strength fluorine with much less reaction.

Liquid Fluorine (F_2) Problem Areas

Ceramics [*2*]. No known ceramic materials are completely satisfactory for use with liquid fluorine. Some have found limited use. Ceramics such as calcium fluoride and alumina are resistant to attack by gaseous fluorine even at high temperatures but lack mechanical strength. Both materials exhibited corrosion rates lower than metals tested at the same temperature, and can be used where physical properties of ceramic materials can be tolerated.

Organic polymers [*1–2*]. Organic polymeric materials are generally not suitable for service in liquid fluorine. Some fluorinated polymers have found limited service at moderate pressures with gaseous fluorine. Impurities introduced in the manufacturing process appear to be responsible for accelerated decomposition of Teflon in liquid fluorine service.

Wet and dry lubes [*1*]. Fluorine reacts with organic, aqueous, and siliceous materials normally considered inert, as well as with all oxidizable materials. There are no reliable lubricants for fluorine service.

Effects of leakage [*1*]. Leakage of fluorine represents potential toxic, fire, and explosion hazards. The threshold limit of fluorine is 0.1 ppm; it reacts vigorously with most substances at ambient temperatures, frequently with immediate ignition. Space leakage of propellant is discussed in the section on hydrazine.

Soft seats [*1–2*]. Under static conditions, fluorocarbon plastics are satisfactory for liquid fluorine service. Limited applications of some fluorinated soft-seat materials (Teflon) have been found for dynamic-flow conditions; however, it is recommended that the material be shielded and a minimum of surface area exposed to liquid fluorine.

Hard seats [*2*]. Valves having metal-to-metal seats have been used quite extensively in handling fluorine. It is desirable to use dissimilar materials to obtain a good seal and to prevent galling or binding when the valve is operated. Because metal valve seats are not as leaktight as those which utilize a resilient material, double valving is recommended for critical applications.

Disconnect [*2*]. Further availability tests and evaluations are required before a more realistic prediction of performance can be made. Icing due to water vapor in the atmosphere can cause problems in disengagement of the separable units.

LIQUID GELLED PROPELLANTS

Physical Properties

A gelled propellant is one that exhibits non-Newtonian properties. So many propellants have been gelled (e.g., hydrazine RP–1, nitrogen tetroxide, etc.) that it is impossible to make any generalized statements about the compatibility of the gels, as such, with other materials. However, as a rule the additives needed to cause gelling (carbon black, silica, etc.) are not reactive and do not modify the chemical characteristics of the parent propellant, so the compatibilities of a gel can be considered to be the same as those of the parent propellant.

One type of gelled propellant is characterized by thixotropic properties; that is, the viscosity of the propellant decreases with increasing shear and stress decreases with time at constant shear. These properties are of interest for several reasons. For example, when the propellant is locked in a tank, no shear is being applied and the propellant is very viscous. When pressure is applied and the propellant valve is opened, shear stresses are induced, viscosity is reduced, and the propellant flows. As the propellant flows through the tank, valves, feed lines, and injector orifices, the shear becomes greater, the viscosity becomes less, and the propellant behaves more like a low-viscosity liquid. From the standpoint of safety, the highly viscous state of the propellant when locked in a nonpressurized tank is advantageous.

The properties of gelled propellants also present some problems. For example, a relatively thick coating of propellant may adhere to tank and valve walls. Pressure drops through lines, valves, and other components are larger than those of comparable liquids. Valve maintenance and cleaning may be difficult if there are inaccessible cavities where gel may collect. Gels may break down under the influence of radiation. Excessive cold working (i.e., pumping into run tanks, overmixing, etc.) can reduce the strength of the gel to an unacceptable value. Evaporation of the propellant can leave a fine powdery residue which could interfere with the normal action of moving parts such as valve stems.

Gelled propellant systems will probably be schematically similar to liquid propulsion systems that perform the same task. In general, the interaction between typical valve configurations and gelled propellants is not overly severe and the associated problem areas can be circumvented by proper design techniques.

Liquid Gelled Propellant Problem Areas

Metals and ceramics [*2–3*]. It is presumed that metals compatible with the liquid constituent will serve equally as well for gelled propellants.

Organic polymers [*2*]. As with metals, it is necessary to select polymeric material based on the compatibility of the liquid constituent because the gelling agents are present in small quantity and are normally inert.

Wet and dry lubes [*1–2*]. Compatibility of lubricants with gelled liquids depends on the compatibility of the liquid constituent.

Lubricity [*1–2*]. No experimental data are available. It is reasonable to consider that the gelled liquid will have lubricating properties similar to those of the liquid constituent. These properties are the result of solvent characteristics and reducing properties of the liquid constituent under a frictional force.

Viscosity [*2*]. Viscosity varies with shear force (i.e., velocity, tube diameter). This presents a definite problem of retention in pockets and cavities of components, resulting in poor propellant utilization and difficulties in cleaning.

Radiation tolerance [2]. While no major problems have been envisioned for the liquid constituents under consideration for gels, some gelling agents appear to break down under various types of radiation and, in turn, cause the gel to be degraded.

Effects of leakage [2]. The vapor pressure of the parent propellant is unaffected by gelling. Toxic hazards of the liquid constituent must be considered. The rate of vapor evolution, however, is reduced, which in turn reduces fire and explosion possibilities. Leakage, predominantly from volatilization of the high vapor pressure liquid constituent of the gel, will leave a residual powder, the gelling agent, creating problems such as sticking or malfunction of valves.

Control of flow [2]. Difficulties in accurately measuring flow rate are anticipated. It may be expected that pressure drop will generally be higher than for the parent propellant and design allowances may be necessary.

Shutoff [1]. A very serious problem exists with the last valve in a system exposed to space environment. Rapid volatilization of the liquid component of a gelled liquid would leave a solid residue, possibly of sufficient quantity to produce leakage or sticking of the valve on subsequent operations.

METALLIZED GELLED PROPELLANTS

Physical Properties

To increase the thermochemical performance of rocket propellants, or to increase propellant bulk density for volume-limited applications, powdered metals have been added to liquid propellants. The solid phase of these mixtures can vary from 10 to 90 percent by weight. The metallic gel is generally prepared with metal having an average particle size range of 5 to 50 microns so that: (1) its effect on the flow characteristics of the propellant will be minimized, (2) burning in the combustion chamber will be less difficult, (3) suspension in the liquid will be easier. A gelled mixture is necessary to suspend the metal powder in the propellant. Like other gelled propellants, these gelled propellants often exhibit thixotropic properties.

Because of the solid particles, metallized gelled propellants present problems not encountered with standard propellants. For example, abrasive action of the particles could cause galling of the valve stems or scarring of valve seats and plugs. Particles may become trapped between a valve seat and plug, allowing the valve to leak. Evaporation of the liquid phase of the propellant will leave a solid matrix as a residue that can hinder valve operation and restrict flow. The presence of solid particles in the propellant can result in galvanic corrosion of certain standard materials of construction. Any intimate contact of the propellant with lubricants could result in the transfer of particles to the lubricant, thereby reducing the effectiveness of the lubricant. In addition, the problems listed previously for nonmetallized gels arise. Although it is not certain whether special components will be necessary to utilize metallized gels in propulsion systems, modified designs of standard components will be required to reduce the effects of the undesirable characteristics.

Metallized Gelled Propellant Problem Areas

Metals and ceramics [2]. Metallized gels subject metal components to erosion and/or galling under dynamic conditions. Suitable metals for construction require compatibility with both the liquid and metal constituents of the gel.

Organic polymers [2]. Erosion and galling of plastic materials are anticipated under dynamic conditions. Generally, polymeric materials compatible with the liquid constituent are expected to serve as well under static conditions.

Wet and dry lubes [1–2]. Gels containing metal particles have considerably greater abrasive action under dynamic flow resulting in greater "washout" of lubricants. For static conditions, the restrictions placed upon the liquid constituent dictate selection of a compatible lubricant.

Lubricity [1]. The incorporation of metal particles in a gelled propellant tends to decrease the lubricating properties due to the abrasive nature of such metals.

Viscosity [2]. As with gelled liquid propellants, viscosity varies with sheer force and presents similar problems of propellant utilization, cleaning, and greater pressure drop.

Radiation tolerance [2]. Some gelling agents appear to deteriorate under radiation and in turn cause the gel to be degraded.

Effects of leakage [1]. Loss of liquid component from a gel by leakage will leave a solid matrix to interfere with proper valve operation. The volatile liquid will present toxic, fire, and explosion hazards, but to a lesser extent due to the probable slower rate of evolution.

Control of flow [1]. No satisfactory way has been found to measure flow rate accurately.

Soft seats [1–2]. It is expected that the metal particles of gels will have detrimental effects on soft seats. Metal particles, trapped against a valve seat when the valve is closed, can cause the valve to leak initially or after several cycles. Selection of seat material should be based principally on compatibility with the liquid constituent of the gel.

Hard seats [2]. Metal particles may prevent valves utilizing hard seats from sealing, and under dynamic flow the metal particles may erode the seat by abrasive action.

Shutoff [1]. Metallized gels present a serious problem for valves exposed to vacuum conditions because of the greater quantity of residue on volatilization of the liquid constituent.

CORROSIVE CHEMICALS

In the development of new chemical processes, questions invariably arise about materials for certain equipment. Corrosion information is widely scattered through the technical literature, and these questions are frequently hard to answer. Although materials free from corrosion are desired for such an item as a valve seat, this is not always possible. The most economical procedure overall is to provide for a small loss of metal and provide maintenance. Some of the references cited in this section may be helpful in the search for compatible fluid-material combinations.

The *Corrosion Data Survey* compiled by G. A. Nelson (ref. 3) summarizes data from many sources. Feasible materials may be quickly determined and unsuitable ones rapidly eliminated by use of charts, but they are only a guide, and in some instances corrosion tests and pilot-plant experience may be necessary. However, the charts have been checked against actual plant conditions and a good correlation has been found. An arbitrary set of corrosion rates was established. Data for more than 1500 fluids from abietic acid to zirconium nitrate are presented. The materials of construction listed include: steel, cast iron, Ni Resist, 12 CR steel, 17 CR steel, 18–8 stainless, 31b stainless, Worthite and Durimet 20, silicon irons, copper, tin bronze, aluminum bronze, red and yellow brass, silicon bronze, Monel, nickel, Inconel, Hastelloy, aluminum, lead, titanium, zirconium, gold, platinum, tantalum, silver, glass and stoneware, rubber, asbestos, plastics, impervious graphite, concrete, wood, and special alloys such as 25 Cr-12 Ni and molybdenum. Because of the large quantity of information in this reference, no attempt is made to reproduce it here. The interested reader may consult this work as his requirements dictate.

Guides for choosing materials to be compatible with various fluids are available in many cases from equipment manufacturers; Black, Sivalls & Bryson, for example, have published a corrosion guide (ref. 4) which indicates the results to be expected from using particular materials in combination with 376 different corrosive media. The materials include the following metals: cast iron, cast steel, 316 and 304 stainless, 440C stainless, Hastelloy C, Monel, nickel, Durimet, cast bronze, and 17–4PH. Also included are the following nonmetals: Teflon, Buna-N, Kel-F, Teflon asbestos, graphitized asbestos, and asbestos.

In addition to the various treatises, books, and manufacturers' guides, trade publications have featured corrosion articles that provide useful design information. *Materials in Design Engineering* has devoted an entire issue to corrosion (ref. 5). It includes a guide to the corrosion resistance of more than 90 materials in almost 70 of the most common atmospheres, waters, acids, solvents, and other chemicals. Five groups of materials are covered: metals and alloys; plastics; elastomers; other nonmetallics, such as glass and various forms of graphite and stoneware; and coatings and liners. Almost all the information is based on data obtained from the 50 or more major materials producers. A later issue of this same publication (ref. 6) summarizes the behavior of about 90 engineering materials in approximately 70 of the most common corrosive environments.

The importance of a compatible choice of materials for use with a specified fluid and environment is well illustrated by an article in *Materials Protection* (ref. 7), a periodical dealing with corrosion problems. Fourteen case histories of valve-corrosion problems are presented, and the cause of failure and the method of solution are explained. These examples include gate, plug, ball, and safety relief valves. The fluids involve steam, dry chlorine gas, acetic acid, sulfuric acid, hydrogen fluoride, chlorides, and river water. The metals discussed include stainless steels, carbon steel, cast iron, chrome-plated carbon steel, titanium, and nickel-molybdenum. While no particular design guides are given, the case histories are a strong argument for exercising caution in the selection of valve materials. In one case, a titanium valve was inadvertently installed in a hot acid line for which a stainless-steel valve was intended. The titanium valve lasted only minutes.

Often the failure of a minor and inexpensive valve part can be expensive and time consuming. Valve packings may cost little compared to other parts of a system, but they frequently represent a thin spot in the fabric of construction, and are vulnerable to embarrassing failures which require time-consuming repairs (ref. 8). Packing for polymer valves operating at 600° F is an example. The valves were equipped with 12 percent chromium-steel stems with Teflon-impregnated asbestos braid packing. Pitting of the valve stems occurred at the packing location, which was concluded to be due to minute amounts of fluorine compounds emitted by the Teflon. These compounds were trapped during long periods (up to 2 months), which probably caused the pitting corrosion. Reasonably frequent stem motion or stainless stems would prevent such a problem. In this particular case the problem was solved by repacking with a braided AAA-grade asbestos, surface finished with mica.

A method of providing a hard, wear- and corrosion-resistant valve seating surface on a cast aluminum alloy valve body was recently developed by North American Aviation under contract to the Marshall Space Flight Center (ref. 9). The fine finish required on the valve seat is susceptible to handling damage as well as to damage from foreign particles during operation. Corrosion products form on the critical seating surface of the casting after extended exposure to high humidity at elevated temperatures. These corrosion products interfere with proper seating of the valve element and cause leakage. A hard anodic coating provides a smooth, wear-resistant surface. Vacuum impregnation with a thermosetting monomer, diallyl phthalate, seals the pores on the anodic coating to prevent galvanic corrosion. The seat area is machined and lapped to the proper configuration and finish. A hard anodized coating approximately 0.002 inch thick is applied to the seat area itself as well as to surfaces immediately adjacent to the seat. To apply the impregnating monomer, the part is first exposed to a maximum absolute pressure of 1 millimeter of mercury. This procedure effectively removes any moisture and gases entrapped in the pores of the anodized coating. The impregnating solution is then forced into the pores under a pressure of 100 psi at approximately 200° F, after which the surface is washed lightly in a suitable solvent to remove excess material. A final curing process at approximately 285° F sets the resin. A final lapping process to remove the slight film of resin is usually required for good seat-leakage characteristics. Removal of this thin film does not impair corrosion resistance, since the pores remain impregnated.

Hard anodic coatings have been used for valves and other equipment for several years. Michigan-Dynamics, Inc., uses this technique (ref. 10).

A study of compatibility between O-rings and alkali metals has been made by the Jet Propulsion Laboratory (ref. 11). A series of 11 O-ring materials was exposed to static potassium at 300° F for 500 hours. Buna-N was the only material which retained its initial shiny, smooth and elastic characteristics. Six other O-rings remained in fairly good condition: polyacrylic, butyl, silicone, fluorosilicone, natural rubber, and nitrile MIL–P5315A. Neoprene was attacked and acrylate was completely destroyed. Teflon and Buna-S were hardened and roughened, respectively. An identical series was exposed to solid lithium at the same temperature for the same time. Buna-N, acrylic, Buna-S, neoprene, and nitrile MIL–P5315A were hardened and attacked. Silicone was completely destroyed and butyl was softened to a plastic unusable state. Polyacrylate, natural rubber, Teflon, and fluorosilicone appeared to be in satisfactory condition. Another series of O-rings is currently in test at 450° F in molten lithium. Table XIX summarizes this information.

A bibliography of O-rings (ref. 12) covering the period from March 1953 to February 1963 contains numerous references. Various operating temperatures, pressures, and fluids are reported, as well as dynamic and static operating conditions. Many of these citations may be of interest to a valve designer. Also, Battelle Memorial Institute (ref. 13) performed a test to determine space environmental effects on materials and components. Included in the materials were elastomerics and plastics for seals, O-rings, and gaskets.

A survey of components, including valves, for gas-cooled nuclear reactors was prepared by the Franklin Institute for the Atomic Energy Commission (ref. 14). This report, based on a literature survey and manufacturer interviews, was primarily written to assess the state of the art in design and the availability of components for gas-cooled nuclear reactors. A considerable portion of this work is devoted to valves: the basic designs available and their characteristics, leakage and pressure and temperature capabilities, manufacturers, and typical problems encountered in reactor applications. Galling and self-welding of mating surfaces are cited as problems in high-temperature helium-cooled reactors. Studies of galling characteristics of materials in a 100° F helium atmosphere by the Crane Co.

TABLE XIX.—*Summary of O-Ring-Alkali Metal Compatibility*

Number	Compound	Condition after potassium exposure [a]	Condition after lithium exposure [a]
1	Buna-N	Very good	Hardened
2	Acrylate	Destroyed	Hardened
3	Buna-S	Surface roughness	Hardened
4	Polyacrylic	Good	Good
5	Neoprene	Attacked	Hardened
6	Natural rubber	Good	Good
7	Silicone	Good	Destroyed
8	Teflon	Hardened	Good
9	Butyl	Good	Softened
10	Fluorosilicone	Good	Good
11	Nitrile MIL P5315A	Good	Hardened

[a] Exposures were at 300° F for 500 hours. The potassium was molten and the lithium was solid at this temperature.

are reported and referenced, but no results are reported. Self-welding in a high-temperature helium atmosphere was explored, and it was found by test to occur in many cases. A reference is given, but no direct report of materials compatibility is given.

Compatibility problems such as corrosion remain and are unlikely to be entirely eliminated. Evidence of this is given by a recent survey (ref. 15) of the top 10 military corrosion problems. An effective nondestructive inspection program to detect concealed corrosion inside valves is needed. Without reliable inspection, a critical component may fail catastrophically after equipment has been in service or storage for several years.

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CHAPTER 5

Contamination

In both industrial and aerospace systems, contamination by foreign and particulate matter in valves causes erratic system operation, frequently leading to complete shutdowns. Contamination can cause valve leaks, plugging of small orifices, sticking of sliding surfaces, high friction forces, scoring, and wear. Figure 18 shows damage to a valve spool caused by contaminants. Note the blunted metering edge and the horizontal and diagonal scores on the used valve spool on the right.

NASA activities illustrate the importance of contamination. Ames Research Center has experienced drain-valve problems in a helium recovery system that processes up to 1 million standard cubic feet of helium per day. In addition to the helium, these drain valves must pass oil, water, and other foreign matter that has contaminated the system. Marshall Space Flight Center has been troubled by contamination in the Saturn

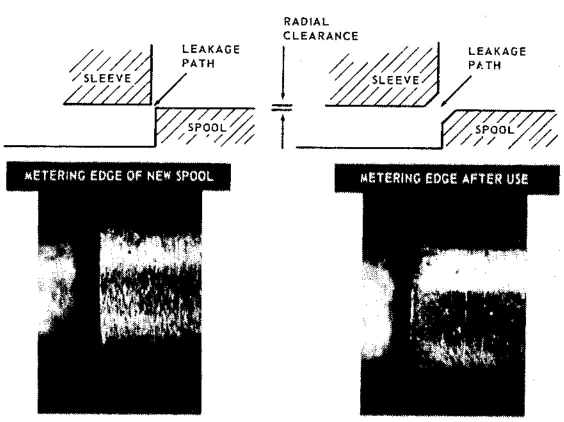

FIGURE 18.—Damage to valve spool resulting from contaminants. (Courtesy Aircraft Porous Media, Inc., subsidiary of Pall Corp.)

program. It supported a "Hydraulic Servo Valve Reliability Improvement Study" by Cadillac Gage Co. (contract no. NAS8–5485) in which servo valves in electrohydraulic circuits were singled out as one of the components most vulnerable to contamination. The Center also has found it difficult to clean components to the levels required in such gas supply systems as the ST–90 and ST–124 Air Bearing Gyro System. Maintaining the cleanliness level of components during assembly and installation is important for satisfactory operation of the system. Use of hoods, shields, and possibly "portable cleanrooms" during assembly and installation operations is planned.

Valve contamination became such a critical problem that a valve clinic was built at the Center to provide ultraclean working conditions. All valves requiring cleaning, disassembly, or repair for the Saturn program go through this valve clinic before final installation in a system. Logbooks are kept for every valve going into the Saturn piping systems. Numerous studies are underway to improve the packaging materials for handling and storing valves after assembly and before installation. The containers for transporting valves must also be ultraclean, moistureproof, and designed to keep out foreign matter. Routine practice is to use Teflon closures and caps on the valves before wrapping them individually in Aclar[1] sheet material within the ultraclean areas.

The critical requirements specified for the cleanroom valve clinic at Marshall Space

[1] Trade name of Allied Chemical, General Chemical Division; a material that is less permeable to gases and generates fewer particles than polyethylene.

Flight Center are shown in table XX. Even tighter specifications are called for on the bench area where the valves are inspected, cleaned, and packaged, as shown in table XXI.

TABLE XX.—*Cleanroom Condition*

Maximum particles/ft^3	Normal operation	At rest
Above 0.5 micron	10 000	3000
Above 1 micron	[a] NA	1000
Above 5 microns	65	[b] 20
Maximum size (microns)	([c])	5

[a] Not available.
[b] Should be random in nature; constant counts in this range will indicate a problem exists either in the air-handling system or in the facility structure.
[c] This will be a function of processes carried on in the room.

TABLE XXI.—*Cleanroom Bench Condition*

Maximum particles/ft^3	Normal operation	At rest
Matthews bench:		
Above 0.5 micron	1000	100
Above 5 microns	10	0
Laminar-flow bench:		
Above 0.5 micron	1000	100
Above 5 microns	20	0

COMMON PARTICLE SIZES

It is hard to conceive the true meaning of the 0.5-, 1-, and 5-micron particle sizes specified as limitations for the valve clinic areas without relating them to the common particle sizes. Figures 19 and 20 show common particle sizes and table XXII shows their concentrations in a cubic foot of typical rural and metropolitan area atmosphere. Typical sources of large particle generation in laboratories, manufacturing shops, and storage areas are listed in table XXIII.

SOURCES OF CONTAMINATION

Harmful dirt and foreign particles may be built-in, introduced, or system generated. Built-in contaminants result from manufacture, handling, and installation of components. As an extreme example, a gas distribution line is sometimes so badly contaminated by sand, wrenches, rust, etc., that satisfactory valve operation becomes impossible. Introduced contaminants can enter the system through such avenues as seals or reservoir breather caps. Lint and other foreign matter can enter when the system is opened, perhaps for repairs, or when the fluid used for filling contains dirt. In some cases, e.g., a hydraulic system in a mine, it is almost impossible to prevent introduction of contaminants. Hydraulic lines in a mine are difficult to protect from mechanical abuse, and often must be repaired at the breakage site in an unclean environment.

TABLE XXII.—*Approximate Particle Size and Concentrations in Typical Rural and Metropolitan Areas*

Particle size, μ	Particles/ft^3	
	City	Rural
0.7– 1	1 000 000–1 500 000	40 000–50 000
1– 3	10 000– 125 000	10 000–15 000
3– 6	35 000– 40 000	4 500– 5 000
6–12	3 000– 4 000	1 000– 1 200
12–25	400– 500	5– 10

TABLE XXIII.—*Typical Sources of Larger Particles Generated in Laboratories, Manufacturing Shops, and Storage Areas*

Source	Size, μ
Crumpling paper	65
Writing with ballpoint pen on ordinary paper	20
Vinyl abraded by a wrench or other object	8
Rubbing or abrading an ordinary painted surface	90
Rubbing an epoxy-painted surface	40
Handling passivated metals	10
Seating screws	30
Sliding metal surfaces (nonlubricated)	75
Belt drive	30
Abrading the skin	4
Soldering (60/40 solder)	3

Generated contamination is created within the system and is caused by wear, chemical action of fluids on materials, and fluid

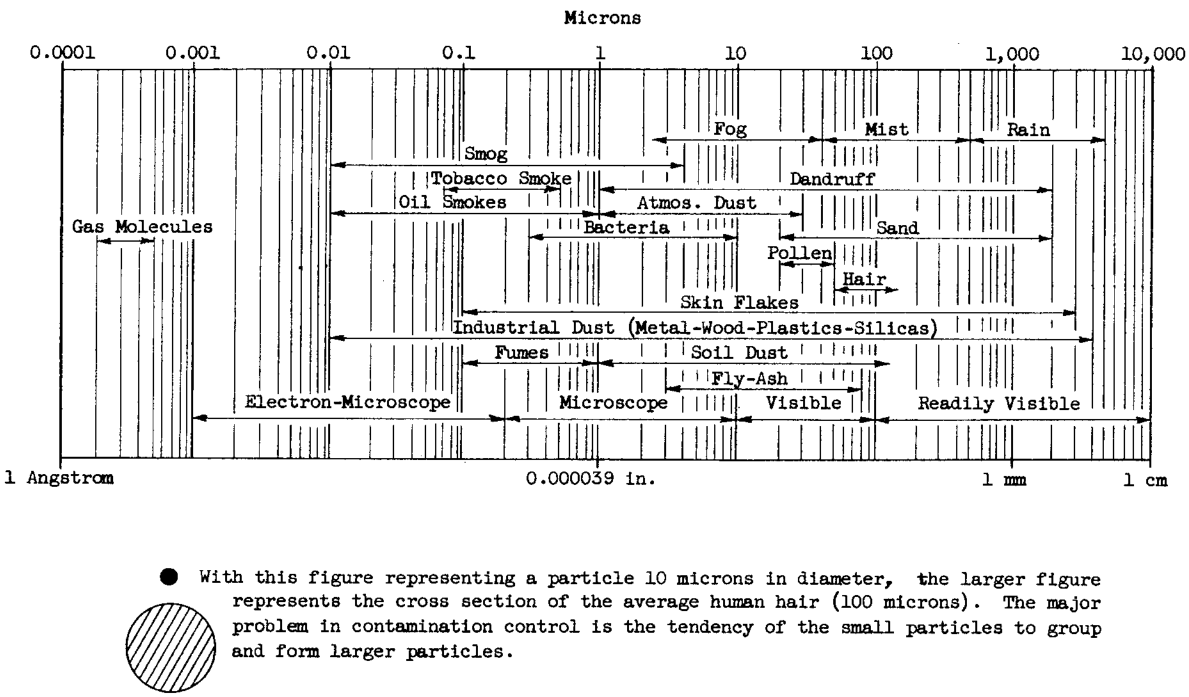

FIGURE 19.—Approximate sizes of common particles.

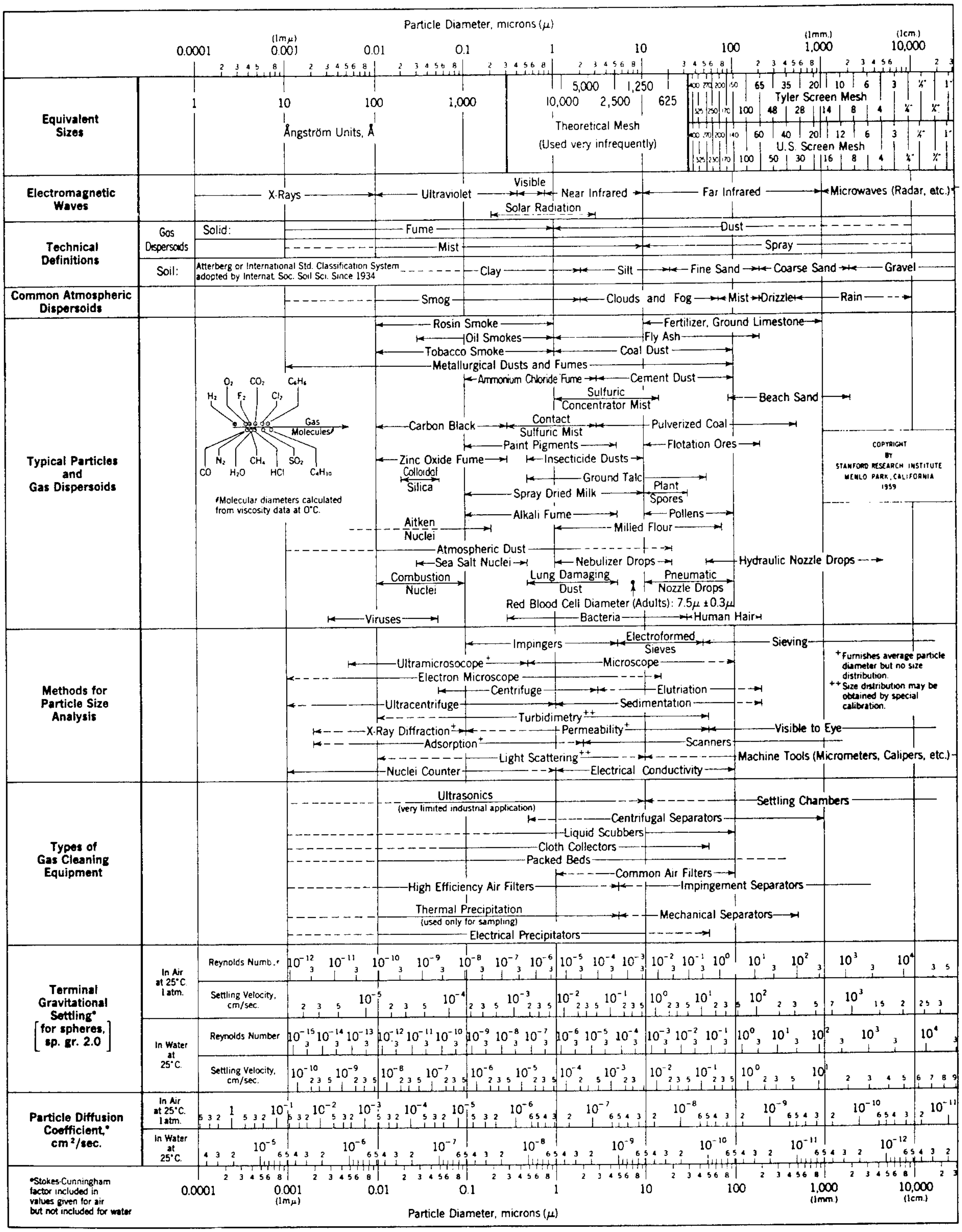

FIGURE 20.—Characteristics of particles and particle dispersoids.

breakdown. Wear contaminants include small particles of metal and sealing materials resulting from the wear of such moving system parts as valve seats. Fluids may react with system materials to form small particles that become contaminants. Fluid breakdown can result in formation of sludge and acids which clog passages and cause pitting and corrosion. Rheopexy, a property exhibited by certain colloidal solutions containing particles that form gels induced by mechanical means, has been observed during high-velocity flow of nitrogen tetroxide (N_2O_4) through small orifices such as are found in filters. Freezing of small amounts of water or other constituents mixed with a system fluid also can cause serious problems.

Since, with care, introduced contaminants can be very nearly eliminated, there are normally two sources of contamination: built-in and system generated. The most troublesome contamination in aerospace systems is system generated.

SURFACE DAMAGE AND WEAR

An analytical and experimental study of detailed aspects of valve seating was performed by Tellier et al. at the Rocketdyne Division of North American Aviation (ref. 1). Their work is an excellent reference and is the source of much of the material presented in this section. It includes extensive citations of pertinent literature. The vast literature on friction and wear attests to the complexity of this subject. Summaries of this literature have been written by recognized authorities (refs. 2–5).

The theory of elasticity has shown that, for direct normal contact, material failure initially occurs below the surface. As tangential forces are applied, the maximum shearing stress moves up toward the surface (ref. 6) and, with sufficient friction, exists at the surface. Interfacial slip with wear results when frictional forces are overcome.

For most valve seats a period of cyclic service usually results in some change in the leakage characteristic. If leakage decreases with cycles, a "run-in or wear-in" process takes place; if leakage increases, a "wearout" occurs. A change in the leakage characteristic certainly denotes a change in the seating compliance (since seat loads are fixed) and, in the absence of changing external forces, a change in the surface profile is not likely because of the probable interplay between surface roughness, macroelastic and plastic strains caused by contaminants, edge contacts, and wear from interfacial shear.

PLASTIC STRAINS

Substrate plastic strains with attendant change of geometrical form are a complex function of both the impulsive load and its distribution. Valve seating members having errors of form (taper, out-of-roundness, parallelism), or which are imperfectly guided, may be subject to such deformations, depending upon the amount of error. Because of the geometrical complexity of most valve-seating errors, a theoretical treatment for stress distribution which might allow an elastic design approach is generally not available. This subject is treated in the Rocketdyne study (ref. 1).

In situations where initial impacts cause plastic deformation, the material is deformed and is strengthened to an equilibrium state. Subsequent cycles under a similar condition will produce predominant elastic deformation. Interfacial compliance will be affected in proportion to the amount and location of plastically displaced material, since this material must henceforth be elastically deformed by the fixed seat load. If the displaced material is above the plane of seating, leakage will increase. Conversely, if material has been pressed into the planed seating, a decrease in leakage will result.

Experience has shown that for functional valves, leakage generally decreases with cycles up to some point, holds constant, and then increases. The reasons involve a combination of (1) plastic strain of the surface asperities and supporting substrate, and (2) a wear process. The amount of leakage change and number of corresponding cycles are related to the surface profile and roughness, seating errors, material properties, and impact energy.

Valve seats undergoing plastic flow with each cycle (a "brute force" approach) generally have a relatively low cyclic capacity because of work hardening of the interface material. With work hardening and a fixed seat load, leakage will increase with the number of cycles, ending in abrupt failure from surface fatigue. This is not to say such valves do not seal well. Indeed, many industrial valves hold very low leakage levels, but at the expense of weight, heavy seat loads, and short life.

Other valves employing liquid metal as interfacial materials have been investigated (ref. 7) and found capable of holding molecular leakage levels. Many severe technological problems must be overcome, however, before reliable cyclic capability is developed.

WEAR

Because of the complexities and the difficulties of measuring the minute dimensions associated with even large leakage changes, we have little factually correlated experimental data on the mechanism of valve failures. The many descriptions and hypotheses of seating mechanisms and subsequent failure modes are largely conjectural. The predominant wear process in valve seating has not been identified.

A review of the literature indicates that, although the laws of friction have been fairly well substantiated, there are no satisfactory laws for wear. As a result, wear considerations in the design of equipment must be based upon direct experimental evidence and guided by the documented test data of many researchers.

Ernest Rabinowicz of MIT (ref. 4) describes four basic types of wear as—

(1) *Adhesive wear.* This is the most common form. Junctions are formed between sliding surfaces, which subsequently shear in either of the two metals or at the interface, depending upon relative strengths.

(2) *Abrasive wear.* Surfaces are worn by the plowing action of hard particles.

(3) *Corrosive wear.* Surface films formed by a corrosive environment are worn away so that corrosion continues.

(4) *Surface fatigue wear.* Repeated high-contact loads induce surface and subsurface cracks, which eventually break up the surface, leaving relatively large pits.

Wear also has been classified in its most common forms by Bowden and Tabor (ref. 2) as mild and severe wear. Below a certain load, interfacial shear occurs, and wear is small (mild wear); above this load, wear rises catastrophically to values that may be many times greater (severe wear). In severe wear the loss of material is mainly due to subsurface shearing of the weaker metal.

When wear is due mainly to the shearing of junctions (i.e., adhesive wear), an empirical relation for the value of wear per unit distance of travel is

$$Z = \frac{KW}{3p}$$

where K represents the percent of friction junctions producing wear, W is the applied load, and p is the hardness of the wearing surface. Tabulated values for K show that the wear rate between two surfaces is reduced by (1) use of hard materials and (2) use of materials with low interaction; i.e., dissimilar materials of low solubility.

The above relationship does not, however, provide any information about the change in surface texture with wear. Rabinowicz (ref. 4) has shown that in some cases wear particle sizes can be predicted on the basis of elastic surface energy and hardness properties. His theory has been applied to sliding metallic seals (ref. 8). While the theory has been correlated for experiments with soft metals, no data are given for the various combinations of harder metals, cermets, and ceramics known to have the best wear characteristics.

Considerable attention has been given to fretting or, more commonly, fretting corrosion. A thorough summary of fretting prepared by Harry Diamond Laboratories (ref. 9) describes the test methods, results, and conclusions from over 200 reports obtained from English sources.

Fretting is caused by small oscillatory tangential motions between two materials. The process normally involves relatively large numbers of cycles, occurs over a wide range of loads, and is usually accompanied by formation of corrosion products. The mechanism of fretting appears to be a wear process involving adhesion that is further accelerated by corrosion and abrasive wear from the corrosion products. However, fretting has been produced in literally all combinations of materials and in various atmospheres, including a vacuum. Therefore, corrosion is not necessarily a part of the process. Fretting corrosion factors (ref. 9) of potential significance to poppet and seat design are listed below:

(1) *Load.* As load on the test part is increased without changing other factors, relative slip gradually decreases. Weight loss (due to wear) correspondingly increases, reaches a maximum, and then decreases to zero when the parts are sufficiently loaded to preclude slip.

(2) *Slip amplitude.* Fretting is generally recognized to occur from oscillating motions between 5×10^{-8} and 10^{-2} inches. In general, the wear rate is proportional to the slip amplitude.

(3) *Slip frequency and duration.* While fretting wear is not greatly dependent upon the frequency of oscillations, it is proportional to the total number of oscillations. Therefore, parts subject to the higher frequencies are expected to suffer the greatest wear (assuming constant slip amplitude).

(4) *Temperature.* Fretting increases with lower temperatures, the worst conditions being reported at −240° F. Fretting has also been produced at −300° F. Between 32° and 300° F the wear rate due to fretting was almost independent of temperature. It appears that fretting does not generally cause high surface temperatures.

(5) *Atmosphere.* Oxidation is an important consideration in determining the amount of damage; hence, the presence of oxygen and water vapor affects the rate of fretting. For ferrous materials oxidation is essentially the controlling factor in the damage. Increases of from 3 to 10 times in wear have been observed for changes in atmosphere from nitrogen to air. However, tests made in a vacuum indicate that damage may increase even though oxidation is suppressed. Tests in a helium atmosphere indicate that damage is caused almost entirely by metal transfer.

(6) *Materials.* In general, softer materials fret more than hard materials. Metals that form hard, abrasive oxides, such as aluminum, chromium, and tin are particularly susceptible. Aluminum and stainless steel form hard oxide debris (Al_2O_3 and Cr_2O_3), oxidize very rapidly and absorb oxygen upon exposure to fresh air. Of all materials, stainless steel is reported to be the most susceptible to fretting, but all materials fret to some degree, even with lubrication.

(7) *Surface roughness.* In general, the finer the surface, the more susceptible it is to fretting wear. No data have been reported on terminal surface roughnesses produced by fretting.

In the case of curved contact surfaces, such as bearings, fretting can result from small tangential loads even though there is no gross slipping (refs. 10–13). Because of the elliptical pressure distribution under a normal load between curved surfaces, a small annular area at the edge of contact exists in which any externally applied tangential force must exceed the frictional restraining force. This area is created by the elastic lateral deformation in the higher loaded central area of the contact that governs the total motion between the contacting parts. Thus, as the contact pressure diminishes to zero toward the contact edge, a point is reached where the frictional force from local normal load is exceeded by the tractive force. Tractive forces between curved surfaces can be externally applied as a straight tangential load or couple.

Another more subtle source of interfacial tractive forces occurs when materials that are together do not deform laterally in like fashion, due to either dissimilar configurations or to different elastic properties (e.g., shear modulus and Poisson's ratio). Goodman has analytically described the second case between normally loaded rough spheres (ref. 14).

Even when like materials were used in cyclic normal loading between spheres, fretting has been observed (ref. 11). However, this fretting was attributed to the likely possibility that the vibration within the cycling apparatus caused small tangential forces to be applied to the contact area.

SEATING

For the most part, the stringent weight and performance requirements for rocket engine control valves have precluded the brute-force approach to seating. A survey of numerous valve design (ref. 15) and other component development and test programs (refs. 16 and 17) has shown the need for hard poppets and seats with light seat loads and close guidance of moving members. The consequence is a requirement for finer surfaces and narrower seat lands with an attendant increased sensitivity to contamination.

EXPERIMENTAL OBSERVATIONS

Cycle testing (ref. 1) has shown that seating degradation is closely related to the peak load and the interfacial motion during loading. With motions restricted to elastic differential deformation resulting from different poppet and seat geometries or elastic moduli, the basic wear mode was one of fretting. Wear particle size was probably much less than 1 microinch for the materials tested (440C, tungsten carbide). Attendant wear roughness was of a pitted nature, having a peak-to-valley dimension up to about 4 microinches. Leakage change was commensurated with the peak-to-valley roughness change. With the rougher models a decrease in leakage due to wear reduction of the peak-to-valley roughness dimension was observed as expected. The less-than-1-microinch AA models also showed decreased leakage because of fixed orientation of poppet and seat; however, reorientation testing revealed increased leakage.

Fretting was significantly reduced by flat poppet and seat lands of similar dimension and geometry. Elastic deformation of conical and spherical configurations caused interfacial slippage, and thus greater fretting wear than the flat configuration did. The limited scope of testing precluded any numerical correlations which might allow wear predictions under other conditions. Therefore, it should not be construed that conical or spherical configurations are unsuitable for any given application. Their suitability depends upon loading, concentricity, impact velocity, interface spring rates, materials, and interface geometry.

The significance of these cycle-test results lies in the positional and load controls by which they were obtained. Few valves are constructed so that repeated interface contacts occur within the estimated 10-microinch variation maintained by the cycle tester. Nor do most valves impact to produce the forces obtained in these tests. It is therefore reasonably certain that usual valve-seat degradation or wear is caused by gross interfacial sliding and contamination. Naturally, any plastic deformation due to edge contacts would be likely to further seat degradation.

Because of the inherent complexity of valve positional and impact relationships and many allied variables, there are no data on actual valves which might indicate a numerical basis from which to evaluate the cyclic seating phenomena. The results presented here provide a datum from which performance comparisons may be made. They also indicate the substantial benefits derived from "fixed relationship seating" (ref. 1) as might be obtained by a flexure-mounted poppet.

Contamination effects were studied on cycle-test models. There was considerable evidence of particle-caused deformation of greater than 50-microinch depth. However, the stress-leakage results were basically unaffected. As post test inspection revealed, heavy impacts appeared to disintegrate most particles, leaving only slight particle traces and the resultant depression in both surfaces. It is possible that the combination of high impact forces and rigid positional control greatly reduced the potential deteriorating effects of contamination. A series of tests was also run for lead and diamond contaminant particles of about 0.004-inch diameter in order to determine the characteristics of hard and soft particle envelopment. Potential particle envelopment is indicated at an apparent seat stress of about 15 000 psi for a 440C steel seat and poppet.

FILTERS

Filters, which are normally used to protect the mechanical function of a valve from contamination, are usually found in the line and sometimes in the valve itself. They cannot remove wear particles created by bearings, actuators, and the valve seats themselves if these sources lie between the filter and the part to be protected. The filter is often selected to remove all particles of size greater than or equal to one-half the smallest fit or tolerance between moving parts of system components. However, the filters should also remove particles of the size generated by the system, particularly for recirculating systems. For this reason, it is possible to have a cumulative buildup of small-size contaminants with resultant valve leakage, even though the filter performs as specified. The filter industry has not yet solved the problem of producing a clean filter.

Three separate, extensive test programs have recently been conducted on filters of various types. In one program conducted by Space Technology Laboratories for the Air Force at a commercial laboratory, filters of 16 separate manufacturers were tested and evaluated. The filters ranged in size from small airborne hydraulic filters to 12-inch line filters used in propellant loading systems. In general, the filters were examined for initial cleanness, bubble point, cleanness after vibration, differential pressure and rated flow, filtration efficiency and dirt-holding capacity. A similar but completely separate test was conducted by the Army Corps of Engineers, and a third test was conducted by the Materials Section of the Structures and Mechanics Laboratory, Marshall Space Flight Center. A remarkable agreement among these three separate test programs was obtained on one point in particular: the lack of initial cleanness of the filters in the as-received condition. As a result of a general disregard among the manufacturers for even elementary care with respect to cleanness during the manufacturing process, almost all the filters contained built-in dirt or contamination particles that were greatly in excess of the absolute rating of the filter. While the reports of these recently issued tests have been very critical of the filtration industry's product and may be expected to bring reforms, corrective action will take time. It is necessary, therefore, to be aware that a filter may be a contaminant generator and thus defeat the very purpose for which it was installed. A recently approved SAE document, ARP 599, defines the various procedures in cleaning and inspecting filters.

DISCUSSION

The *Aerospace Fluid Component Designers' Handbook* (ref. 18) presents a comprehensive summary of contamination, its nature, externally viewed effects (sticking poppets, leakage, etc.), cleaning and control methods, and considerations in design. Review of this and other literature will show that not enough has been accomplished in quantitatively defining the source or volume of contaminants, or their effect on valve seating. Consequently, filtration and cleaning requirements are based upon generalization of contamination sensitivity and the availability of commercial products rather than upon detailed knowledge of the real problem. This practice is manifest in numerous failure reports which cite contamination as the cause of malfunction.

The present state of knowledge on friction and wear is sufficient to justify prediction that the friction force can be reduced for materials in sliding contact in a vacuum by proper selection and combination of materials. The optimum would be a hard material sliding on a softer material plated to a harder material. However, the softer material will produce larger wear particles that may influence the leakage.

Normally, a valve seat or closure is protected by a filter. The size of the wear particles generated by the operating action of the valve may dictate the micron size of the filter to be used. For example, if the wear particles generated by the closure were equal to or greater than 10 microns, there would be no justification for using a 5-micron filter.

Knowledge of the effects that various planetary atmospheres have on wear particle size may be essential to aerospace component design. The effect that atmospheric composition has on wear particle size was investigated by John N. Elliott. His experimental findings support the theory of E. Rabinowicz that the average wear particle size should vary inversely as the reactivity of the atmosphere. For the atmosphere of the planet Venus, which is estimated to be 90 to 95 percent nitrogen, the mean particle size would be important for many mechanical functions; for instance, in close-fitted moving parts such as a shaft sliding in a bushing, the clearance should be larger than the expected wear particle size. For close-fitted parts, seizure may take place in the Venus atmosphere or in the space vacuum, although they work perfectly in the Earth's atmosphere. A leak rate for a given valve in an Earth atmosphere may increase for the same valve operating in the atmosphere of Venus or Mars because of the larger particle size produced. Similar effects may also be observed in industrial processes which use nitrogen or other inert gases.

The surface finish on valve seats is important for leakage control. The initial surface finish should be comparable to, or better than, the characteristic wear particle size for the softer material.

Valve design recommendations for improving contamination sensitivity (ref. 19) are listed as follows:

(1) Valves should be designed to operate with a reasonable amount of contaminants, since they cannot be entirely eliminated.

(2) Flexures (unplated) should be used in lieu of close-fitted sliding parts.

(3) Welded joints should be used rather than threaded connectors, if possible.

(4) Vibration and shock loads around a filter should be prevented.

(5) Tubing length, tees, bends, and fittings that trap particles should be minimized.

(6) Hard, smooth surfaces which produce the smallest wear particles should be used.

As we have seen, contamination is one of the major causes of valve malfunctions. Filters used to trap contaminants do not remove all particles and cannot protect components from self-generated particles. Detailed analytical and experimental investigations of valve seating shed light on the seating process. These studies may aid in the design of contamination tolerant valves.

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CHAPTER 6

Valve Lubrication

Valve lubricants may be liquid or solid. The former have been used for many years, while the second classification is relatively new, dating from the middle 1940's.

LIQUID LUBRICANTS

A liquid lubricant is used in a valve to accomplish purposes other than reduction of friction and increase of wear life. One of its primary functions is to act as a sealant. Hence, liquid lubricants for valves are sometimes referred to as lubricant-sealants. They permit leaktight shutoff even though the actual sealing surfaces may be scored or otherwise degraded. Another function, in the case of plug valves, is to lift the plug slightly from the valve body to prevent locking. Liquid lubricants also provide corrosion protection to the sealing surfaces.

The plug valve is the principal type of liquid-lubricated valve. Some gate, ball, and globe valves also use liquid lubricants; but these are few in number, and are usually for special-purpose applications. A reason for this is that liquid-lubricated valves require maintenance. Since gate, ball, and globe valves do not usually need additional sealing, a liquid lubricant is rarely used with valves of these types.

Commercially available liquid lubricants cover the temperature range of $-80°$ to $+500°$ F. This range is not, of course, covered by any one lubricant. The Walworth Co., for example, developed a series of seven lubricant-sealants to do this. Actually, good sealing is the rule only for temperatures in the $-50°$ to $+250°$ F range. Outside this range, good performance is dependent on circumstances of the application.

Liquid lubricants are developed to adhere to metal surfaces as much as possible to inhibit migration. Nevertheless, migration occurs due to extrusion from pressure differentials and to solution in the process fluid. This makes replacement or maintenance necessary. Lubricant migration is influenced, as might be expected, by viscosity. It, in turn, is greatly affected by temperature. High sealant viscosity constitutes a low-temperature limitation. At high temperatures, lubricant-sealant viscosity is so low that the material is quickly lost and frequent replacement is required. Within the temperature range of advertised applicability, no decomposition is encountered. If temperatures exceeding this are encountered, decomposition and caking can occur. It is fortunate that a plug valve is tolerant of caking, since the operation of opening and closing tends to be self-cleaning in nature.

A zinc chromate putty was used by the Rohm & Haas Co. as a viscous lubricant-sealant in a hot gas valve (ref. 1). A simple conical seat was used. The male seating member was made of molybdenum with a groove vented to an internal cavity in which the zinc chromate putty was stored. This sealant was pumped into the groove by nitrogen pressure which also forced the seats into the closed position. Hence, a new sealant supply was pumped to the groove with each valve closure. The rest of the valve was constructed from stainless steel, with asbestos-reinforced phenolic resin insulation for protection against excessive heating and erosion. Sealing tests were conducted on a hot propellant gas of plastisol composition 163 (1 percent Al) at a temperature of

2877° K and a pressure of 567 psia. Good sealing characteristics were observed for the 28 seconds and 25 cycles of closure encompassed by the test. No appreciable valve damage was noted. However, the performance was unsatisfactory in a test that used hot gas from an aluminized plastisol composition propellant (15 percent Al) at 3340° K and 514 psia. The test lasted 28 seconds, but the valve sealed adequately only for 14 seconds. Apparently the zinc chromate putty failed from overheating and hardening at this high temperature.

Six viscous lubricants were used by the Jet Propulsion Laboratory in a program to develop fill valves for its Advanced Liquid Propulsion System (ref. 2). These valves handle nitrogen tetroxide, hydrazine, monomethylhydrazine, and unsymmetrical dimethylhydrazine; compatibility of the lubricant with the fluid was a matter of great importance. The viscous lubricants, their manufacturers, and compatibility comments follow:

(1) Acid Safe Lubricant No. 2031 (Hercules Powder Co., Hercules, Calif.). This lubricant is a compound of silicones, hydrocarbons, and graphite. It should not be used in contact with hydrazine because the silicone and petroleum constituents go into solution. A hard graphite residue is left which cannot be removed by either solvents or ultrasonic cleaning. Neither should it be used submerged in nitrogen tetroxide because the chemical reaction with the hydrocarbons forms shock-sensitive compounds.

(2) DC–11 Silicone Grease (Dow-Corning Corp., Midland, Mich.) and Molykote Z (The Alpha-Molykote Corp., Stamford, Conn.). A grease of 70 percent (by weight) DC–11 silicone grease and 30 percent Molykote Z was considered. It is not suitable for direct contact with either hydrazine or nitrogen tetroxide. Molykote is not compatible with hydrazine. Nitrogen tetroxide dissolves the DC–11, leaving the free Molykote particles (0–62-micron size) which might damage the seat.

(3) Fluorocarbon Lubricant No. 95–1 (Dixon Corp., Bristol, R.I.). Hydrazine was discolored by contact with this grease, indicating a chemical reaction with constituents leached from the grease which was opaque white before contact.

(4) Insoluble Pump and Packing Lubricant No. 1A (Crane Packing Co., Morton Grove, Ill.). This vegetable-based, honey-consistency lubricant went into solution with hydrazine, and therefore is unsuitable for direct contact.

(5) Apiezon L Grease (James G. Biddle Co., Philadelphia, Pa.). This grease was by far the best lubricant for use with hydrazine and nitrogen, of all those tested. It was neither dissolved nor discolored by hydrazine. Because of its high film strength, Apiezon L is an excellent grease for the nitrogen fill valve.

(6) Kel-F No. 90 Grease (Minnesota Mining & Manufacturing Co., St. Paul, Minn.). At present, this grease is considered the best for direct contact with nitrogen tetroxide.

SOLID LUBRICANTS

Dry or solid lubricants can be any nonliquid material which prevents seizing. Soft crystalline materials like graphite, soft metals like silver and gallium, and slippery plastics such as Teflon and nylon are used as lubricants in many instances (ref. 3).

In general, solid lubricants are used under conditions in which a liquid lubricant would freeze or boil away. Solid lubricants are available for vacuum conditions of 10^{-12} torr (a pressure equivalent to an altitude of 800 miles above the Earth). Other solid lubricants can survive at temperatures as high as 1000° F; still others lubricate at temperatures down to $-400°$ F. A variety of radiation- and oxidation-resistant lubricants has also been developed.

Probably the largest single area of solid lubrication is bonded solid lubricants, sometimes referred to as solid film lubricants. Campbell et al. (ref. 4) have recently summarized developments in this area. These lubricants consist of a lubricating solid called the "pigment" and a bonding agent.

The pigment provides the wear reduction and the low friction required for the surfaces to be lubricated. The binder holds the lubricating pigment to the metal surface. Molybdenum disulfide (MoS_2) is the most commonly used pigment, but graphite, lead oxide, zinc chromate, polytetrafluoroethylene, and other materials can be used either alone or in conjunction with MoS_2 for special applications. The binder is generally a phenolic or silicone resin; in special cases, a binder such as sodium silicate or a vitreous enamel is used.

Pretreatments

The resin-bonded solid lubricants are generally applied in thin films to metal surfaces. The metal surfaces are first subjected to a pretreatment process, the nature of which depends on the surface material; pretreatment is very important to insure adhesion, long wear life, and low friction. Pretreatments and application processes for various metal surfaces are listed by Campbell (ref. 4). Figures 21 through 27 present essentially the same information (ref. 5). In these figures the straight line down from the metal specification is generally the best route to achieve a long wear life.

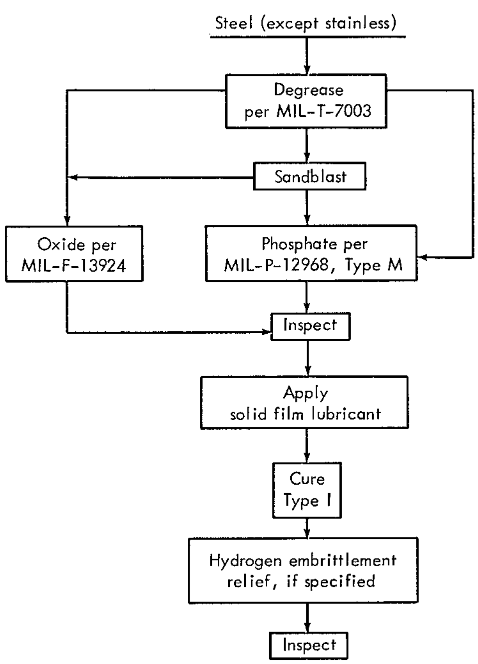

FIGURE 21.—Steel (except stainless) pretreatment and lubricant application processes. (Cure Type 1—400° F for 1 hour or as specified by manufacturer.)

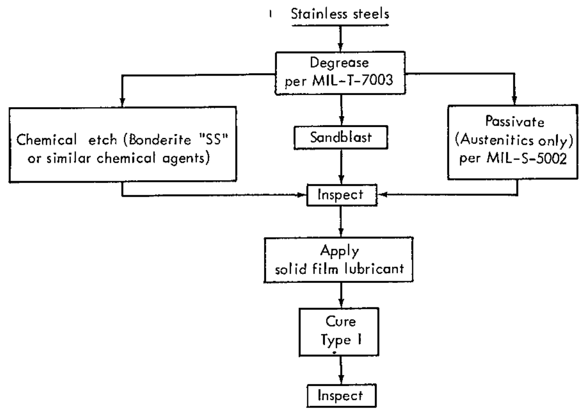

FIGURE 22.—Stainless-steel pretreatment and lubricant application processes. (Cure Type 1—400° F for 1 hour or as specified by manufacturer.)

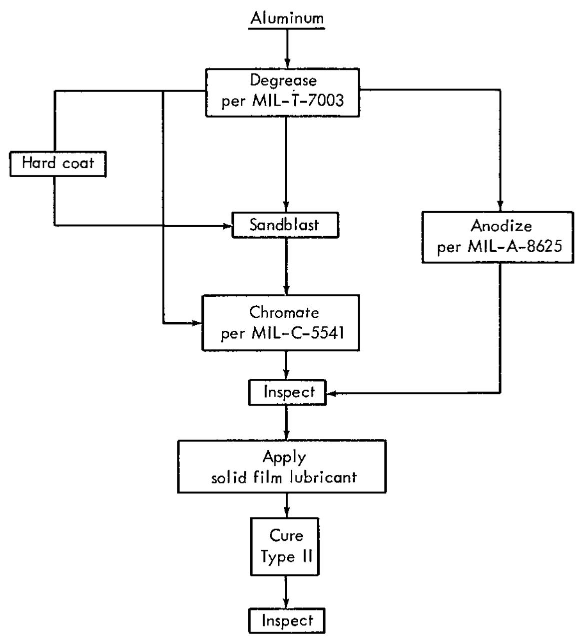

FIGURE 23.—Aluminum pretreatment and lubricant application process. (Cure Type II—325° F for 1½ hours or as specified by manufacturer.)

The degreasing treatments are taken from military specifications; they are generally vapor degreasing in either trichloroethylene or perchlorethylene. Under some well-con-

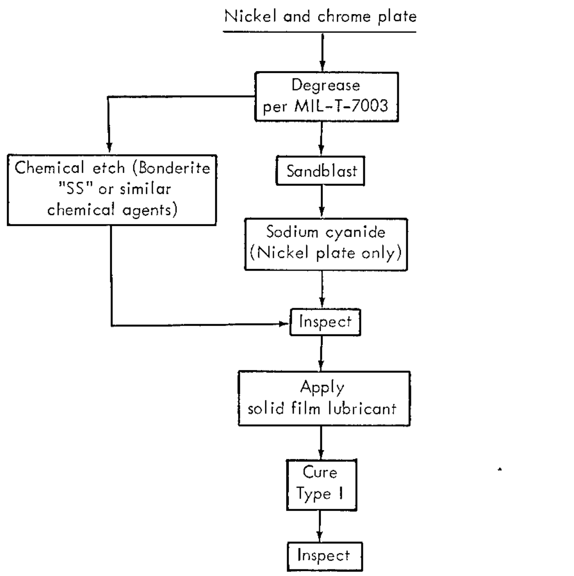

FIGURE 24.—Nickel and chrome plate pretreatment and lubricant application processes. (Cure Type I—400° F for 1 hour or as specified by manufacturer.)

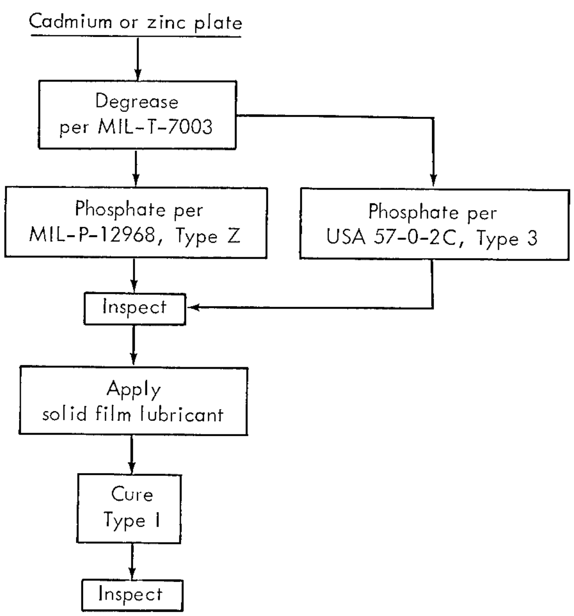

FIGURE 25.—Cadmium or zinc plate pretreatment and lubricant application processes. (Cure Type I—400° F for 1 hour or as specified by manufacturer.)

trolled conditions, caustic degreasing is acceptable.

The most common pretreatments are phosphating, anodizing, and mechanical

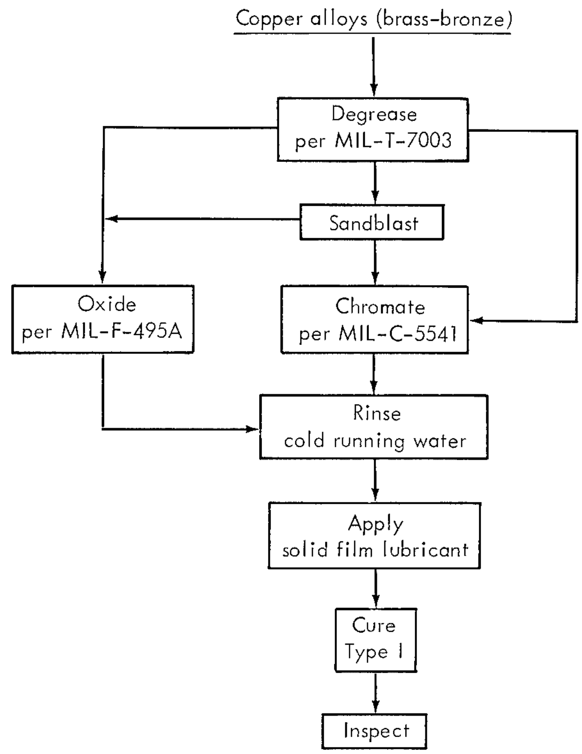

FIGURE 26.—Copper alloys (brass-bronze) pretreatment and lubricant application processes. (Cure Type I—400° F for 1 hour or as specified by manufacturer.)

blasting. These have been widely used, particularly in the aerospace industries where common bearing or wearing surfaces are steel, aluminum, stainless steels, titanium, etc. In the past the phosphate coatings have been used extensively with liquid lubricants to improve bearing life and to gain a small measure of corrosion protection. They are used under solid lubricants for the latter purpose, since the solid lubricants do not provide corrosion protection. Type M phosphating is a surface pretreatment with manganese phosphate. It gives the longest wear life on steel of any of the phosphate pretreatments. Likewise, anodizing is preferred on aluminum parts to obtain the necessary corrosion protection. A sandblasting pretreatment is accomplished by using 120 grit dry silica sand with pressures less than 60 psi; this pretreatment gives a nearly optimum 32- to 63-rms finish. The passivation pretreatment of figure 22 is the normal

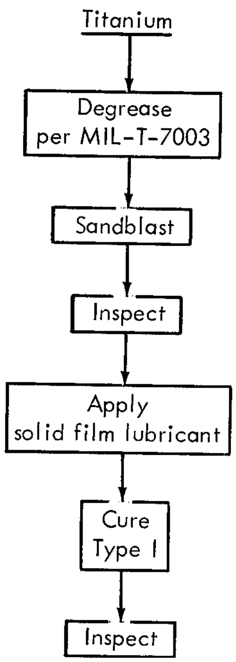

FIGURE 27.—Titanium pretreatment and lubricant application processes. (Cure Type I—400° F for 1 hour or as specified by manufacturer.)

type of passivation used on corrosion-resistant steels.

Lubricant Application

Resin-bonded solid lubricants are generally applied in thin films after pretreating the surface. Application is conventionally accomplished by spraying, dipping, or brushing (refs. 4 and 5).

Solid lubricants are most often applied to valve parts by spraying because of the intricate shape of the parts and the uniform and long wear-life coating desired. Nuts and bolts can be dipped. Spray coatings should be from 0.0002 to 0.0008 inch thick. Too thick a film will flake or peel during sliding, and too thin a film may fail prematurely by rupturing. The optimum film thickness seems to be about 0.0005 inch. Further information on solid lubricant application methods can be found in Campbell's survey (ref. 4).

Solid lubricants can also be applied to nonmetallic surfaces such as ceramics, plastics, and rubber. A solid lubricant manufactured by Electrofilm, Inc., has a Buna-N binder and has been used with rubber O-rings, silicone rubber excepted. A combination of molybdenum disulfide, graphite, and mica is acceptable for elastomer lubrication because of the mica's compatibility with most elastomers.

It is sometimes desirable to apply a solid lubricant to a sealing surface. This can be done by first roughening the surface, then applying the solid lubricant, and, finally, honing the lubricant film to obtain the desired sealing characteristics (ref. 5).

Ceramic balls of aluminum oxide are used by JPL in fill valves (ref. 2). These balls are spherically accurate within 0.000025 inch and have a surface finish of 2-microinch rms or better. Two types of burnished surface treatments for these ceramic balls are used to reduce metal pickup from seat to ball. The first is Teflon burnishing. The second, which is far more effective, is burnishing with Molykote-Microsize (1-micron molybdenum disulfide powder). The ball is Teflon burnished by rubbing the ball on a sheet of paper with a block of Teflon which has a dimple to receive the ball. The ball should roll on the paper and slide in the Teflon socket. To obtain better traction between ball and paper, the paper should be placed on a sheet of rubber. The block and ball should be rotated in a circular path until the paper shows evidence of being polished by the transfer of Teflon from block to paper via the ball. Rotation should be in both directions with moderate pressure at the start and light pressure at the finish. The Molykote-Microsize burnishing process is identical to that for Teflon burnishing, except that a little Molykote-Microsize powder (MoS_2) is sprinkled on the ball path on the paper. Proper application of the coating can be determined visually because the originally milk-white color of the ceramic ball will become a uniform light-amber color.

SOLID LUBRICANTS IN VALVES

Solid lubricant materials are, of course, selected for their resistance to the environmental conditions in which a valve must operate. Generally speaking, they are more compatible with fluids than conventional oil-and-grease-type lubricants are.

Table XXIV presents the results of tests (ref. 5) run on solid-film lubricants. The tests consisted of a 1-week immersion of a film at the indicated temperature, followed by a trichloroethylene rinse to remove residual fluid. The adhesion of the film to the base material was then tested. In most cases there was no adverse effect. Solid-film lubricants have been developed for use in most fluid environments, including such exotic fuels as unsymmetrical dimethylhydrazine and nitrogen tetroxide. Coatings that are compatible with oxygen and have no detonation effect are available and are now used

TABLE XXIV.—*Fluid Compatibility Test on Solid-Film Lubricant*

Material	Temperature, °F	Effect on SFL
ANFA fuel	75	None
ASTM oil No. 3	75	None
Acetic acid	75	None
Acetic anhydride	75	None
Acetone	75	None
Alcohol, ethyl	75	None
Ammonium hydroxide (concentrated)	75	None
Aniline	75	None
Benzene	75	None
Butyl acetate	75	None
Butyl alcohol	75	None
Carbon tetrachloride	75	None
Chloroform	75	None
Cottonseed oil	75	None
Dioxane solvent	75	None
Di-2-Ethyl hexyl sebacate with ½ percent phenothiazine	75	None
Diacetone alcohol	75	None
Ethyl acetate	75	None
Ethylene glycol	75	None
Fluid, anti-icing, Mil–F–5566	75	None
Formaldehyde	75	None
Glycerine	75	None
Grease, Mil–G–3278	160	None
Hydraulic fluid, Mil–H–5606	75 and 160	None
Hydraulic fluid, Mil–H–8446	75	None
Hydrazine	75	None
Hydrocarbon fluid, Mil–S–3136, type II	75	None
Hydrocarbon fluid, Mil–S–3136, type III	160	None
Hydrochloric acid (concentrated)	75	Blistered
Hydrogen peroxide	75	None
Isopropanol	75	None

TABLE XXIV.—Concluded

Material	Temperature, °F	Effect on SFL
Isopropyl ether	75	None
Jet fuel, Mil–J–5624, JP–4	75	None
Lubricating oil, Mil–L–7808	75 and 160	None
Lubricating oil, Mil–L–6082	75	None
Mercury	75	None
Methyl alcohol	75	None
Methyl isobutyl alcohol	75	None
Methyl isobutyl carbinol	75	None
Nitric acid (8 percent)	75	None
Nitric acid (15 percent)	75	Decomposed
Nitric acid (concentrated)	75	Decomposed
Nitrobenzene	75	None
Phosphoric acid (concentrated)	75	Blistered
Pine oil	75	None
Potassium hydroxide (50 percent)	75	None
Skydrol 500	75 and 160	None
Silicone fluid, DC 550	75	None
Silicone fluid, DC 200	160	None
Sodium hydroxide (20 percent)	75	Blistered
Sulfuric acid (10 percent)	75	Blistered
Sulfuric acid (80 percent)	75	Blistered
Toluene	75	None
Trichloroethylene	75	None
Tri-N-Butyl phosphate	75	None
Water	160	None
Water, distilled	75	None
Water, tap	75	None
Water, salt 20 percent	75	None
Xylene	75	None

in welding system valves and regulators at pressures up to 2000 psi.

To date, however, no solid-film lubricant has been developed for the acidic types of fluorine materials. Further, even such seemingly harmless environments as damp air have given trouble on occasion. Water vapor, for example, can cause molybdenum disulfide lubricants to exhibit an increased coefficient of friction and a corrosive effect on the base metals (ref. 4). Solid lubricants are not a panacea. Wear debris (less than 5-micron size) from a molybdenum disulfide coating can contaminate a fluid system and be unacceptable on this basis.

Solid lubricants were used in valves manufactured by the ITT-Hammel Dahl Co. for a high-purity liquid oxygen system; the solid lubricant was sodium-silicate-bonded molybdenum disulfide and the valves were constructed of 316 and 347 stainless steel. A 316 stainless-steel stem and guide bushing from a 4-inch valve were cycled with and without solid lubrication on the guide bushing. The metal surfaces in the first case were free of oil as indicated by a black-light test. Galling occurred after 1200 cycles and 60 200 cycles for the unlubricated and the solid lubricated specimens, respectively. A fiftyfold increase in wear life resulted from the use of the solid lubricant.

A 347 stainless-steel plug with a ground, hard chrome finish was cycled in a 347 stainless-steel guide in the same test series as above. Galling occurred after 600 cycles. With sodium-silicate-bonded molybdenum disulfide on a hard-chrome-plated 347 stainless-steel guide, galling did not occur until after 11 400 cycles. In another test, this solid lubricant on a coated-plug cycling in a ground 347 stainless-steel guide did not gall or show signs of noticeable wear after 85 200 cycles. These examples indicate the increased valve life attainable by use of solid lubricants.

Solid-film lubricants commonly reduce valve-operating torques by 30 percent. In one case, two 2-inch plug valves were both lubricated with a plug-valve grease. However, one plug and body were coated with a phenolic molybdenum disulfide solid lubricant. Both valves were set at an opening and closing torque of 20 foot-pounds and cycled every 3 minutes. After 48 hours, operating torque of the solid lubricated valve decreased to 10 foot-pounds, while the other valve's operating torque increased to 60 foot-pounds and was seizing. The solid lubricated valve operated for 80 000 more cycles and was still in good condition.

An air valve using a solid lubricant exhibited very good wear life. An aluminum bore in this valve received a stainless-steel spool which reciprocated with a ¼-inch stroke at 800 strokes per minute. The aluminum bore was coated with a solid lubricant, and the valve was operated for 6 million cycles. Less than one ten-thousandth of an inch of wear was observed.

Not all lubricant coatings are molybdenum disulfide. The Electric Boat Division of General Dynamics, for instance, has applied fluorocarbon coatings to submarine ball valves since 1959; the techniques involved are reported by Dussia (ref. 6).

Valves and components for vacuum service may use solid lubricants. Their outgassing properties are therefore of interest. A study by Bowen and Hickam (ref. 7) gives outgassing data for dry lubricant materials at temperatures ranging from 160° to 1160° F at a pressure of 10^{-6} millimeters of mercury. Eleven plastic and carbon compositions, 10 powders, and 6 composites of metals plus lubricants were tested. Among them were molybdenum disulfide and tungsten diselinide (WSe_2). Results show that most of the tested materials are more suitable for vacuum applications than is generally believed.

Where temperatures are so high as to preclude the use of liquid lubricants, solid lubricants should be considered. When properly used, valve lubricants significantly improve a valve's operating characteristics, wear life, and range of applicability.

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CHAPTER 7

Reliability

In defense and aerospace systems, the failure of a single part can cause the failure of an entire mission. This has motivated technological advances in valve reliability.

An article by John De S. Coutinho, Reliability Director, Lunar Excursion Module of Apollo, Grumman Aircraft Engineering Corp. (ref. 1), succinctly discusses the reliability problem.

In conjunction with the "big picture" of reliability, one must remain aware of the many "little" reliability problems that have been around for a long time. They refuse to go away. Among them are leakage, wear, fatigue, and the effect of time on operational integrity. Many systems are expected to be inactive for prolonged periods, and yet be immediately operational on call. Valves in cryogenic fuel storage tanks are a problem in this regard.

Although much has been written on some subjects such as fatigue, few practical techniques suitable for use in design are available. What techniques exist are often proprietary and guarded carefully.

McNorton and Teitelbaum (ref. 2) have classified reliability literature as follows:

(1) *General articles* that cry for more reliable equipment, but offer few concrete suggestions. The authors agree that unreliable components are "indeed" a costly and irritating problem which should be eliminated. Approximately 25 percent of the literature falls in this classification.

(2) *Mathematical articles* written by mathematicians for mathematicians. They convey little usable information to the individual with the reliability problem. About 15 percent of the literature falls in this classification.

(3) *Practical articles* written by individuals with reliability problems who prove that their units are reliable without actually obtaining failure figures. Some of these articles are very good. About 35 percent of the literature is of this type.

(4) *Specialized articles* covering specific important topics in reliability such as redundancy techniques, failure distributions, sampling techniques, testing methods, etc. These articles usually are very informative. Unfortunately, only about 25 percent of the literature falls into this classification.

Our discussion, as we hope to show, will be neither "general" nor "mathematical."

VALVE FAILURE ANALYSIS

Determination of component reliability requires collection of data by testing. Failures occurring during a test give valuable information concerning the reliability of the components. If reliability is to be improved, however, the cause and mode of failure must be ascertained.

The Milmanco Corp. of Seattle, Wash., is presently conducting research on design criteria for valves under Air Force contract AF01-(601)53635. A part of its program involves analysis of the failure of the missile valves. Causes for valve failures reported by Milmanco are summarized below:

(1) Insufficient clearance between body and poppet carrier

(2) Eccentric loading of internal components

(3) Material incompatibility

(4) Poor diaphragm materials

(5) Pulsation caused by large bore and small volume of flow

(6) Galling of internal parts (material selection)
(7) O-ring extruded over edge of poppet undercut
(8) Similar hardness of seat surfaces
(9) Elastomer (Buna-N) hardened and deformed because of heat generated by normally energized solenoids
(10) Galled poppet disk and stem
(11) Poor construction
(12) Poor workmanship
(13) Improper assembly
(14) Improper fabrication
(15) Faulty O-ring
(16) Actuator piston binding
(17) Improper installation of seals.

Twenty-one failures due to contamination were also reported. These involved bleed valves, check valves (6 cases), fuel drain valves, hand valves, regulator, regulator and release valves, release valves (hydraulic), safety valves, and solenoid valves (8 cases).

Also reported were 25 failures caused by incorrect maintenance procedures, and 14 miscellaneous failures.

A survey by Marshall Space Flight Center (ref. 3) lists valves among items which most require improvement. Needed areas of improvement are: compatibility, contamination, design-hot gas, inspection, materials, pressure, seal leak, space environment, temperature, and (significantly) reliability. Though not as specific as might be wished, this list shows that contamination is an important contributor to valve failures.

Piston-type servo valves frequently cause trouble because of friction or even jamming of the spool in its liner (ref. 4). Theoretical considerations of the forces on the piston created by leakage flow past the sealing lands show that a straight and parallel spool in a straight and parallel bore is in neutral equilibrium with regard to forces at right angles to its axis. If, however, the sealing lands are slightly tapered, so that the diameter at the high pressure end is slightly less than at the low pressure end, the piston tends to float in the center of the bore. Conversely, if the lands are tapered the opposite way, the spool is in unstable equilibrium and moves sideways until it touches the liner. This latter condition is known as hydraulic lock. A similar state of affairs arises if the piston is parallel but the liner bore is slightly tapered. A valve may be jammed both by the friction of hydraulic lock and by the accumulation of contaminants in the clearance between the piston and liner. Two methods of reducing friction have been experimentally tried: very fine filtration of the oil supply, and tapering of the sealing lands of the servo piston. The first method reduced valve friction by 80 percent, and the second method reduced it by 60 percent. A combination of the two methods reduced friction by more than 90 percent.

Information on various phases of component reliability is available from the U.S. Naval Ordnance Laboratory (NOL) at Corona, Calif. The Missile Evaluation Department monitors and analyzes the data of Navy missile programs for reliability; it does no testing itself. Three data-exchange programs have grown out of this task: CRHS (Component Reliability History Survey), GMDEP (Guided Missile Data Exchange Program), and IDEP (Interservice Data Exchange Program).

CRHS is the oldest program. Certain organizations can request NOL to perform a survey on certain types of components. NOL then finds out all it can about the component, the various manufacturers, and the experience of users. Reports may be a "narrative" type in which NOL makes an evaluation and recommendation, or a "summary" type in which the essential data are summarized and the party to whom the report is sent draws his own conclusions. CRHS treats a component type only when specifically requested, not automatically. This program has been primarily Polaris oriented.

GMDEP provides for reporting of component-reliability data on all Navy guided-missile programs, and making it available to all in a convenient, expeditious manner. IDEP is essentially the same, but covers ballistic missile and space programs for all the services. The mechanics of the two programs differ only in minor details. Both use "summary" type reports, not "narrative," and material on any component type goes into the system automatically when a participating organization generates any information. This information is generally test data: environmental, acceptance, qualification, receiving inspection, requalification, production, field, and flight.

Components and reports at NOL are numerically coded for convenience in the data-exchange systems. The major participating Government organizations and facilities automatically receive summaries of all reports periodically. Other public or private organizations with need to know may request report summaries on specific components.

An outgrowth of the above programs is the Failure Rate Data (FARADA) Program (ref. 5) directed by the Bureau of Naval Weapons (BUWEPS) and administered and implemented by the U.S. Naval Fleet Missile Systems Analysis and Evaluation Group (FMSAEG) at Corona, Calif. It is a Navy, Air Force, Army, and NASA sponsored program to provide parts/components failure rate and failure mode data to prime contractors and major subcontractors engaged in the design, development, and production of hardware for the entire spectrum of military and space equipment and systems. The failure-rate data distributed by the FMSAEG FARADA Information Center is intended to provide an updating of and an expansion to MIL–HDBK–217A, "Reliability Stress and Failure Rate Data for Electronic Equipment," dated December 1, 1965. Also, such data will be used as a basis for reliability prediction as outlined in MIL–STD–756A of May 15, 1963, "Reliability Prediction Procedures."

The purpose of the FARADA handbooks is to provide reliability engineers, design engineers, and maintainability engineers with failure-rate information in a convenient form for use during the design phase. If properly applied, the information will provide a means of numerically assessing the probability of survival (reliability) of a system prior to or simultaneously with the construction of hardware. After sufficient hardware has been manufactured and a designated number of systems have been operated for a specified length of time, the accuracy of survival estimates made on the basis of failure history can be checked. As experience in the use of this method is gained, refinements can be made, and improved design should result. FARADA data are also in demand for use in maintainability and logistics studies where valid field-generated data are specified.

These failure rates are obtained from specific engineering data and test results. The data are compiled from long histories of part/component failures based upon material gathered by the FARADA Information Center.

Failure-mode distributions for valves of various types as given by the FARADA Failure Rate Data Handbook (ref. 6) show that leakage is an important failure mode for these valves.

RELIABILITY REQUIREMENTS

Valve reliability can be considered under two headings: structural (ability to hold the required pressure without failure), and operational (ability to perform the required valving function).

The first category is probably the primary reliability sought by commercial manufacturers. A catastrophic valve failure under high pressure is a traumatic experience for the offended customer and can be dangerous and costly as well. The various manufacturers' codes provide a guide and minimum requirements. Responsibility for this aspect of reliability (perhaps better called safety) rests upon the manufacturer.

Operational reliability for commercial valves is commonly expressed in the form of a guarantee that the users' requirement for a particular number of cycles or hours of duty in a specified environment will be satisfied. In defense and aerospace applications, a mission reliability number is determined. With this number as a base, reliability is further apportioned to subsystems and, eventually, to individual components such as valves. As mission times increase (ref. 7) from the approximate time of 2 hours for fighter aircraft to the 200-hour lunar-probe requirement, component reliability greatly increases to meet the required mission reliability figure. This has led to attempts to design and develop "super" nonelectronic components (ref. 2). A manned Martian flyby mission of about 10 080 hours is almost sure to be plagued by component and subsystem failures. This realization leads to the "Availability Concept." As explained by Carpenter (ref. 7) and Calabro (ref. 8), this concept maximizes effectiveness by achieving a balance between high reliability and good maintainability. In an untitled and somewhat less organized form, it has guided commercial activity for many years.

DESIGN CONSIDERATIONS

Reliability is best designed into a product on the drawing board. This is particularly true when only a few items are to be produced. The testing needed to demonstrate reliability becomes much more expensive if many devices must be built for this purpose, only a few of which are to see ultimate use. Spacecraft applications pose problems of this nature in contrast to commercial applications, where a relatively large number of devices constituting a standard product line can often be developed and proved in a fairly extensive testing program.

Major considerations in achieving reliability in fluid components are adequate specifications, good design, adequate inspection of materials and components, and adequate testing. Some of the important design considerations influencing reliability are given below (ref. 9):

(1) *Select reliability goals appropriate to the system.* When reliability requirements are high, the designer must put greater emphasis on these requirements as he considers other design factors such as weight, cost, ease of fabrication, and testing costs.

(2) *Design for simplicity.* The designer should attempt to minimize the number of parts, and avoid moving parts, delicate mechanisms, and close-clearance sliding fits.

(3) *Design for component assembly and installation.* Common fluid component problems such as overtorquing and reversal of both electrical and fluid connections can be greatly minimized by careful design and clear and distinct markings. One good way to protect a design against human error is to design for one-way assembly and installation. Another way is to so design a part, such as a seal, that it can be installed correctly in more than one way.

(4) *Design for maintenance.* Components should be designed for easy maintenance when it is required. There should be a minimum need for maintenance training and for judgment by maintenance personnel. Procedures should be carefully and properly specified.

(5) *Design for contamination tolerance.* Since a certain amount of fluid contamination is inevitable, a primary consideration should be to design a component which will tolerate a reasonable amount of contamination rather than try to eliminate all contaminants through excessive filtration.

(6) *Design for minimizing contaminant generation.* The effects of surface finishes and seal and packing materials on the contamination level in a system should be carefully considered. Improper surface finishes, combined with shreddable packing in space-system shutoff valves, have resulted in serious contamination problems caused by packing migration.

(7) *Use proven designs.* Items which have been in quantity production are usually more reliable than new items which have been especially developed for a component or system. Novel design approaches should generally be avoided in favor of proven concepts. This is particularly true in the design of modules such as springs, bearings, etc. However, a proven design may not always adequately meet the requirements, and in

such cases new designs are perfectly justified.

(8) *Design for safety.* The components should be so designed that failure will cause a minimum of impairment to system operation and will minimize personnel hazards. Fail-safe features should be employed so that loss of power will not present an unsafe condition. Safety considerations often dictate the use of valve designs that will automatically close in the event of actuator or power failure.

(9) *Design for environmental extremes.* The designer should consider the worst possible chemical and physical effects that could result from the environmental extremes to which the components must be exposed.

(10) *Design modules with liberal stress and load margins.* The designer should supply generous safety factors and performance ratings to extend service life and increase reliability of component parts (modules) such as springs, bellows, housings, bearings, etc. Vital springs, for instance, should be designed to operate at 20 percent of the normal design stress for the material.

(11) *Design components with liberal performance margins.* System reliability can be increased by components which have liberal performance margins. Thus, successful system performance can be attained in spite of failure of a component to meet design requirements completely. The following are some examples of how reliability can be improved by designing components with increased performance margins:

(*a*) Actuator force margins can be provided so that failure of an actuator to develop its design force still gives adequate force to actuate the valve.

(*b*) Components can be designed large enough so that partial opening of a valve or blockage of a flow passage still provides sufficient flow.

(*c*) Regulators and relief valves can be designed to operate with narrower regulation or crack and reseat bands than required by the system, so that failure to meet component design requirements still fulfills the system objective.

Arbitrary tightening of design requirements may result in a component so complex that its inherent reliability is considerably lower than a simpler component which just meets system performance requirements. Performance margins should be liberal only when apparent system reliability gains are not offset by added component design complexity.

(12) *Design for redundancy.* Redundancy is a common means of increasing component and system reliability. A good example of redundancy in fluid component design is the use of primary and secondary seals in both static and dynamic applications so that leakage through the primary seal is stopped by the secondary or backup seal. The secondary seal is unnecessary as long as the primary seal is functioning properly, but if the primary seal fails, the redundant or secondary seal increases the probability of successful operation of the component. Redundancy in design must be used with great care, since it is possible to decrease reliability through improper use of redundant design techniques. The theoretical gain in reliability achieved by redundant design must be carefully weighed against such factors as increased cost, increased weight, and added overall complexity. Increased complexity alone could potentially offset the theoretical gain in reliability achieved by the use of redundant design.

(13) *Review reliability design.* The reliability of the final product can be greatly improved by a systematic design-review program. Questions should be asked regarding:

(*a*) *Bearings.* Are they protected from corrosion and galling due to dirt, moisture, and inefficient lubricants? Are bearings protected from brinelling due to vibration, shock, or soft materials? Are bearings adequately protected against the adverse effects of vacuum exposure?

(*b*) *Filters.* Are integral filters used to protect the sensitive elements of a component from contamination failure? Do filters have sufficient dirt-holding capability?

(*c*) *Mechanical linkages.* Are actuated surfaces and arms protected from overtravel? Have lubrication requirements for linkages been kept to a minimum?

(*d*) *Seals.* Are bolt-torque requirements specified on the assembly drawings? Are locking devices provided? (If lock wire is used, its length should be kept to an absolute minimum.) Can O-rings and seals be installed easily without being cut by sharp edges?

(*e*) *Flow passages.* Are there flow passages small enough to become clogged with contaminants?

(*f*) *Fasteners.* Do all fastened assemblies contain adequate locking devices or possess practical, but effective, torquing requirements? Are all fasteners (nuts, bolts, etc.) easily accessible to maintenance personnel?

(*g*) *Corrosion.* Are there water or liquid traps formed by brackets, etc.? Is the component splashproof, waterproof, iceproof, and salt-sprayproof? Are there dissimilar metal shims, fittings, or miscellaneous hardware in intimate contact? Are lock washers the type that break through protective films? Have all corrosion-prone surfaces been protected?

(*h*) *Maintenance.* Are all lines, devices, etc., designed so that they cannot be used as handles, steps, or seats? Will all routine maintenance points, drains, etc., be accessible after installation? Have parts been designed so that they cannot be assembled incorrectly? Has the number of special tools been kept to a minimum?

(*i*) *Vibration.* Are there cantilevered parts, brackets, arms, or linkages which will vibrate? How close are resonant frequencies to the environmental imposed spectrum? Can damping be added if vibration problems are encountered?

(*j*) *Fluid fittings.* Are the number of fittings in external lines kept to a minimum to reduce the number of leakage points?

(*k*) *Materials.* In the selection of materials, have the following been investigated: weldability, machinability, formability, fluid compatibility, heat-treat distortion, heat-treat contamination, cost, and availability? Have materials, heat treatments, and stress levels been considered in terms of possible stress-corrosion effects? Has the effect of creep been determined? Has material fatigue been determined and provided for? Have the effects of elevated and low-temperature service upon the material been determined? Have the effects of thermal gradients been considered?

(*l*) *Manufacturing.* Are tolerances so stringent that the shop will not be able to fabricate within these tolerances without excessive cost? Are the capabilities of the manufacturing equipment and facilities within the requirements? Have critical dimensions and properties been designated on the drawings for special attention during the manufacturing and inspection process? Has the proper heat treatment been specified on the drawings for each material heat number? Have parts and subassemblies been adequately identified? Has the cleaning method been specified? Have allowable torques been specified on the drawing? Have distortion and buckling as a result of fabrication processes been considered? Are all fillet radii as large as possible? Have steps been taken to eliminate the possibility of burs from machining which could break loose during operation and cut seals or cause clogging of the system?

The reliability requirements for valves in certain systems are so high that alternate systems actuating in a fail-safe way must be provided in case of valve failure. An example of such requirements occurred in Ames Research Center studies of respiratory functions under high acceleration. A special valve was needed for the oxygen supply in the astronaut's breathing supply system. Requirements for the valves were quick-switching operation, extreme reliability up to 20/g's, fail-safe, low back pressure, low leakage, and capability for remote control. Commercially available valves failed to meet all requirements. A valve of special design was developed and tested at the Ames Research Center, then used in flight simulation and actual flight. This valve, shown in figure 28, is unique in that it is driven by a rotary solenoid in one direction and is spring loaded to return in the other. It requires very low holding currents in one position, and none in the other. The spring loading insures the valve's return to a nonactuated position in the event of electrical failure. A rocking-tee design is utilized whereby the rotary motion of the tee extends to dead center on each end of its stroke. Figure 29 shows the five locations in the Ames closed-circuit respiratory system where this valve design is used.

The Battelle Memorial Institute undertook to develop analytical techniques to predict the performance of flight-control components with respect to time so that an inference regarding their reliability could be made without resorting to life testing. The analytical techniques developed are primarily intended for application during the design stage of a device, and may be helpful to valve designers. In brief, the analytical model relates performance characteristics of a device to its input and internal parameters. From this model, partial derivatives of each performance characteristic with respect to internal parameters are computed. A propagation of variance formula utilizing the partial derivatives and parameter variance values is used to compute the variance of each aspect of device performance. A reliability estimate can then be obtained by comparing the expected range of variability with appropriate specification limits. The method was applied, with good results, to an investigation of the static characteristics of an electrohydraulic servo valve for drift-type failures. The authors tested and explained the method by applying it to a 20-penny common wire nail intended to be driven through a 1.7-inch-thick plank and into a wooden beam.

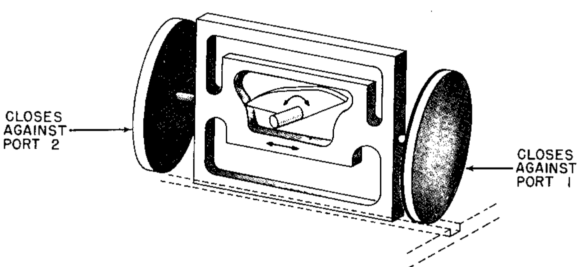

FIGURE 28.—Ames closed-circuit respiratory system valve.

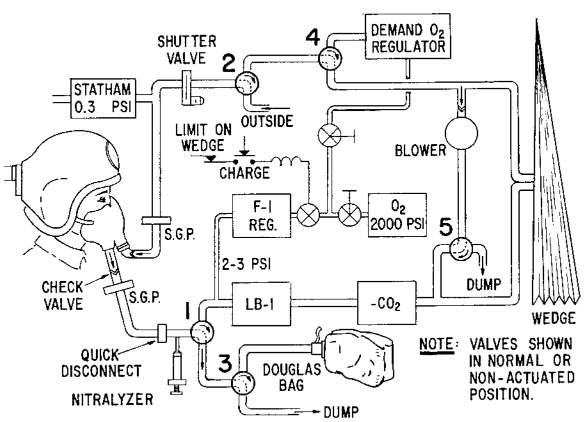

FIGURE 29.—Ames closed-circuit respiratory system.

MANUFACTURING CONSIDERATIONS

One of the manufacturing functions is to produce components for testing and further development. The manufacturing process should be flexible enough to incorporate the design modifications necessary to remedy faults and weaknesses that may be demonstrated by testing.

The manufacturing process is believed (ref. 1) to be largely responsible for unit-to-unit variation in component resistance to failure. Attention must be directed toward identifying the basic physical causes, determining and specifying allowable limits, and insuring that the processes are designed to hold variations within specified tolerances. This requires before-the-fact assurance of adequacy of the manufacturing process. The role of inspection in attaining reliability is limited. If the process produces 50 percent bad parts, it is likely that some bad parts will escape detection and will be delivered with the good. If a process is not capable of producing 100 percent good parts, the emphasis should be on improving the process. Valves produced by different manufacturers to the same design often show unexpectedly large reliability variations. This reflects variations in cleanliness, quality of raw materials, attention to detail, etc.

A concrete example is most helpful and is, unfortunately, difficult to find in documented form. However, a program by the Bendix Corp. (refs. 2 and 10) to improve the reliability of flight control systems is pertinent. The objective of this program was to define the requirements for achieving high-reliability components. The main divisions of the work were:

(1) A survey of representative organizations and the literature to establish the state of the art of reliability engineering;

(2) Selection of a specific component, detailed analysis of its failure modes, and derivation of possible reliability improvement measures to convert the standard version of the component to a "super" version;

(3) Investigation of reliability evaluation test procedures and statistical methods; and

(4) Description of a reliability program generally applicable to component development procedures.

The component selected for reliability improvement was a proportional reaction controller. It is a device to control the attitude of a vehicle in space by metering gas flow through a thrust nozzle, and was a standard production item on the Agena satellite. Following the initial phases of the program as outlined above, experimental verification of the program effectiveness was undertaken. Three "super" controllers and four "standard" controllers were fabricated and tested according to a plan which simulated, as far as was practical, a mission profile. The test results tended to show that the "super" units had advantages over the "standard" units in regard to specific failure modes. However, no statistically significant difference in life characteristics between the two groups was demonstrated. This is attributed to the combination of the small numbers of samples in the test groups, and the relatively low-magnitude improvement in life characteristics actually attained.

CONDITIONS OF USE

In testing, the environment should be duplicated as closely as possible with regard to such important factors as temperature, pressure, acceleration, radiation, corrosion, etc. These same factors should be considered in the design stage as well.

The engineering procedure for developing a product which will not fail in service does not necessarily require a mathematical definition of reliability. It does, however, require knowledge of two items: The operating conditions and the strength of the device. The variables influencing reliability should be clearly understood. For example, on-off cycling of equipment is often a more significant statistic than operating time. As an example, in a check valve used in a rocket engine, the Rocketdyne Division of North American Aviation found excessively rapid wear of a spring-retained flapper. Investigation revealed that an unspecified part of the valve usage consisted of maintaining a slight purging pressure during extended periods of motor inactivity. This caused the check valve to operate as a relief valve (an unintended mode) and the high cycling rate of opening and closure was causing the wear. The environment and application had been incompletely specified. Once this was realized, corrective action could be taken.

A substantial gap between theory, and conventional laboratory fatigue tests, and the fatigue behavior of actuators in service was found by Little and Bacgi (ref. 11). Primarily, this gap occurs because the actuator's service loading is not known. Naturally, some actuator failures resulted from poor design or rating practices. But, lack of knowledge concerning conditions of use, i.e., fluctuating pressures, was the salient feature.

At times, improved reliability can be accomplished with relatively minor modifications of standard components. At Langley Research Center, quick-response plug valves (of the 90°-turn type) were failing in service because of valve-stem binding. Figure 30 illustrates how the valves were modified to include ball bearings to provide free movement of the plug, and yet retain a means of externally adjusting the plug in a vertical direction to control leakage.

A spring was installed backward in a small valve in an aerospace application. This spring had one end squared and the other end plain. The squared end was to push a piston forward; however, when the spring was installed backward, the plain end cocked and wedged the piston. A simple redesign of the spring to square both ends was accomplished so that it would be impossible to assemble the spring backward.

Preventive maintenance is used to increase system reliability by replacing valve parts after a given number of cycles. Shelf life is important since such valve parts as springs, soft seats, seals, and diaphragms can deteriorate through nonuse in as little as several months' time. In some NASA installations, all switches are removed from storage and functionally inspected after 3 months of shelf life. All valves or regulators are removed from storage and functionally inspected after 6 months of shelf life. This inspection indicates the condition of springs, seals, and lubrication, and answers the question whether reservice is required.

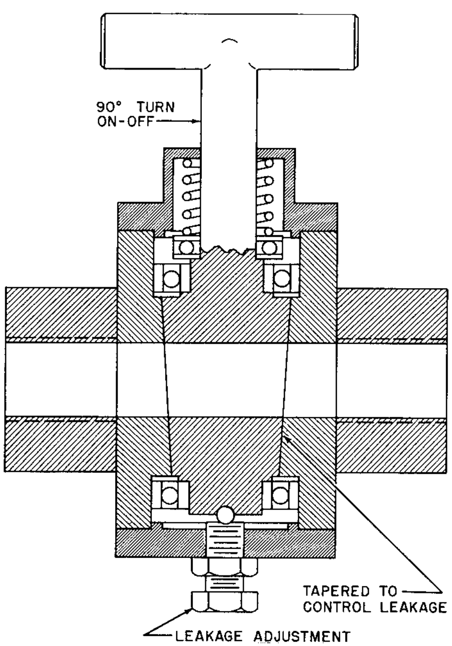

FIGURE 30.—Nonbinding plug valve.

Methods currently advanced for reliability improvement are usually not new or radical. The challenge is in the organizing and planning of effort to identify, control, and eliminate failure mechanisms. Conscientious application of known scientific, engineering, and management techniques, i.e., attention to detail, is most effective in this regard.

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CHAPTER 8

Response Time

A parameter frequently important to valve performance is response time. While there are many requirements for rapid valve response, there are also instances in which a fast response is undesirable.

Some examples in which rapid valve response is necessary follow:

(1) Extraction of gas samples from combustion engines demands that sampling times be quite small; consequently, the valve for this application must have a short response time.

(2) If protection from pressure surges is required, as discussed in chapter 12, rapid valve actuation is a necessity.

(3) Overpressures must be relieved quickly to prevent pressure vessel rupture in many cases, and rupture disks are frequently employed to provide the fast action demanded.

(4) The response of a regulator or a vent valve must be fast enough to compensate for the most rapid change that can occur in the system. Normally, a vent or regulator valve is subjected to relatively slow changes and demands. However, at the instant of application of an inlet pressure surge to a regulator, a high-speed transient pulse may be imposed, especially if the downstream ullage is very small.

(5) Control valves for hydraulic systems must have transient responses which are not so slow that they limit the response of the entire system. Quick-opening characteristics are frequently provided for control valves by sculpturing the valve trim to achieve a large change in flow capacity with a small change in valve lift.

(6) In nuclear reactors, safety may require rapid shutoff for fluid streams when a leak of radioactive fluid occurs. The valves performing this function must, therefore, have short response times.

Typical response requirements are 5 to 50 milliseconds for liquid bipropellant shutoff valves and 12 to 200 milliseconds for gaseous shutoff valves. Valve designs capable of response times ranging from less than 1 to 5 milliseconds are presented in chapter 13.

Fast valve operation may cause pressure surges large enough to be injurious to parts of the system and even to the valve itself. This is particularly the case where a high-velocity liquid stream is to be shut off. Accordingly, valve response time sometimes must not be too short.

DESIGN AND MANUFACTURING CONSIDERATIONS

It is not within the scope of this work to undertake a detailed exposition of designing for fast valve response. Only some of the factors to be considered are discussed.

Assuming that galling, binding, surface finish, pitting, contamination, and other such valve problems do not exist, then the predominant factors affecting valve response time are (refs. 1 and 2):

(1) Mass of moving parts
(2) Pressure differential
(3) Fluid viscosity
(4) Travel and distance
(5) Friction
(6) Seat material
(7) Actuating force (if solenoid actuated, current and coil winding would be important).

Inertia of moving parts should be kept to a minimum for short response times. In most cases this can be done by keeping the mass of moving parts as low as possible. In some designs rotary inertia may be an important factor, and attention to the moment of inertia of rotating parts is highly appropriate. The effect of the mass of moving parts is exemplified by results of a test run on blast closure valves designed and constructed by the Mosler Safe Co. (ref. 3). This valve consists of two sets of stainless-steel dowels, one set stationary and the other movable. The movable dowels move against the stationary ones to effect a seal under the impulse of a shockwave. With hollow dowels, the closing time was about 5 milliseconds; with solid dowels, the closing time fell between 4 and 9 milliseconds.

When soft-seat materials are substituted for hard-seat materials, response time is lengthened, since the softer materials compress and require longer strokes. If solenoid actuated, the longer strokes usually mean a reduction in the magnitude of available actuating forces.

The actuating force is a prime factor in determining speed of response. Within limits, response time can be shortened by incorporation of more powerful actuators. However, actuators tend to increase in size as their force output increases. This factor provides a constraint to allowable actuating force. In some applications, explosive squibs may be admissible as compact sources of the necessary actuation force.

The signal transport time must be included in the case of a valve having pneumatic actuation if the sensing point is remote from the valve and the signal is transmitted by pneumatic lines. In the case of a pneumatic actuator, the piston or diaphragm chamber volume must be given time to fill in many cases. The ability to supply the actuating fluid can, as a result, become a design consideration.

Associated with fast-response times are high seat and poppet loads; a suddenly applied load will produce a much higher stress than a gradually applied load. A valve having a high poppet velocity will introduce high seat stresses which tend to reduce the life expectancy of the unit.

If springs are used to produce poppet motion, the velocity at which the spring can expand may be a limitation. This possibility is illustrated by a BB gun in which a pellet is propelled by an expanding, coiled spring. The velocity attained by the pellet is limited by the velocity that the end of the spring can attain.

A quick-response bipropellant valve developed by Moog Servocontrols, Inc., is a good example of a short-response time valve. Its design is illustrated in figure 31. The need for fast response arose because the response time of attitude-control rockets on space vehicles is a function of the valve response; and the minimum obtainable impulse bit is significantly influenced by the propellant valve characteristics. This valve opens two ports simultaneously with one signal. It has no sliding parts; the poppets are retracted from their seats by the rotation of an armature which is actuated by a torque motor. As reported by Bailey (ref. 4), this valve for use with a 100-pound rocket thrust chamber has response times of less than 5 milliseconds. Valves of this design for very-low-output thrust chambers, i.e., ½ to 10 pounds thrust, are also available with response times as low as 2.5 milliseconds.

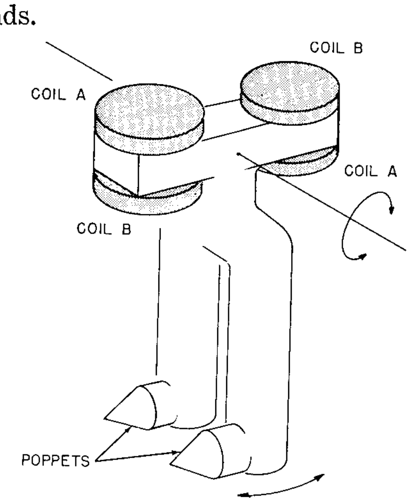

FIGURE 31.—Quick-response, bipropellant valve with no sliding parts.

The considerations which led the designers to choose a torque motor rather than a solenoid as the actuator illustrate the importance of the actuating device. A torque motor can convert electrical power into mechanical motion with considerable force. A solenoid also converts electrical power into mechanical motion, but the torque motor utilizes permanent magnets to provide additional flux in the airgaps. A signal applied to the torque motor coils generates an increased flux in one pair of the working airgaps and reduces the flux in the opposite pair. An unbalanced force is thus produced for rotating the poppets. The requirement for fast response demands that the flux level needed to move the armature be reached quickly. A simple analysis shows that the current level required for a torque motor is less than for a solenoid. Therefore, in view of current buildup characteristics in a resistive-inductive circuit, the time required to reach the current level needed to move the armature is less for the torque motor.

One limitation, which is being corrected in a new design, is that the pressure of both the fuel and the oxidizer must be within 50 psi of each other. When a pressure difference greater than 40 psi is encountered, one poppet opens or closes about 0.5 millisecond too early. This permits an oversupply of either fuel or oxidizer to the thrust chamber of an engine. The valve body is assembled with press fits followed by welding. The seat material is 17–7 PH stainless steel with a Teflon insert in the seat to minimize leakage.

Experimental test data on the valve illustrated in figure 31 are of particular interest. When tested as an all metal-to-metal poppet/seat design, leakage rates were:

Pressure, psig	Oxygen port, cc/hr	Fuel port, cc/hr
50	56	8
100	124	16
200	408	40
300	672	276

After incorporating Teflon inserts in the valve seats of another valve, the leakage tests were repeated. The leakage rates then were:

Pressure, psig	Oxygen port, cc/hr	Fuel port, cc/hr
50	4	0
[a] 100	1/16	3/4
200	1	1
300	1	0

[a] From a 16-hour test.

The response times for both the hard-seat and the soft-seat tests were recorded and are:

Seat	Opening time, msec.	Closing time, msec.
Hard	4 to 4.5	1.5 to 2.25
Soft	4 to 5	5 to 6

Important tradeoffs exist between valve response and weight, cost, and operating life. A requirement for shorter response time than is actually needed will incur penalties in increased weight and cost, and in decreased operating life.

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CHAPTER 9

Actuators, Position Indicators, and Computer Control

Many valve problems in both industrial and aerospace applications concern accessories rather than the valve itself. These accessories include actuators, position indicators, and computer systems used to control valves.

The most basic valve actuator is a manually operated handwheel or lever. Most aerospace valves, however, are powered by remotely controlled actuators. The variety of powered actuators commercially available for industrial uses further testifies to the importance of these accessories.

Despite research and development efforts, personnel at NASA's Lewis Research Center report a number of problems associated with valve actuators, particularly electrohydraulic control actuators. These contain pistons, hydraulic servo valves, and means for position feedback. Some of the problems are:

(1) Leakage of hydraulic fluids through seals

(2) Poor seal life

(3) Special nonstandard piston-size requirements

(4) Lack of rigidity in the attachment of the linkage for feedback position control

(5) Inadequate control over fits and tolerances

(6) Inadequate resolution. Honed and lapped cylinders and pistons are used in place of O-ring seals where high positioning resolution is required.

(7) Different flow rates depending upon the direction of piston travel. Only double-ended pistons are used so that identical rods extend in both directions from the piston. By this means, differential speeds and flow rates are not present when the direction of piston movement changes.

DIGITAL ACTUATORS

The increasing use of digital computers to control systems has accentuated the need for actuators compatible with the computer signals and the frequent requirement for a continuous actuator output. Digital-to-analog converters sometimes are used to render digital command signals compatible with conventional final control elements, It is more desirable to use a digital device that does not require digital-to-analog conversion, but such devices are not yet highly developed and often are quite complex.

Actuators accepting a digital input have been developed that approximate the desired continuous output of one consisting of numerous discrete steps. Many of these involve some type of stepping motor (usually electric), which moves or rotates in a discrete step for each signal pulse received. If electric, these motors generally have multiple windings which are energized in a specific sequence to cause clockwise or counterclockwise rotation. This type of actuator may be used to control a conventional pneumatic pressure regulator, which in turn may position a valve mechanism; or the stepping motor may be used directly for positioning the desired mechanism.

As these devices themselves require extended discussion, only an introduction is attempted here. A paper by Holben (ref. 1) may be of interest in this regard. A further reference exemplifying this approach is a report by Seidel (ref. 2) in which an electrohydraulic servo valve for use in digital flight control systems is discussed. He describes a servo actuator consisting of an analog servo actuator utilizing a multicoil torque motor for binary weighting and summation operations which is responsive to parallel digital electrical information.

A. E. Stone, Chief Engineer, and R. K. Madsen, Application Engineer, of the Honeywell, Inc., Valve Division in Fort Washington, Pa., are the authors (ref. 3) of the representative paper which follows:

There has been recent widespread interest in the concept of direct digital computer controlled processes. The reason for this interest is, quite naturally, one of economics. Conventional process control methods call for a process sensor, a controller, and a final control element for each loop in the process. If the process has enough loops, there will be a point where a single digital computer can be substituted for all of the controllers at a net financial gain.

The traditional stumbling block of this concept is the final control element. Almost all final control elements are analog in nature. That is, they operate from a continuous signal rather than from a signal composed of discrete steps or digits. For this reason conventional valve actuators are incompatible with the type of signal output that would come from a digital computer. The most direct way to get around this problem is to insert some sort of converter between the computer and the actuator which would receive the computer output and convert it into an appropriate analog signal suitable for the actuator. This has been done and it works. But the cost is high.

Background

The purpose of this paper is to describe an actuator which receives a signal which is digital in nature and operates a control valve directly. In the development of this actuator it was decided that everything possible should be done to retain the basic concept of the diaphragm actuator. The reasons for this were:

1. The diaphragm actuator is the simplest and most dependable way to operate a control valve.
2. The maintenance requirements are very low and maintenance people are familiar with them.
3. They are the least expensive way of getting high thrusts.
4. They are mechanically compatible with existing control valves.

It would be possible to develop a special-purpose actuator which has a number of discrete positions available on command from a computer, but this actuator would need 1000 such positions to give the same degree of resolution available from today's diaphragm actuators. An actuator of this type would probably be very large, very complicated, and very expensive. It was therefore decided that the best approach to getting a digital actuator was to develop a valve positioner which could receive a step-type input and drive a diaphragm actuator with the output. This seems even more practical when one considers that an electropneumatic positioner or transducer is required on every valve used with modern electronic control systems anyway. The cost of a digital control valve would be higher than a conventional valve by an amount equal to the difference in cost between an electropneumatic positioner and the digital positioner. It turns out that this difference in cost is modest, and digital control valves can thus become practical.

Principle of Operation

We can learn the principle of operation of this positioner by comparing it to an electropneumatic positioner which is by now a familiar instrument. An electropneumatic valve positioner can operate on the force-balance principle wherein the input signal and feedback motion are both converted into forces which are then applied to a common member. Any error is reflected as a net "unbalance force" which can be used to operate a pneumatic flapper-nozzle system. In the case of an electropneumatic positioner the input signal is a current, usually of milliampere magnitude, which can be converted to a force by feeding it into a coil situated in a magnetic field. The force is then directly proportional to current. The positioner is controlling valve stem position, so this position must be converted into a feedback force. Of course, motion can easily be converted to force by means of a spring, and this "feedback spring" is the method usually employed in a force balance positioner. Now, if we have an electropneumatic positioner we should be able to change the nature of the input by substituting some other force for that force supplied by the input coil. This is exactly what was done to make the digital valve positioner.

In the digital positioner a dc stepping motor is the element that receives the input signal and converts it into a force. This motor replaces the input signal used on the electropneumatic positioner. The motor moves in discrete steps, each of which amounts to 1.8 degrees of rotation. The shaft of the motor is threaded and has a fixed nut which cannot rotate but is free to move up and down the shaft as the shaft rotates. The nut is in contact with a spring which provides the input force.

At this point it would be helpful to study the dc stepping motor in more detail in order to learn its principle of operation and the nature of the input signal. These motors have a rotor which is really no more than a permanent magnet with teeth milled into it. The stator has coils which can be energized with either positive or negative polarity. Figure 32 illustrates a motor as it goes through one sequence of steps. The end of the rotor shown is the north pole. When phases A and B are both energized positive as at position 1, it creates the effect of a north pole at the top and a south pole at the bottom. Therefore, a rotor tooth will align itself opposite the south pole at the bottom, and a gap will appear opposite the north pole at the top. This is consistent with the principle of attraction of opposite poles since the teeth of the rotor have a higher magnetic flux density than do the slots. If we reverse the polarity of the phase A coils, the magnetic field rotates 90° clockwise as shown in position 2, and the rotor turns ¼ tooth. Reversing phase B polarity shifts the magnetic field another 90° in the clockwise direction, and restoring the original polarity to phase A shifts it once again. It can be seen that four polarity changes cause the rotor to turn 1 tooth. The illustration shows a rotor with 5 teeth for the purpose of simplification. Actually, the motor has 50 teeth, and 200 steps are required for a complete revolution of the motor.

There is another type of motor which uses the same principle described above, but has a slight modification. This motor has half of each coil wound in the opposite direction so that it can be stepped by switching from one half of the winding to the other rather than changing the polarity of the coil. This motor is said to have bifilar windings and is the motor principally used in the digital positioner. Figure 33 shows the switching sequence required to get clockwise rotation. Counterclockwise rotation is obtained by simply reversing the switching sequence.

A schematic representation of the positioner is shown in figure 34. A dc switching sequence is fed into the motor windings causing rotation of the threaded shaft. The nut then travels up the shaft because it is restrained from rotating. This motion lets the nut bear against a spring which places a force on the flapper beam and moves the flapper away from the nozzle a slight amount. When the flapper moves away from the nozzle the air pressure behind the nozzle is reduced because the resistance to flow is reduced. This back pressure is directed to a pneumatic booster relay, and since the pressure into the booster was reduced, the pressure output is also reduced. This lower output pressure allows the diaphragm actuator to move upwards

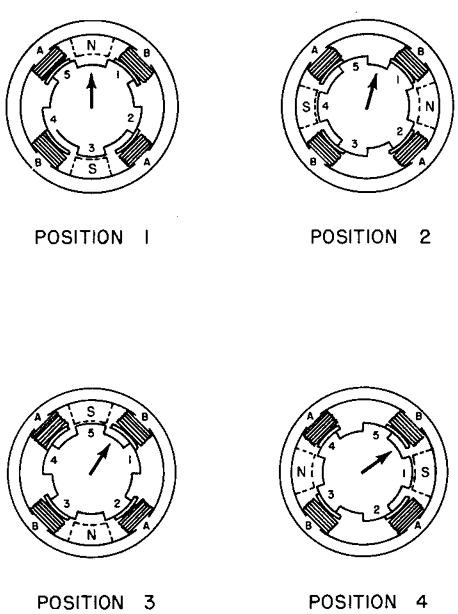
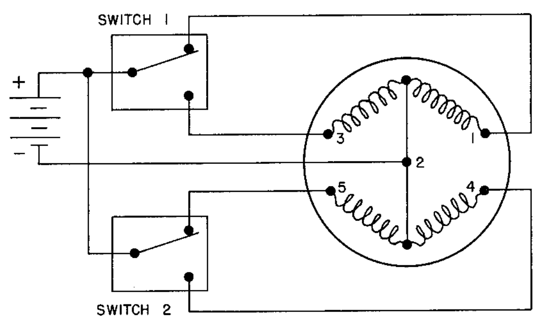


FIGURE 32.—Position illustrations.



STEP	SWITCH 1	SWITCH 2
1	1	5
2	1	4
3	3	4
4	3	5
1	1	5

FIGURE 33.—Switching sequence.

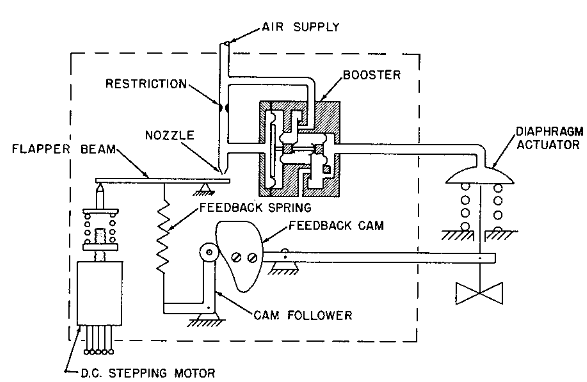


FIGURE 34.—Schematic of digital valve positioner.

under the influence of the actuator spring. As the actuator stem moves up the feedback cam rotates counterclockwise as does the cam follower. This action results in a downward force on the flapper beam which opposes the upward force applied by the input spring. When the moments on the beam are equal, the instrument is in equilibrium and the control valve has reached the new position called for by the computer.

Characteristics

The digital valve positioner is set up to have a 1000-step span for full valve travel. This gives a resolution roughly equivalent to a standard control valve with a positioner. The span remains constant for any valve travel within the ½-inch to 4-inch range of the instrument. The motor will accept dc steps at any rate up to 400 steps per second. This means that the instrument can give full output in a minimum time of 2½ seconds. The motor has a power requirement of about 5.5 watts.

The positioner has an air handling capacity of 7 standard cubic feet per minute with a 20-psig supply pressure. The maximum supply pressure is 50 psig. Air consumption is about 0.3 standard cubic foot per minute when the instrument is in equilibrium.

The feedback is accomplished through a cam which normally gives a linear relationship between valve position and number of input steps received. Other cams can be used to give nonlinear relationships when required.

The positioner can be used with several types of control. One type is valve stem position control. Here the computer measures the process variable and computes the position to which the valve should move to correct the error, if any. It then sends the required number of dc steps to the positioner at a rate not exceeding 400 steps per second. With this type of control the computer has to remember the last position of the valve.

Another type of control is valve stem velocity control. In this case there are several switching frequencies available to the computer on demand. The computer then determines how far the process is off set-point and computes what corrective action is required. Depending on the magnitude of the correction, the computer selects one of the programed switching frequencies and sends the signal to the positioner. If the error is large, a high frequency is sent to the positioner resulting in a high stem velocity. If the error is zero, no signal is sent to the positioner and the valve stem holds its position. One advantage of this type of control is that once the frequency is selected, the computer is free to scan the next loop. The switching frequency selected will continue to go to the positioner until the computer selects a new one. Another advantage is that no valve position memory is required.

Summary

The digital positioner was developed with the aim of retaining the conventional pneumatic diaphragm actuator as the means of operating the control valve. The positioner receives a dc step which is actually a switching sequence directing current to the motor coils in a predetermined program. The output of the positioner is an air pressure which is applied to the diaphragm actuator. Valve stem position is fed back into the positioner through a cam arrangement, so the instrument becomes a closed loop device.

Digital hydraulic servo valves developed in the past have not met the prime requisites of basic simplicity and high reliability. Hybrid systems have been developed as compromises. The Autonetics Division of North American Aviation, Inc., for example (refs. 4 and 5), has extensively studied a system in which all electrical signals are digital, but the hydraulic devices are conventional analog components. A standard two-stage electrohydraulic servo valve was modified so that it would accept digital signals. The modification was made by winding the valve torque motor coil in several segments. Each coil segment is driven by a separate current source that is switched on or off at the command of the servo loop position error control. Analytical and experimental data show that performance comparable to analog servo systems can be obtained with this hybrid. The basic concept has inherent redundancy and, therefore, high reliability; the hybrid servo operates at reduced performance with most failures of individual components.

A hydraulic digital actuator developed by Vickers, Inc., division of Sperry Rand (ref. 6), allows digital control systems to position an output shaft lineally. No intermediate digital-to-analog converters, servo amplifiers, or proportional servo valves are needed. The prototype model shown in figure 35 has 10 pistons: 8 binary-weighted active pistons and 2 passive pistons. Active pistons 1 through 8 determine the output shaft's steady-state position; when the pistons are pressurized they move until they touch mechanical stops. Passive pistons 9 and 10 dissipate the kinetic energy imparted to the

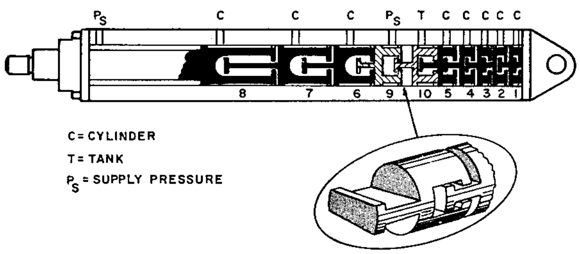


FIGURE 35.—Eight active and two passive (cross-hatched) pistons convert binary-coded electrical input signals into lineal motion of the output shaft.

load on full-scale movements; one buffers forward and the other buffers reverse motion. The pistons are pressurized and depressurized through three-way, two-position, on-off valves. The fixed mechanical stops determine the allowed amount of position motion. The output piston can assume 2^n discrete positions, where n is the number of binary-weighted active pistons.

The prototype model output shaft can take up 256 discrete positions in a full stroke distance of 2.55 inches, with a minimum movement of 0.01 inch.

Two versions of the three-way, two-position, on-off valves were made. One is a miniature fast-response type requiring only 0.12-watt electrical input power. Its two-stage design needs hydraulic flow only when the valve changes state. The actuator with miniature valves was primarily intended for aerospace applications. Typically, a 271-pound load can be moved 1.92 inches in 90 milliseconds.

The other version integrates the valves with the actuator housing and was intended for industrial uses such as process control valve operation. This version is larger than the miniature version and has a slower response. It takes 28 watts of electrical input power.

FLUIDIC ACTUATORS

The fluid amplifier is receiving increased industrial attention. Its similarity to an active electronic element has led to use of the term "fluidics" to refer to these devices. High reliability, environmental tolerance, and design simplicity make fluid amplifiers attractive. A brief review of some pertinent references which show some industrial systems and control circuits will serve to indicate current applications.

An article by Auger (ref. 7) on fluid logic control systems mentions several applications for TA (turbulence-amplifier) fluidics. Means for using TA systems to operate conventional pilot-operated valves are discussed and illustrated. Valve stem positioning through use of a cam which is followed by an IJ (interruptible jet) with a TA as a pneumatic amplifier is also discussed. So is a scheme by which high-pressure valves can be directly actuated by TA's controlled by IJ's to eliminate relays and solenoid-actuated valves.

The commercial state of many fluidic devices is reviewed by Reason (ref. 8), who illustrates many presently available products and cites the manufacturers. A diverting valve, available in a two-position or a proportional version from Moore Products Co., is described in this article and is more extensively explained by Mamzic (ref. 9). This valve can completely divert flow in 0.1 second and is free of shock or water hammer.

Direct electrostatic control of fluid jets is possible, according to analytical and experimental results reported by Jorgensen and Lee (ref. 10). Present fluidic devices rely on a momentum or pressure interaction with the main power jet. However, an electrically charged jet experiences a body force when exposed to a uniform electric field; an electrically neutral jet experiences a body force when exposed to a nonuniform electric field. If electrostatic control proves feasible, a very-high-temperature electrostatic valve actuator may be possible.

A pneumatic nutator actuator motor for drum control of a nuclear reactor was developed by the Bendix Corp. (ref. 11). A high-torque, low-speed motor contains an integral transmission consisting of a pair of bevel gears, one of which is operated in a nutation motion by pressurized bellows located around its periphery. Pneumatic flow is commutated to the bellows by vortex-type fluidic devices.

REPEATABILITY

In automatic system control, a definite series of events must occur in a proper, timed sequence. Nonrepeatable valve response times lead to erratic total system behavior.

For example, in industrial processes which require mixing of two fluids, the response time and repeatability of the valves that start and stop the flow of each fluid may be critical factors. Recent advances in aerospace valve technology have produced improved "bipropellant valves" ideally suited for simultaneously controlling flows of several fluids in industrial mixing operations. These valves have a single-actuating mechanism to control the flow of two or more fluids through separate ports, thereby eliminating the synchronization problem associated with the use of separate, individual valves in each fluid line. Since only one actuating means is used, both valve poppets act together. When separate actuating mechanisms are unavoidable, an analysis of valve repeatability is important.

A detailed study of the repeatability of solenoid valves in synchronized, paired operation was accomplished at the Manned Spacecraft Center in Houston. A careful review of the solenoid valve current traces produced in this study gives valuable insight into the many problems associated with synchronization of valves, whether they be electrical, pneumatic, hydraulic, or mechanical types.

Two identical, high-quality, commercially available, solenoid valves were tested in the as-received condition. Fuel flowed through one valve and an oxidizer flowed through the other valve. The object of the test was to determine the factors related to obtaining exact mixing ratios, since an excess of either fuel or oxygen could result in system malfunction.

The solenoid valves tested used a spring to hold the poppet against the seat. The solenoid pulls the plunger and poppet away from the seat and compresses the spring. Upon release of current to the solenoid, the spring moves the plunger and poppet to a closed position against the seat. A current probe was used in conjunction with an oscilloscope to produce the chart in figure 36.

Figure 36(a) shows the solenoid current trace with no plunger motion. Figure 36(b) illustrates the change in the curve when a plunger motion is introduced. From time zero, current builds up in the solenoid until point T_1 is reached. Motion of the plunger introduces a voltage that opposes the buildup of current. Because the plunger motion starts quite abruptly, the peak in the waveform at T_1 is actually somewhat sharper than shown in figure 36(b) and, for practical purposes, the instant of initiation of plunger motion may be taken as occurring in coincidence with T_1, the instant when the current actually begins to decrease. Current continues to decrease as plunger motion

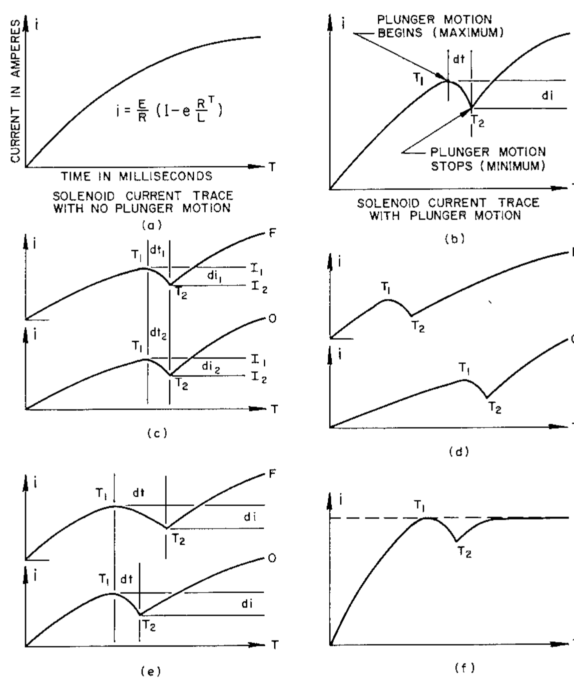


FIGURE 36.—Solenoid valve current traces. O = oxidizer valve; F = fuel valve.

continues until the plunger motion stops at point T_2. Then current begins to build up and levels off when the solenoid steady-state current value is reached. The decrease in current during plunger motion is indicated by di. The duration of plunger movement is indicated by dt.

Figure 36(c) illustrates the ideal synchronization of two valves, one controlling fuel and the second controlling the oxidizer. In this theoretically perfect, matched pair of valves, both plungers begin motion at the same instant, T_1, end motion at the same instant, T_2, and thereby achieve the same time of travel, dt_1 being identical to dt_2. Also, the current levels when plunger motion begins, I_1, are identical; the current levels when plunger motion stops, I_2, are identical; and the change of currents is identical for both valves, di_1 being equal to di_2.

Figure 36(d) illustrates a test condition where both valves received a signal at the same instant but the motion of one valve was retarded. In this illustration, the durations of plunger motion between points T_1 and T_2 are identical. This pair of traces indicates that either a different voltage level existed between the two solenoids, or else there was a difference in the solenoid winding.

Figure 36(e) illustrates a condition where electrical characteristics of both valves are identical but, due to some mechanical problem, the fuel valve plunger moves slower than the oxidizer valve plunger.

Figure 36(f) indicates valve-sticking problems where the current required to begin plunger motion reached the final steady-state current level before movement began. In figure 36(f) the curve would rise to T_1 and then level off if the plunger of the valve should stick.

The effect of pressure upon two supposedly synchronized valves is illustrated in figure 37. As the pressure increases from 0 psig through steps up to 500 psig, the time lapse from signal input to the beginning of plunger motion also increases. In addition, as the pressure increases there is an increase

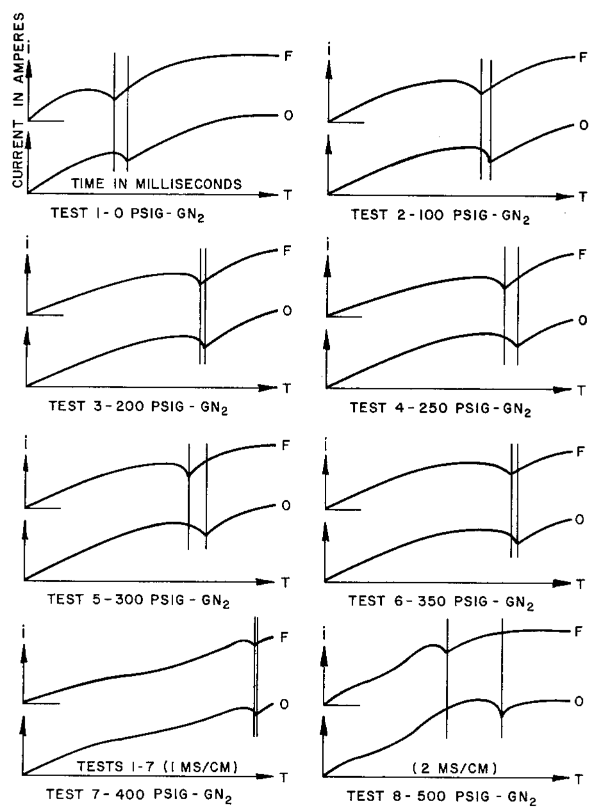


FIGURE 37.—Solenoid valve tests.

in the level of current necessary before plunger motion begins, since high pressure is exerted on the poppet from inside the valve to hold the poppet against the valve seat. Should the flow be reversed, with pressure applied to the valve seat, the pressure would tend to open the valve, thereby decreasing the current necessary to operate the valve. The change of plunger velocity with pressure is readily apparent. As pressures and, therefore, actuating current increase, plunger velocities also increase. The slope of the curve between the points where plunger motion begins and plunger motion stops is indicative of plunger velocity. At lower pressure, this slope is gentle and longer periods of time are used for plunger travel; at the higher pressures approaching 500 psig, the slope of the curve becomes very steep. This curve indicates the presence of extremely high plunger velocities, high impact forces, and high stress levels in the valve structure.

It should be noted that the valves tested in this program were not representative of the best available solenoid valves. The results, however, depict problems that are still being found in many solenoid valve designs. This is especially true in valves with an excessive number of sliding parts and fits.

VALVE POSITION INDICATORS

The position of valves is monitored in many aerospace and industrial systems to furnish a signal indicating the true position of the valve. The monitoring operation is normally accomplished through the use of metal diaphragms and springs. Trouble can arise from the temperature cycling of these springs and metal diaphragms; changes from ambient temperatures to either cryogenic or liquid metal process temperatures may cause these parts to weaken and take a permanent set. Also, when these parts remain in a given position over a long period of time, the position actuators may stick and produce erroneous position signals.

Pressure switches have been a problem on the Saturn V program at Marshall Space Flight Center. Component response requirements are coupled with specifications for wide environment ranges in both temperature and pressure. Development work is being done to improve the reliability of operation of these components. Klixon (Texas Instruments), pressure-sensitive switches are being used in environments below —100°F, but with a minimum specified relief force much greater than that used at ambient temperatures. It may be that fluidic devices will be useful to satisfy the requirements of these position-indicating applications (ref. 7).

COMPUTER CONTROL OF VALVES

In many cases, computers are receiving input signals from valve position indicators and feeding output signals to valve actuators for the control of process liquids, gases, and mechanical equipment using pneumatic and hydraulic systems. Information is available on the use of analog computers for controlling processes. Digital computers are now beginning to take their place in the automatic control of industrial processes and systems. As digital computers become more economical, they can be used to control a relatively small number of valves. This would alleviate the requirement for very high computer reliability and hasten their adoption for such service. Fluidic logic circuits also show promise and have already been applied to natural gas pipeline systems.

Apparently the integrated design of valve and actuator assemblies is a promising area. The advent of digital controls and fluidics may well provide the means for solving some current problems.

When industrial processes require synchronized valve motion for mixing, repeatability of valve motion (specifically, poppet motion) should receive critical review. Consideration should be given to the use of the newer type of single-actuator bipropellant valves.

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CHAPTER 10

Valves for Extreme Pressures and Temperatures[1]

What is considered an extreme pressure or temperature in one application may be considered moderate in another. A consumer usually will not buy much more valve than will reliably do the job for him. Its construction will be that which is most suitable for the specified operation conditions, and which will develop the potential of the materials used.

Valve operating pressures range from vacuum to 100 000 psi; temperatures sweep from the cryogenic level to over 2000° F. Materials for valve fabrication must be selected for availability, workability, economic feasibility, and satisfactory physical properties under these conditions. The suitability of a valve design and the materials embodied in it depend upon both the operating temperature and the pressure.

The ability of a material to withstand loading typically decreases as temperature increases. Thus, if operating pressures are high (and not even extremely so), the high-temperature capability of a valve is significantly reduced. Many materials gain strength as temperature decreases into the cryogenic range. The utility of this characteristic, however, is somewhat offset by the relative rarity of high pressure-cryogenic temperature requirements. More importantly, a loss of ductility frequently accompanies the gain in strength; a pressure-containing part's failure could be of a brittle nature and might be catastrophic.

The low pressures of vacuum conditions pose unusual material selection problems. Since the severity of these problems depends upon the level of vacuum, it is helpful to review the classification of the degrees of vacuum. According to the standard vacuum terminology proposed by the American Vacuum Society, the classification of degrees of vacuum are: rough vacuum, 760 to 1 torr; medium vacuum, 1 to 10^{-3} torr; high vacuum, 10^{-3} to 10^{-6} torr; very high vacuum, 10^{-6} to 10^{-9} torr; and ultrahigh vacuum, 10^{-9} or less. In vacuum applications for valves, materials must be chosen judiciously with respect to outgassing characteristics. Outgassing may otherwise be of such a magnitude as to preclude attainment of ultrahigh vacuums. In sealant materials, a degradation of properties may permit an influx of air at a rate beyond the capacity of the vacuum pumps. Further, cold welding of metals can render a valve inoperative and reduce it to little more than a useless piping extension.

Materials for valves to operate at pressures up to 75 000 psi are equally difficult to select. These pressures exceed the tensile capabilities of many materials commonly used. Because of the hazards of the polymerization processes in which these valves are used, testing to 130 percent of the maximum operating pressure is required. In the case of ethylene gas at 75 000 psi, the test pressure is 97 500 psi. Allowable stress levels must be established at some percentage of the material yield limit, and operational safety factors must be set up. Further, the material must be tough enough not to shatter or to permit preferential stress cracking. Laminated construction is difficult to apply to a structure as complex as a valve

[1] Based on a paper, "Valves for Extreme Pressures and Temperatures," by D. J. Easton, Rockwell Manufacturing Co., presented at the Valve Technology Seminar, Midwest Research Institute, Kansas City, Mo., Oct. 21–22, 1965.

body. Although auto-frettage[2] tends to alleviate the stress situation, it is apparent that significant problems are posed.

The types of commercial valves suitable for various combinations of temperature and pressure are shown in figure 38. Operating conditions greatly influence the choice of materials.

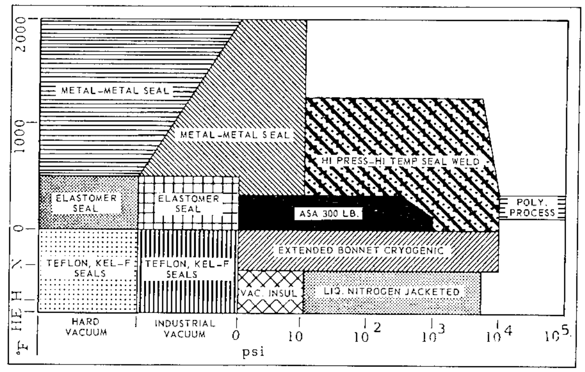


FIGURE 38.—Valve coverage by type.

VALVE TYPES

Basically, there are three major classifications of valves: (1) the quarter-turn plug, (2) the gate, and (3) the globe. All three types are used for extreme conditions. In plug valves, the plug-flow passage is rotated 90°; in gate valves, a flat plate is slid across annular seats which circumscribe the flow passage; and in globe valves, a sealing member is pushed into the flow passage in much the same manner as a cork is pushed into the neck of a bottle. Other so-called valve types, such as the butterfly or vane-type valve, fall into one of these three categories or some combination of them.

Plug Valves

The tapered plug valve is the oldest design, probably because it is the most tolerant of inability to maintain close dimensional control. Simple quarter-turn tapered plug valves (see fig. 39) are limited in high-pressure application by the rapid increase of operating torque with increasing pressure differential. If the fluid has access to the large or the small end of the plug, the plug is unbalanced into or out of the valve body. The plug taper greatly influences the operating torque; if the plug has a "locking taper" and if the plug is considerably unbalanced into the taper, even very high operating torques are liable to be insufficient to operate the valve. Although the pressure at which "locking" occurs may be low by many standards, it is an extreme condition for this design and constitutes a pressure limitation.

A lubricant such as a grease reduces the

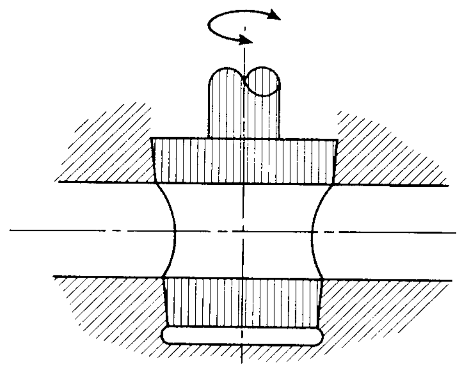


FIGURE 39.—Tapered plug valve.

tendency of the plug to "lock," but plugs occasionally lock even though lubricated. In such a case, the plug can be loosened in its taper by mechanically unseating it. The plug cocks used on household gas lines are familiar to most people. These plug cocks are usually lubricated. When they "lock," the nut at the bottom of the plug is loosened and the plug is tapped out of its body taper with a hammer—not a recommended procedure, but an effective one. The point is that mechanically unseating the plug requires a much smaller force than that previously necessary to turn the plug. The pressure level to which tapered plug valves can be applied is, therefore, extended by the lubrication and unseating techniques.

The purpose of the tapered plug valve is to shut off flow in a leaktight manner. If the lubricant film is nonuniform in thickness, leakage will occur. The lubricant can be washed away by the process fluid when

[2] Locally overstressed surface fibers yield to produce a work-hardened skin of higher strength than that originally possessed by the material.

unseating the plug, or the lubricant can be extruded by the applied pressure differential. Again a pressure limitation of this design is encountered.

The three functions of the lubricant (lubrication, plug unseating, and viscous sealing) are combined in plug valves of the Nordstrom design illustrated in figure 40. This externally energized design increases the pressure level that can be considered extreme for the tapered plug valve. Increasing temperatures degrade the properties of the lubricant sealant. As a result, maximum temperatures for this valve type are presently 800° to 1000° F. The greatly increased lubricant-sealant viscosity found at very low temperatures precludes application of grease at cryogenic temperature levels.

Gate Valves

The selection of material for gate valves is very critical. A gate valve seals by sliding a flat plate over sealing surfaces in a valve body that is perpendicular to the fluid stream. This is illustrated in figure 41. The sealing surface must endure the sliding necessary to move from the fully open to the fully closed position. Consequently, the galling characteristics of materials and their ability to withstand bearing stresses are very important. The galling tendency generally increases and the strength generally decreases with increasing temperature. These two factors are not the only limitations, however. To provide good sealing surfaces, the gate and the seat should be flat. As a result, the gate must be thick, especially in large valves for high-pressure service. The shape of the valve body itself is such as to make the design problem difficult. Flexible and self-adjusting gates are available to compensate for distortion due to high pressures; their use extends the range of application and increases the pressure considered to be extreme for gate valves.

Viscous sealing is used in gate valves as well as in tapered plug valves. Sealant is injected into grooves which circumscribe the flow passage in the gate or in the seat. In the case shown in figure 42, an injection pressure which is proportional to the flowing fluid pressure is maintained on the sealant. Viscous sealed gate valves exhibit lower operating torques than conventional parallel slide gate valves. The general advantages and limitations previously described for lubricated tapered plug valves also apply to viscous-sealed gate valves.

A tapered-gate or wedged-gate valve as illustrated in figure 43 removes the neces-

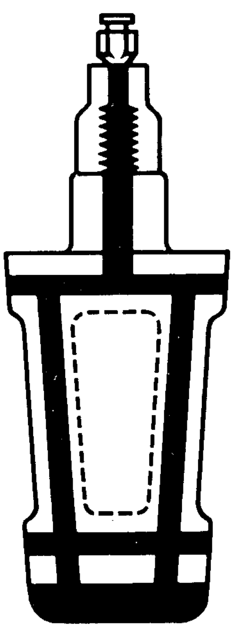


FIGURE 40.—Nordstrom plug valve.

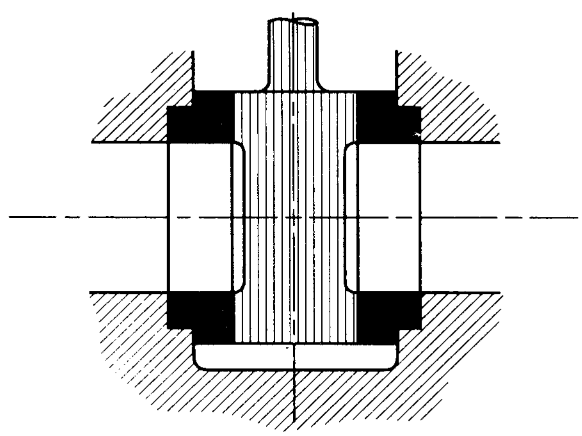


FIGURE 41.—Gate valve.

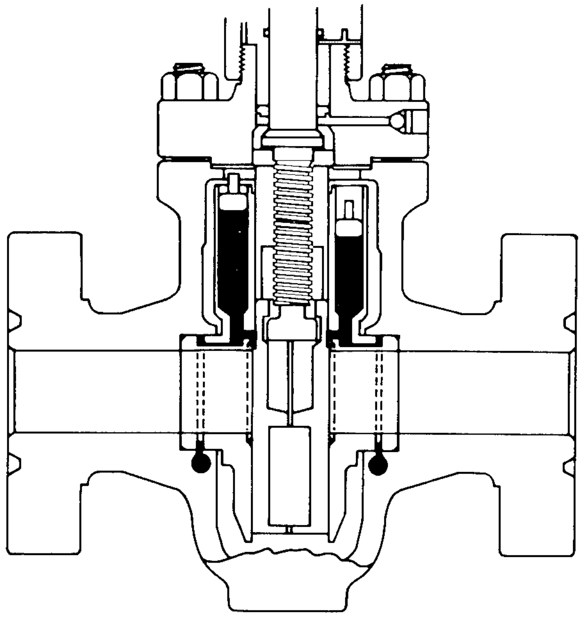


FIGURE 42.—Viscous seated-gate valve.

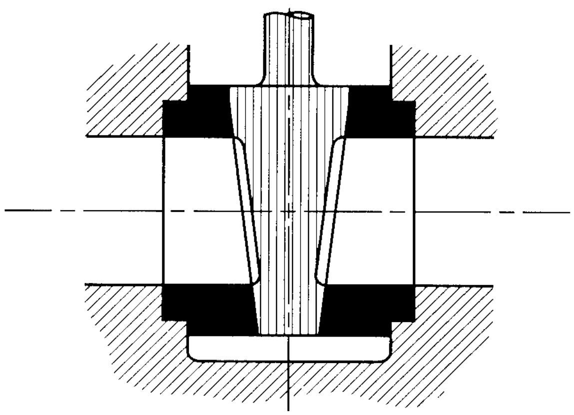


FIGURE 43.—Tapered gate valve.

sity of sliding the gate across the entire sealing surface. In principle, the seat and gate are not in contact once the gate has lifted a fractional part of its total travel. Side ribs on the tapered gate resist the tendency of flow impingement and pressure differentials to move the gate toward the downstream seat, keeping the sealing surfaces from sliding contact. When the valve is closed at a high temperature and then cools, the wedge-type gates tend to bind (sometimes immovably) because of the different coefficients of thermal expansion of the gate and the body parts. Flexible wedge gates relieve the severity of this problem and permit gate-valve application for high-temperature service.

Non-viscous-sealed gate valves are troubled by the difficulty of repairing the flat sealing surfaces and by leakage. The former problem arises principally from the inconvenient orientation of the sealing surfaces within the valve body. Damage to the gate and seat surfaces can easily occur in a short time when large-pressure differentials are imposed. The resultant "wire drawing" is often considered sufficient reason to preclude non-viscous-sealed gate valves from high-pressure service. Viscous-sealed gate valves, on the other hand, are successfully applied to gas and petroleum line pressure of 15 000 psi. Gate valves sealed by elastomer or fluorocarbon inserts are successful where contamination of the flowing fluid is a problem. Plastic flow of these inserts does present a pressure limitation.

Globe Valves

For globe valves expected to operate at high pressures, the closure plug or disk is of a frustoconical shape which fits into the mating body seat, as shown in figure 44. The

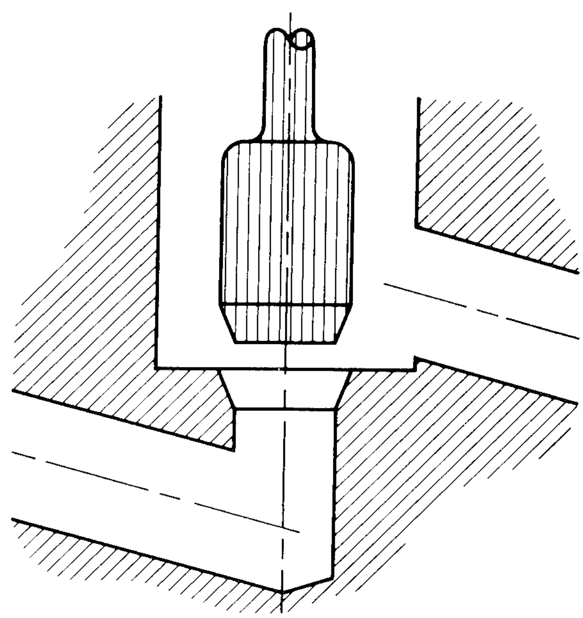


FIGURE 44.—Globe valve.

included angle is generally much greater than for the plug of a tapered plug valve. However, the concept is similar and wedging action is evident in both cases. Frustoconical seats are very resistant to pressure-induced valve-body distortions; and if such distortions occur, the elastic deformation in the seat tends to maintain satisfactory sealing contact. Further, this construction lends itself to lapping and superfinishing operations to assure intimate contact between sealing surfaces. If seat damage does occur, on-site repair can be made.

The basic design lends itself to high-pressure and high-temperature applications. Materials for these applications, of course, must be carefully selected. The configuration of the sealing surfaces permits facing them with a variety of hard and corrosion-resistant materials. Cobalt-based alloys such as the stellites, or materials such as carbides and ceramics, often are used for this purpose. The impact resistance of these materials is low, however, and their failure under tension is sudden and spectacular. Such materials considerably extend the range of conditions of application as far as the sealing function is concerned.

Frustoconical seats are satisfactorily used in valves for the high pressures of polymerization processes; however, very accurate grinding and lapping are required, since even a tiny leak could quickly destroy the seat by wire drawing and then demolish the body by erosive impingement. In such valves, a resistance-type seat is often used to shut off most of the flow before the main frustoconical seat closes. A double-ended stem with identical packing chambers at each end may be used to reduce pressure-generated imbalance and, consequently, the required operating torque. A tail rod catcher should be provided to restrain the upper part of the stem in the event of fracture because the upper part of the stem will be expelled from the valve with the speed of a bullet.

The globe valve is probably the type most universally applicable to high-temperature and high-pressure service, either individually or in conjunction with other valve types. Commercial valves are available for pressures up to 15 000 psi at atmospheric temperatures and also for respectable pressures of 5000 psi at 1200° F.

SEALING

The only job of a valve from the user's viewpoint is to control fluid flow, but many problems of high-pressure and high-temperature valves arise from parts that are not the primary flow-control parts.

Because the inner parts of the valve must be introduced into the pressure vessel or valve body, some closure or cover must be provided. Except for check valves or other internally automatic devices, an operating mechanism to activate the flow-control elements must be introduced through the pressure vessel. These two conditions generally require a minimum of two seals to contain the line pressure. This is illustrated in figure 45 and is typical of all three valve classifications.

Cover sealing can be accomplished in several ways. Each way provides a different attainable upper pressure level, and has a different economic cost.

Flat gaskets are not particularly suited to either high-pressure or high-temperature applications. Such seals depend on high sur-

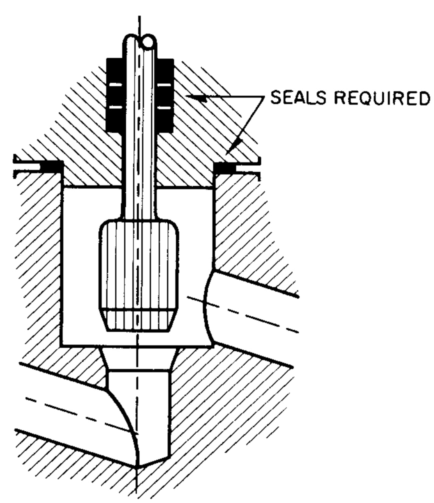


FIGURE 45.—Seal requirements.

face loading stresses to produce elastic and plastic deformations of the gasket and pressure vessel interfaces. As figure 46 indicates, an increasing internal pressure tends to separate the sealing faces. Extremely heavy bolting is required to keep the cover displacement within the limits needed to maintain the sealing stress above the minimum value at which sealing occurs. The bolting becomes disproportionate when applied to high-pressure designs. This has led to the development of self-energized gaskets wherein the sealing load is somewhat proportional to the contained pressure. The self-energizing gasket also tends to offset the effect of bolt extension.

One type of self-energizing gasket is the spiral-wound gasket. It consists of continuous and alternating plies of narrow metal ribbon and a softer nonmetallic filler material. The metal plies are crimped as shown in figure 47, so that increasing internal pressure tends to straighten the metal plies. Thus, the sealing surfaces of the gasket tend to follow the separating metal body and cover faces; the sealing load is generally proportional to the internal pressure. The displacement that the metal plies can accommodate is limited, however. The allowable bolt size, in other words, imposes an upper pressure limitation.

Another useful gasket of the self-energizing classification is the lens ring. The form of such a gasket is lenticular, as shown in figure 48. The outside diameter is provided with a short cylindrical section and the inner diameter, also cylindrical, is generally equal to that of the mating surface. These gaskets are mounted in shallow conical recesses and provide line contact near the inner edge in an unloaded condition. Initial bolt loading causes plastic deformation to produce a seal along a narrow contact area. As internal pressurization occurs, the gasket attempts to expand outward; this effect maintains sealing stresses that are proportional to the internal pressure. Lens rings and variations such as cone rings are suitable for pressures in excess of 100 000 psi.

The O-ring and its family of proprietary designs can be considered to be a self-energizing gasket. The O-ring is mounted in a groove which is dimensioned to provide a degree of radial or lateral interference, de-

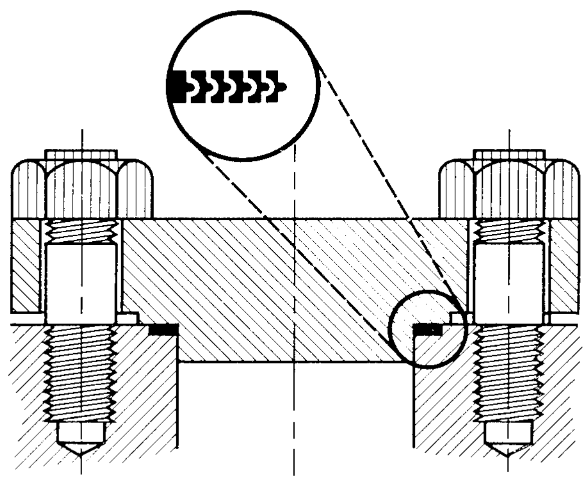


FIGURE 47.—Self-energizing gasket.

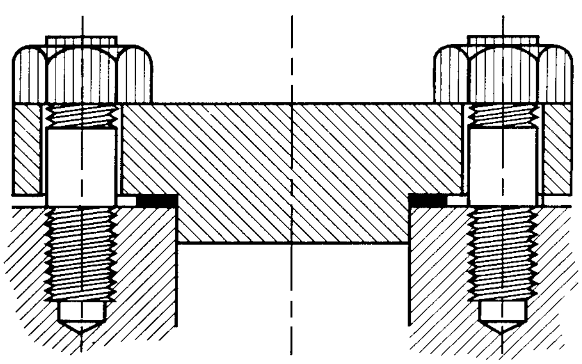


FIGURE 46.—Flat gaskets.

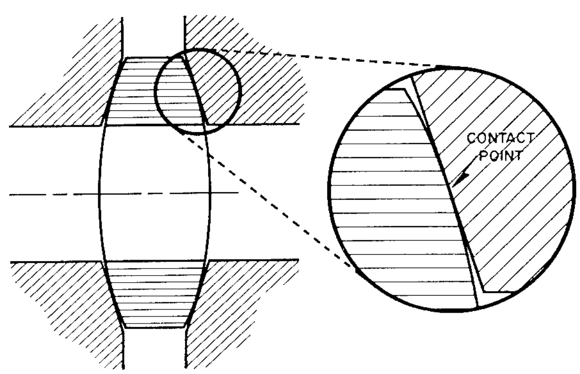


FIGURE 48.—Lens ring.

pending upon the application. As pressure increases, the O-ring deforms as shown in figure 49 to maintain sealing stresses which are proportional to the internal pressure. (See fig. 49.) Very high pressures cause the O-ring to extrude through gaps. This pressure limitation is controlled by the Durometer valve of elastomer O-rings and the size of the gap. It is possible to eliminate gaps by a metal-to-metal backup for high-pressure applications. However, this technique is restricted to static sealing functions. Metal O-rings are available and may be coated with an elastomer or fluorocarbon to insure sealing. Holes in the metal O-rings are sometimes used to admit line pressure into the rings, as in figure 50, in order that the sealing stress can be more nearly proportional to the line pressure. Metal O-rings are suitable for the highest pressures, but they are easily damaged and must be installed in specially designed split grooves (as is also the case for Teflon and other nonelastomerics).

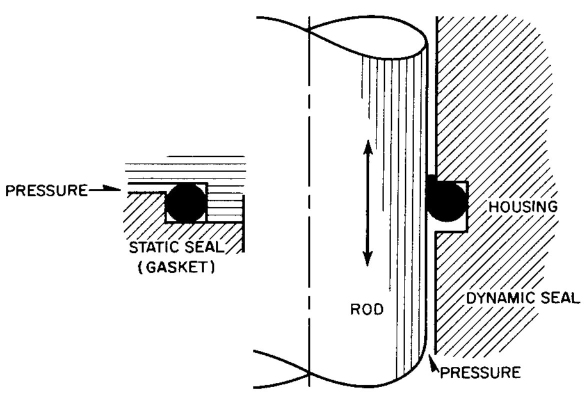


FIGURE 49.—O-ring mounting.

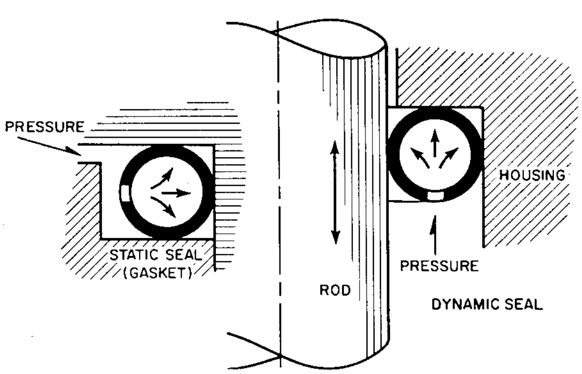


FIGURE 50.—Metal O-rings.

The pressure-seal ring, illustrated in figure 51, relies on displacement of one of its members to develop the sealing stresses. Preload bolting wedges the gasket against the surface to be sealed. Increasing internal pressure then wedges the gasket more and more tightly against its constraints. Unlike the spiral-wound gasket, the pressure-seal ring has no inherent upper pressure limitation. Factors of importance in this seal design are: (1) selection of gasket wedge angle, (2) provision of suitable bearing surfaces, (3) maintenance of adequate shear relationships within the gasket-support structure, (4) proper design to carry the hoop stresses developed in the outer member, and (5) selection of proper materials. Seals of this design can raise the definition of an extreme pressure above the range of current industrial practice.

Lapped seals achieve an initial seal from very smooth mating surfaces which require "super finishing" or lapping. This seal design is suited to either radial or axial installation, as shown in figure 52. Although such seals are suitable for high-pressure and high-temperature service, it is unlikely that they can be economically competitive for static seals with those previously discussed. It is possible, however, that they would be useful as a dynamic seal if tempera-

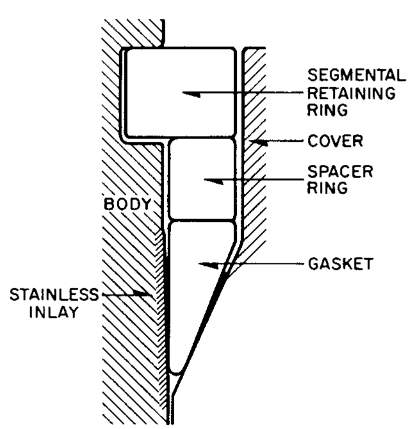


FIGURE 51.—Pressure-seal ring.

ture levels preclude the use of other designs. In a dynamic sealing application, the galling and wearing characteristics of materials must be carefully considered.

Body-cover joints can also be sealed by depositing a fillet weld bead at the interface. (See fig. 53.) Bolting may be used to provide additional support. However, the weld should not extend to the bolt circle—a consideration in conflict with the necessity of placing the bolts as close to the weld as possible in order to reduce flange bending stresses. This problem is sometimes solved by providing the body and cover with threads which are themselves able to bear the imposed loads. Sealing is provided by the fillet weld. This general principle allows seals to be maintained up to about 20 000 psi.

The canopy seal, shown in figure 54, is a variation of the seal-weld principle. It is used where a slight differential movement of mating parts is anticipated. While the canopy seal provides greater flexibility than a fillet weld, care must be exercised if the same high operating pressures are to be achieved. For example, the quarter-toroid segment must be as thin as is practicable in order to obtain flexibility. Hence, corrosion effects must be carefully evaluated. Canopy seals are typically fabricated from a corrosion-resistant alloy, such as stainless steel. If properly constructed, canopy seals offer all the advantages of fillet-weld seals and are even more tolerant of long-term creep of the mating parts.

Stem Sealing

Shaft sealing for high-pressure service is probably best accomplished by self-energized or Chevron packing, as illustrated in figure 55. This type of packing offers the advantages of self-energized sealing and self-compensation of wear. For high-pressure service, Chevron packing is normally preloaded by Belleville spring washers. Increasing pressure thereafter tends to create sealing stresses which are proportional to the contained pressure. Packing material is normally a reinforced elastomer or fluorocarbon, which may either be filled or unfilled. Fluorocarbons are usually filled with glass or carbon to achieve adequate density and resistance to

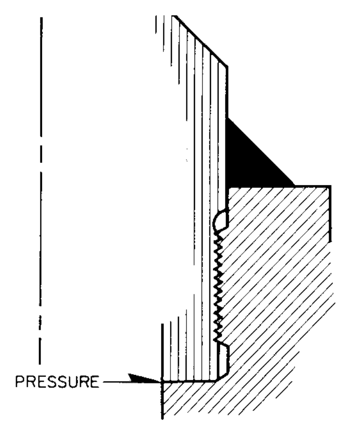


FIGURE 53.—Fillet-weld seal.

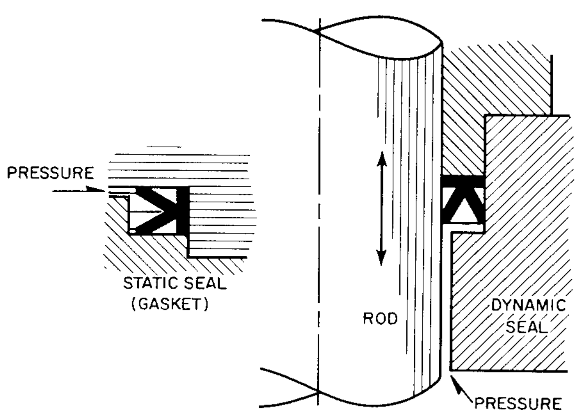


FIGURE 52.—Lapped seal.

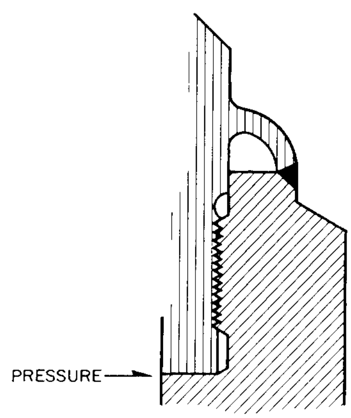


FIGURE 54.—Canopy seal.

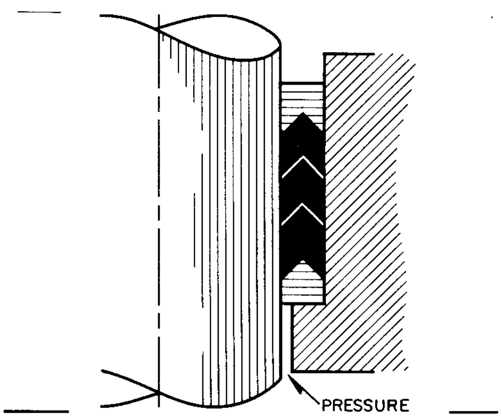


FIGURE 55.—Chevron packing.

cold flow. The Chevron-type packing is suited to high-pressure applications if clearances, material density, and cold-flow characteristics are properly selected.

Resilient seats are common for noncritical services such as household faucets. Their ability to provide tight shutoff also makes them attractive in applications involving large-pressure differentials. However, the resilient material tends to maintain sealing contact with metal mating parts as the valve is opened. It is possible, by correct design, to cause the resilient material to adhere to one metal surface and to release from a second metal surface without damage. Such seals are currently used up to 5000 psi. In high-temperature applications, selection of suitable resilient seating materials is very difficult.

Metal seats for high temperatures are mainly limited by metallurgical considerations. Hydrogen embrittlement is an important factor with fluids containing hydrogen; oxidation or corrosion of the metal by the process fluid must be considered.

Extreme service conditions present unusual valving problems. The materials and designs required lead to specialized valves which, because of their cost, are unlikely to be universally used.

CHAPTER 11

Valves for Cryogenic Applications[1]

Cryogenic processing imposes special requirements on equipment that operates in the frigid environment or comes into contact with the fluids. In control systems, this equipment includes measuring devices for determining process characteristics and devices for physical manipulation of fluids.

The term "cryogenic fluids" applies to gases which must be cooled below their critical temperature before they can be liquefied by pressure. Argon, krypton, methane, neon, oxygen. fluorine, nitrogen, helium, and hydrogen are among the cryogenic fluids. It is economical to store these fluids because of their high gas-to-liquid volume ratio. One pound of hydrogen, for example, occupies only about 0.23 cubic foot in the liquid state, but occupies about 192 cubic feet at 1 atmosphere and room temperature in the gaseous state.

To control these cryogens, valves have been commercially developed with properties compatible with the unusual physical and chemical characteristics of the fluids. Some factors to be considered in valve mechanical design are: the effect of very low temperatures on the strengths of metals and packings, part dimensions, and galling characteristics. Compatibility of the flowing fluid with lubricants and contaminants is also important.

DESIGN AND MANUFACTURING CONSIDERATIONS

The outstanding feature of cryogenic valves is an extended bonnet to keep the packing chamber as remote from the main fluid flow as possible. The extended upper works have more than one purpose. The packing material must be protected from low temperatures in order that it be as plastic as possible. Freezing of valve-stem packings is also avoided with this feature. Further, for manually operated valves it is desirable that the valve handle or wheel not be too uncomfortable to the touch. Because the packing is usually maintained at 0° F or higher, the handle or wheel is maintained at 33° F or higher. In some designs the extended bonnet has fins to absorb heat from the ambient environment or some external source. A limited amount of liquid is allowed to enter the space between the stem and extended bonnet so that it will vaporize and retard the rate of heat flow. However, care is taken that fluid not be trapped in the bonnet, since destructive pressures can arise in the vaporization process. The extended bonnet is usually constructed of stainless steel which, due to its low thermal conductivity and thin-walled construction, gives low heat transmission; a mounting plate may be welded to the upper portion of the bonnet for cold-box installations.

Stem threads are kept out of the liquid stream by the use of rising stem, outside screw and yokes for hand valves. Slip-stem designs are used with cylinder actuators. Some valves have a so-called broken stem consisting of radial bars on the upper stem which drive against axial pins on the lower stem. In this design, the entire valve is vacuum jacketed and the upper stem is retracted to eliminate contact and reduce heat transfer until valve actuation is required.

[1] Based on an article, "Control Valves for Cryogenic Fluids," by Chester S. Beard, Rocketdyne Division, North American Aviation, published in Control Engineering, vol. 13, no. 3, Mar. 1966, p. 67.

Conservation of the cryogen requires minimum heat transfer and as brief a chilldown time as possible. Therefore, cryogenic valve bodies have minimum metal in a small overall envelope. The valve wall must be thick enough to withstand the required pressure rating. The aerospace industry requires that the wall withstand four times the maximum working pressure before bursting. Fortunately, applications combining cryogenic temperatures and high pressure are not numerous, so the primary pressure vessels may have walls only fractions of an inch thick.

Insulation of valves for liquid hydrogen and helium service is critical in many cases. The use of plastiform insulating foams around the valve body markedly reduces heat loss. Styrene with its cellular structure is effective in this regard. The ideal solution would be to surround the valve with a Dewar flask. This principle is adopted in the design shown in figure 56 where vacuum jacketing surrounds the primary pressure vessel. The secondary shell forming the outer wall of the vacuum jacketing is of light construction to minimize the cooldown mass. Brightly finished interior surfaces may be used to reduce radiant transfer.

Valves for use in liquid-helium systems present especially severe problems. The stems are usually much longer to keep all heat out of the helium flow. When possible, the use of valves for liquid-helium systems should be minimized or avoided. In some instances, means other than valves are used to stop and start flow. Generally speaking, valves for liquid helium are not excessively troubled with heat-transfer problems when used in systems where extremely high flow rates occur. At Lewis Research Center, where very low flow rates of liquid helium are encountered, the heat absorbed by the valve stems can create critical problems. This type of problem can be solved by constructing a vacuum chamber or jacket to surround the valve. To prevent thermal radiation from entering the valve area, a radiation shield is then placed around the vacuum jacket. This shield consists of a second jacket enclosure. Then liquid nitrogen (−320.8° to −345.5° F) is circulated between the two jackets, to act as a thermal barrier between the liquid helium and the surrounding atmosphere. As a final precaution, a foamed insulation is installed around the liquid nitrogen jacket.

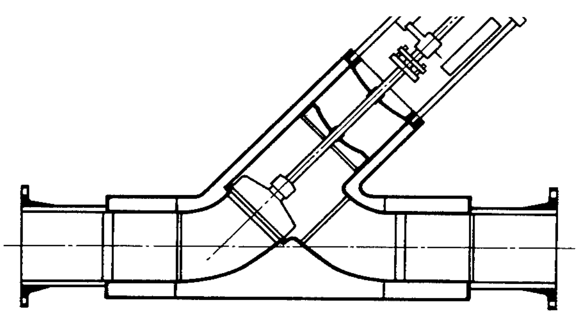


FIGURE 56.—Vacuum-jacketed cryogenic valve.

In this design, metal jacket walls are kept as thin as possible to minimize the area available for heat transfer paths.

The rules of thumb for valve insulation at the Lewis Research Center are:

(1) Liquid nitrogen and liquid oxygen: use insulation, preferably a foam type containing Freon bubbles, to utilize its superior insulating properties.

(2) Liquid hydrogen: use a vacuum jacket in addition to foam insulation.

(3) Liquid helium: surround the valve with a vacuum jacket, then use a thermal radiation shield containing another cryogenic fluid, and finally a foam insulation as an outside covering.

Austenitic stainless steels, aluminum, copper, ASTM B–61 and B–62 bronzes, and nickel have properties that make them useful in cryogenic valves. At low temperatures they increase in tensile, yield, and fatigue strengths and hardness, but they decrease in ductility. Carbon steel, a common material for noncryogenic service, cannot be used because its brittleness increases as temperature decreases. For pressure applications even at liquid helium temperatures, the austenitic stainless steels may be used. Aluminum is acceptable for modest pressure applications.

Ductility—the ability not to become brittle, but to deform before breaking—is an important consideration. Many materials lose their ductility at cryogenic temperatures. Ductility is closely related to crystalline structure; face-centered cubic lattice structures do not exhibit brittleness, while body-centered ones commonly fail due to brittleness. Proper heat treating or adequate alloying will often remedy the brittle condition, however, and alloys with high nickel content are very suitable for cryogenic service. Austenitic steels (particularly the 304 and 304 L varieties) are face centered. Iron is not face centered and, as a result, is not a good cryogenic metal. Aluminum alloys are better for cryogenic applications than pure aluminum. Another face-centered material is copper, which can be used even though an alloy may be preferred because of other favorable properties.

Trim parts such as plugs and stems are frequently made of stainless steel or monel. Teflon and Kel-F are commonly used for seats, although there is some question whether they retain sufficient plasticity at the low temperatures encountered to give lower leakage than properly designed metal-to-metal seals. Valves for fluorine often have copper seats. Bushings made of Ampco (copper with high aluminum and iron content) have been successful with stainless-steel stems; bushings are also made of Teflon or glass-impregnated Teflon.

Packing is normally virgin or filled Teflon, Kel-F, or Teflon-filled asbestos. In some applications, welded bellows are used instead of packings. Gaskets are of Teflon, metal-clad Teflon, or asbestos. Stainless-steel rings are used in ring-type joint flanges. Fluorine requires soft copper, aluminum 25, or stainless steel as gasketing.

Table XXVI summarizes the compatibility of various materials and cryogenic fluids. Although most metals are compatible with cryogenic fluids, some must be avoided because of the effects of very low temperature on their physical properties.

A few comments supplementary to the information of table XXVI may be helpful. Teflon or Kel-F are acceptable for most applications requiring nonmetals. LF_2 (liquid fluorine) is an exception; there are no plastics that can be used with flowing liquid fluorine. Spray-coated or calcined aluminum oxide is resistant to liquid fluorine at cryogenic temperatures. Dacrons, Mylar films, and nylon have been used with LOX.

Lubricants are not used with liquid hydrogen because the low temperature causes them to become brittle and solidify. There are also no suitable lubricants for fluorine service because it reacts with organic, aqueous, and siliceous materials. Beeswax may be a possible exception. Fluorolubes, which are not petroleum based, are used to lubricate liquid oxygen valving.

Key features of shutoff, ball, butterfly, solenoid, gate, and safety valves are illustrated in figures 57 through 64.

The soft-seating material used on the surface of the plug of the shutoff valve of figure 57 is representative. Note that a foam-insulating jacket and a polished body casting are used to reduce heat transfer. The contained gasket and opposed packing seal against both pressure and vacuum. A conventional type of pneumatic, hydraulic, or

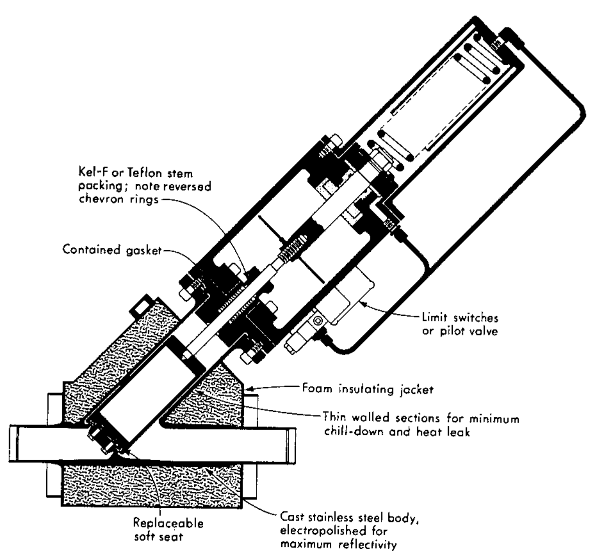


FIGURE 57.—Valve with a soft-seating surface.

TABLE XXVI.—*Valve Material Compatibility Guide*

Cryogenic liquids	Valve body and trim materials															Gasket, seal, and seat materials									Lubricants					Notes
	Aluminum	Aluminum alloys	Copper	Lead	Steel	Stainless steels					Mercury	Monel	Nickel	Tin	Pyrex	Teflon	Kel-F	Nylon	Polyethylene	Neoprene	Butyl	Asbestos	PVC	Buna-N	Fluorolube	Graphite	Silicone	Moly Kote, type 2 powder	Beeswax	
						303	304	316	321	347																				
Hydrogen	A	A	A	I	N	C	C	C	C	C		C	C			C	C	C		N		A	I			C				1
Methane	A	A	A		A	C	C	C	C	C		A	A			C	C													
Fluorine	C	A	A	A	A	A	C	A	A	N		C	C	A	A	A	A								I		I		A	2
Oxygen	N	N	C		N	C	C	C	C	C		C	N			C	C	I	A			C		N	C	C	C			3, 4, 5
Helium	C	C	C		C	C	C	C	C	C						C	C			C	C	C		C		C				
Nitrogen	N	N	C	A	N	C	C	C	C	C	A	C	N	A	A	C	C	A	C	N	N	A	A	N	A	A		A		6

Symbols:
- A—acceptable
- C—compatible
- I—incompatible
- N—conditional (see note)

Blank space represents unknown condition.

(1) Hydrogen temperature determines acceptability.

(2) 347 is crack sensitive. Aluminum or copper gaskets acceptable. Stainless steel sometimes used as seat material.

(3) Compatible with GOX only when moisture is less than 0.1 percent.

(4) Titanium presents explosion hazard.

(5) Buna-N impact sensitive.

(6) Nitrogen temperature determines acceptability.

electrical operator can be used for valve actuation.

The vacuum-insulating jacket illustrated in figure 58 encloses the entire valve body. An expansion bellows for attachment to a piping jacket is provided to prevent transmission of strains. The jacket may also be welded around the end connections, and be individually insulated. Air is sealed in the stem cylinder to achieve long, poorly conductive path, and a vapor space is provided between the cylinder and body extension. Thermal leakage is minimized by not having bolts or studs enter the valve body.

The butterfly valve of figure 59 has an eccentric disk and a special seal design which result in low leakage. The spherical seating disk gives maximum seating pressure at closure and minimum breakout torque. As is characteristic of valves for cryogenic applications, this unit has an extended neck. A low cooldown mass is achieved by the narrow body.

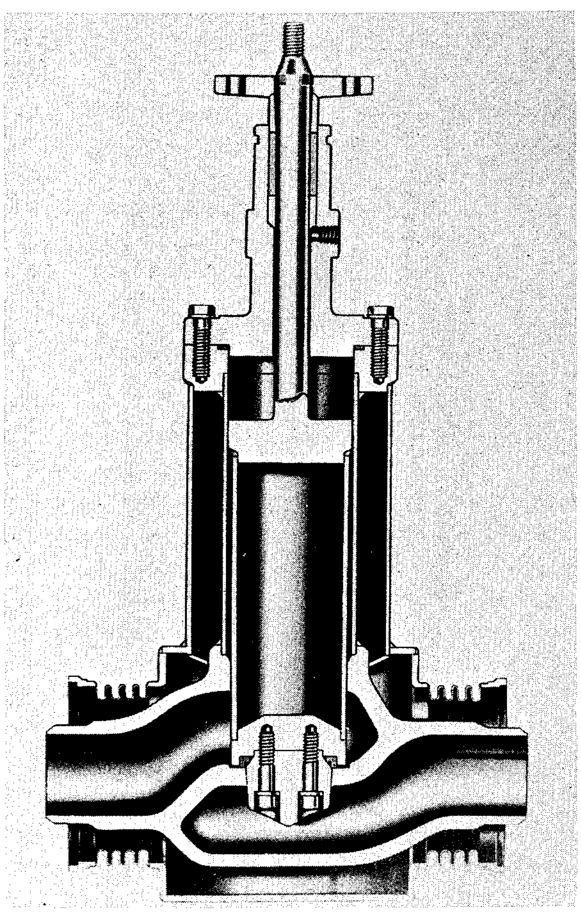
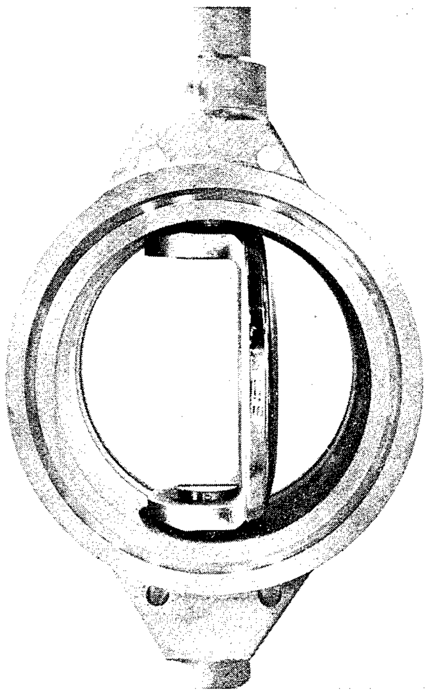


FIGURE 58.—Vacuum-jacket-insulated valve.

Safety or relief valves, as shown in figure 60, are necessary in lines or vessels holding cryogenic liquid because of the possible rapid evolution of gas upon loss of insulation, isolation by valve closure, or exposure to external fire. Most cryogenic valves developed for this service use the typical self-seating surface and an extension neck for isolating springs and seals from the flowing cryogen. The pilot-controlled unit of figure 60 can handle large coulometric flow rates under a wide variety of operating conditions. Pilot exhaust can be taken overboard to decrease the backpressure effect when the valve opens, and the pilot supply line can be taken directly to the point at which pressure control is desired. Chatter from excessive pressure drops into the piping to the main valve is thereby eliminated. A check valve in the pilot supply will preclude backflow if a vacuum is created while the protected vessel is emptied.

Packless solenoid valves work well in cryogenic service if consideration is given to material selection, cleaning, seals, and the fact that no lubrication is allowed. Coils should be encapsulated in an epoxy that will withstand the low temperatures. The in-line piloted valve of figure 61 in the 1½-inch size operates in 100 milliseconds. A spring-loaded push type armature moves the pilot poppet to permit line pressure to actuate the main valve poppet. The large, low-pressure ball valve shown in figure 62 uses metal stem seals to reduce the need for thermal protection. Ball valves can also be readily equipped with extension necks. The resilient ball seal compensates for displacement, distortion, and differential shrinkage. Ball, butterfly, Y-pattern, and gate valves provide the high-flow capacity required for transferring propellants and oxidizers in engine test and ground-support applications.



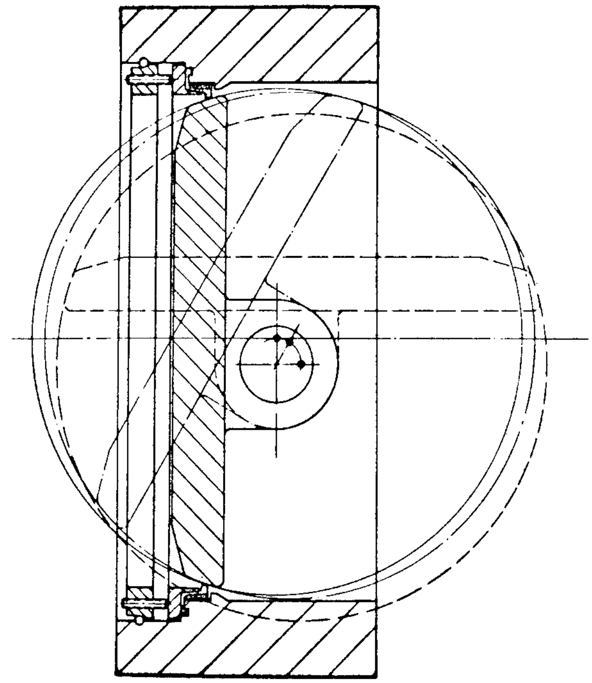


FIGURE 59.—Butterfly valve—eccentric disk.

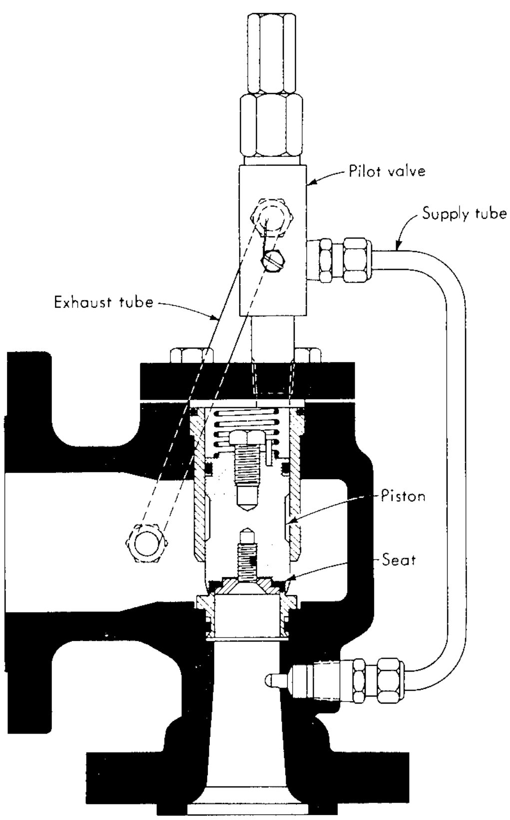


FIGURE 60.—Safety or relief valve.

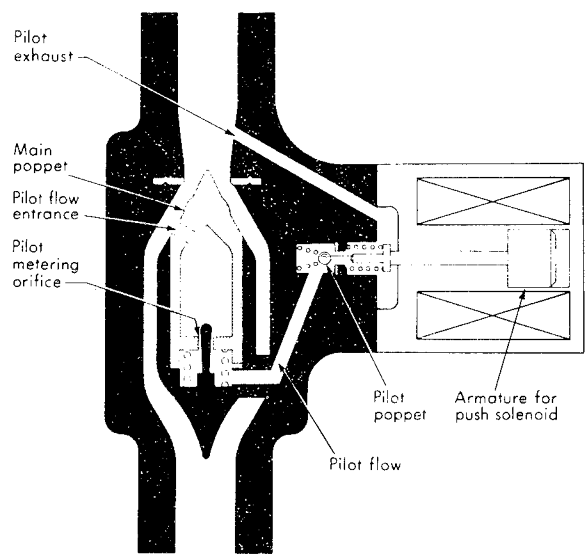


FIGURE 61.—Packless solenoid in-line valve.

A completely replaceable seat on the gate valve of figure 63 or a replaceable soft seating surface on the disk of a gate valve is common. Since a retained gasket is desirable, such valves usually have a round bonnet bolt circle for easy machining. Although gate

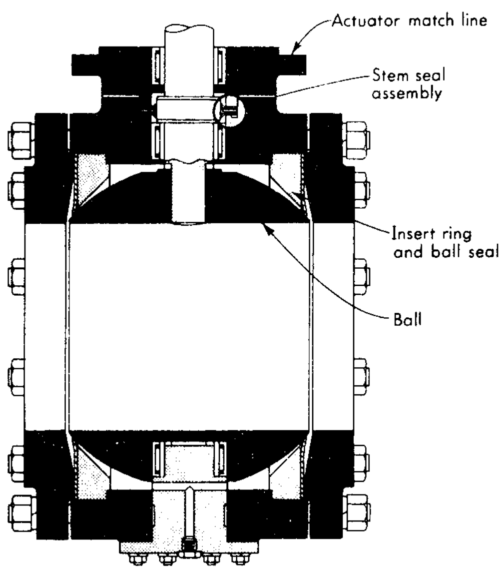


FIGURE 62.—Ball valve.

valves are often limited to pressures below about 1000 psig, their full-flow characteristic makes them popular in the aerospace industry.

The vent hole on the inlet end of the gate valve of figure 64 illustrates the provision necessary to avoid trapping liquid in the bonnet.

CONDITIONS OF USE

Danger from various forms of contamination necessitates strict cleaning requirements for cryogenic systems and components. The formation of solid oxygen in a liquid hydrogen system is dangerous because of the shock sensitivity of the cold equipment. Air or moisture is removed by purging with nitrogen or helium gas; also, equipment that will handle fluorine requires

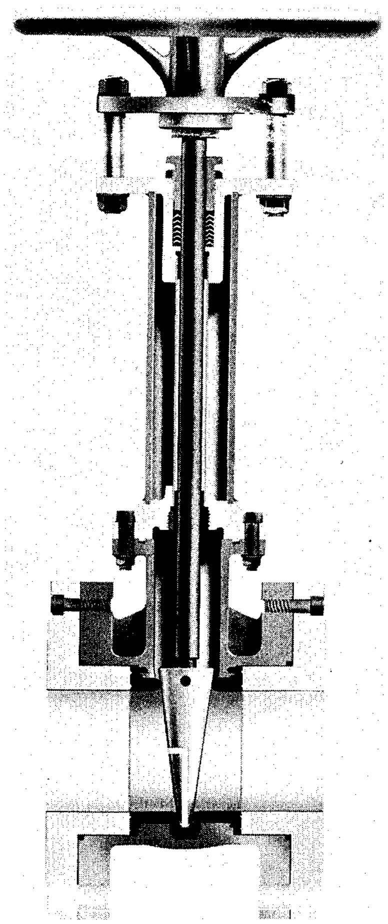


FIGURE 63.—Gate valve with replaceable seat.

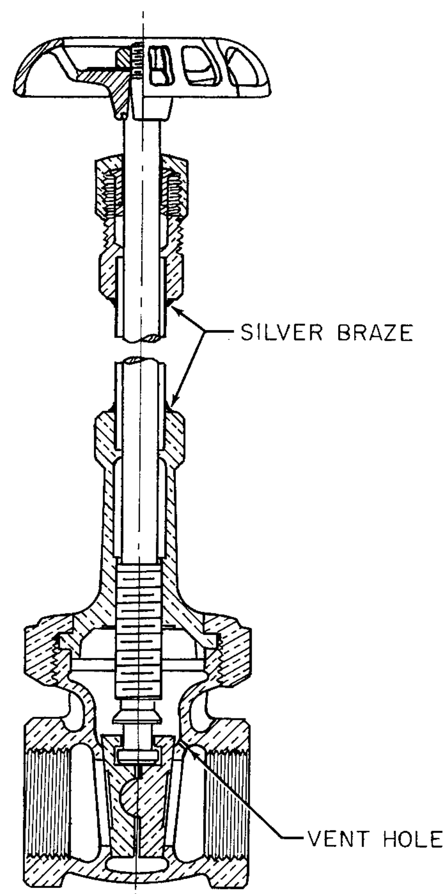


FIGURE 64.—Cryogenic gate valve.

vigorous cleaning, since fluorine is hypergolic with hydrocarbons (it burns upon contact, without ignition). Such systems must be passivated by using the fluid in the gaseous state before liquid is circulated. Standard cleaning techniques for cryogenic equipment are:

(1) Individual parts are cleaned before assembly. Usually, the completed unit is taken apart and cleaned again prior to final assembly.

(2) Parts are solvent degreased. A 4-percent detergent solution is used for stainless-steel and plastic items, while aluminum parts are similarly treated with a 4-percent solution of aluminum cleaner. Extensive rinsing is required after cleaning.

(3) Stainless-steel parts may also be acid pickled in 40 to 50 percent nitric acid for at least an hour, followed by a thorough water rinse and steaming.

(4) Nitrogen is used for drying.

(5) Various inspection methods are used after cleaning. Visible inspection and wiping with a clean white lint-free cloth can disclose contamination. Water droplet formation on a surface indicates an oil film. Black light causes some oils and grease to fluoresce, although this is not a positive test, since some unacceptable products do not fluoresce while other nonobjectionable substances become visible.

(6) Assembled devices or parts must be packaged in polyethylene bags after certification. The openings in large valves may be sealed with moisture proof materials.

A potential fire hazard exists in the use of valves for processing cryogenic fluids. Water vapors in the air freeze on these surfaces to form frost or snow. Then, because of the excellent insulating characteristics of the frost, extremely low temperatures occur at the base of the frost layer, allowing liquid air to form on the pipe or valve surfaces. This liquid air tends to wash off the frost. The liquid dripping from the now wet piping and valves contains both liquid nitrogen and liquid oxygen. Liquid nitrogen in the drippings vaporizes first and leaves essentially pure liquid oxygen, creating a hazard. To prevent this, insulation is placed around the valve and piping, even though it may not be required for any other reason.

To size cryogenic control valves, one should use the C_v (capacity factor) for the valve and the accepted flow formulas. However, one should be extremely careful in selecting the values that are substituted in the equations. The wide ranges of temperature and pressure that are encountered have significant effects on densities, gas-to-liquid ratios, and expansion factors, which can lead to erroneous answers.

Tables have been developed to show the evolution of gas in pipes and vessels under various conditions of heating from both an external source and from exposure to fire. Accepted heat flux values were used to determine these gas evolution tables; necessary parameters include exposed areas, condition of insulation or wetting, and the cryogenic fluid being handled. Such tables can be used to find relief requirements for sizing safety valves.

CHAPTER 12

Pressure-Surge Protection[1]

Pressure surges differ from overpressures in significant ways. Since these differences strongly influence the nature of devices for pressure-surge protection, we will discuss some of the salient features of pressure surges and the way in which they differ from overpressures.

Figure 65 shows a portion of a pressure enclosure. The enclosure may contain a pressurized fluid at rest, or flow may exist with a drop in total head in the direction of flow. From a practical standpoint, it makes little difference whether the pressure front is traveling through stationary or moving fluid. In either case, the initial condition is one of steady state. If a pressure rise occurs in a time interval which is long compared to the time required for a sound wave to traverse the enclosure, the pressure rise will be applied equally to all regions of the enclosure; overpressure protection, as supplied by a conventional relief valve, is required.

A pressure surge, on the other hand, behaves differently. The distinguishing feature of a pressure surge is the presence of a high-pressure region which moves rapidly through the pressure enclosure. A pressure surge originating to the left of the pressure enclosure shown in figure 65 would initially have an interface at position 1 between the high-pressure region and the lower, steady pressure. At successive later time intervals the interface between the high- and low-pressure regions would progress to position 2, and then to position 3. The velocity of progression of the pressure front would be approximately 4000 ft/sec if the fluid were water, and 200 ft/sec if the fluid were air. A pressure disturbance of large magnitude characteristically moves at a velocity greater than the sonic velocity of the medium. The pressure surge may have originated from a sudden closure of a valve, rapid strokes of a pump or actuator piston, a combustion process, or a rupture of the pressure enclosure which admits a much larger pressure from some exterior region.

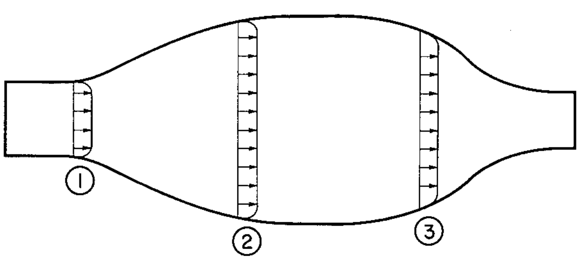


FIGURE 65.—Flow environment for ordinary and surge-pressure conditions.

In addition to the high velocity of the pressure front, several other characteristics make surge-pressure problems differ appreciably from the usual hydrostatic- or fluid-flow problems encountered by the valve designer. The high velocity produces large directional effects in the fluid. The dynamic pressure can be 1 order of magnitude larger than the static pressure. Momentum effects predominate. Events governing the destruction of the pressure enclosure are measured in small time intervals. If protection involves mechanical movement, the movement must be rapid. Accordingly, large activating forces and low-mass moving parts are necessary.

[1] Based on a paper, "Pressure Surge Protection," presented by J. R. Jedlicka, Research Scientist, Magnetoplasmadynamics Branch, Ames Research Center, NASA, Moffett Field, Calif., at the Valve Technology Seminar, Oct. 21–22, 1965, Midwest Research Institute, Kansas City, Mo.

SURGE-PROTECTION DEVICES

Surge protection devices can be categorized: (1) by geometric shape alone (no moving parts), (2) as valves actuated by surge pressure, and (3) as rapid controller-actuator systems.

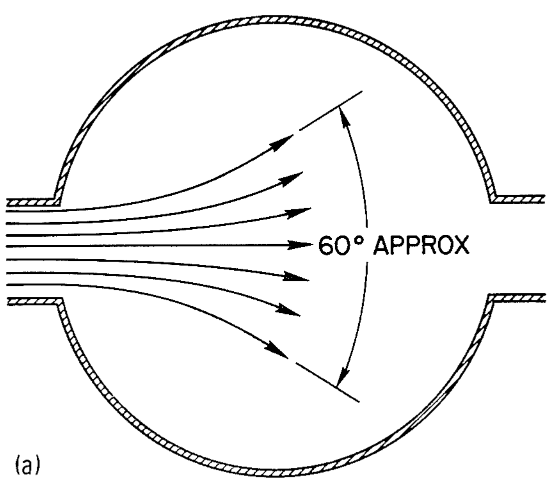
In a device with no moving mechanical parts, the problem of low moving mass and high actuating forces is eliminated. Many effective surge-protection devices have been designed to take advantage of the characteristic differences between pressure surges and conventional fluid flow. The common rupture disk is an example of a valve acuated by surge pressure. Finally, some interesting rapid controller-actuator systems have been developed in which the pressure surge is detected by a pressure sensor or, in some cases, a temperature or electrical conductivity sensor, which may be located upstream from the valve.

GEOMETRIC SHAPE ALONE

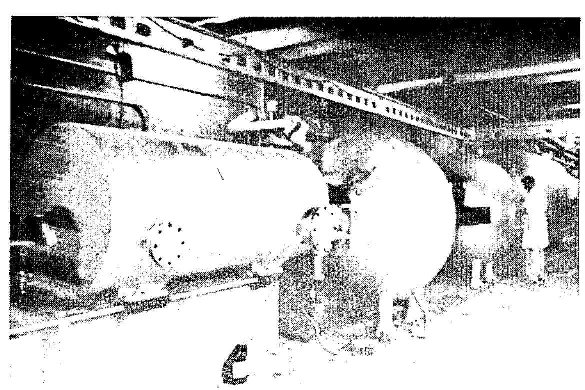
In surge-protection devices, advantage can be taken of the predominant directional effects of the pressure surge. Figure 66(a) shows a surge tank in which the pressure front expands at an angle such as that shown (the maximum angle depends on the fluid and its energy content per unit mass). The expansion of the pressure front is accompanied by a corresponding reduction in the magnitude of the surge pressure. The surge which leaves the expansion tank is, therefore, much lower in magnitude than that which entered. If the ordinary flow condition does not preclude its use, a baffle installed in the expansion tank is effective.

A multiple expansion tank installation at Ames Research Center is shown in figure 66(b). A large-size gun launches free-flight aerodynamic models into a long, partially evacuated tube containing instrumentation. If provision were not made to suppress the pressure surge resulting from the muzzle blast from the model-launching gun, the instrumentation in the tube, and perhaps the tube itself, would be destroyed. The gun is on the far left of figure 66(b). The cylinder and the two spheres in the central portion of the picture comprise three expansion tanks in series. These are made of heavy-wall material and two are spherical to withstand the large surge pressures encountered from the muzzle blast of the gun. The pressure surge is reduced to an acceptable value after passing through the three tanks and then enters the tube on the far right.

A similar problem is shown in figure 67. The flight performance of a gun-launched model was appreciably degraded because the model was immersed in rapidly moving



(a)



(b)

FIGURE 66.—Expansion tanks. (a) Pressure front expansion; (b) Triple expansion tanks on hyperballistic range.

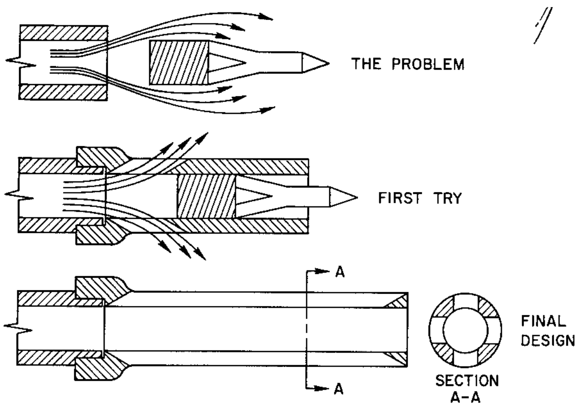


FIGURE 67.—Elimination of muzzle blast from model launching gun.

combustion products as it left the muzzle of the launch gun. While much progress is being made in flying aircraft perpendicular to their normal direction of flight (VTOL aircraft), the aerodynamicist has not learned how to make them fly well when they are moving rapidly in the reverse direction. The problem, in this case, was to remove the muzzle-blast gases so that the aerodynamic model would effectively fly in its forward direction throughout its entire flight. The first try at solving the problem is shown. An extension having large ports was added to the gun muzzle. The port cross-sectional area was made approximately four times larger than the cross-sectional area of the gun barrel. This configuration was expected to be equivalent to a Y-pattern valve with an exceptionally high flow coefficient. The design proved embarrassing. Only a small portion of the exhausting gases passed through the ports. In fact, the model increased in velocity by several hundred feet per second with the addition of the barrel extension and, moreover, was still engulfed by the combustion gases as it exited from the barrel extension. The final design is shown next. The barrel extension was made longer and the port area was increased considerably. In fact, what formerly might have been considered a barrel extension became more equivalent to four guiding rails. The final design worked reasonably well. This illustrates to the valve designer the futility of considering pressure-surge problems as if they were reasonably well represented by conventional fluid flow.

Figure 68 is concerned with the specific pressure-surge problem of a pipeline explosion caused and perpetuated by combustion within the pipeline. Combustion persists as a consequence of the high temperature and pressure which accompany the moving pressure front. If either the temperature or the pressure is sharply reduced, the combustion reaction may be stopped. Figure 68 shows a device intended to work on the principle of extracting thermal energy from the pressure front, with a consequent reduction of temperature in the high-pressure region below that which would sustain combustion. It consists of a section of pipe about 50 diameters in length, which is tightly filled with a bundle of small-diameter copper tubing. This pipe section is made of larger diameter than the remainder of the pipeline to make its pressure drop for ordinary flow acceptably low.

The principle involved is demonstrated in the high school physics experiment in which a metal rod is tightly wrapped with a layer of paper; when the paper-covered rod is held in a flame, the paper does not even scorch because the metal keeps it below its combustion temperature. Such a device is installed at Ames Research Center on a 15 000-psi compressed air line just downstream from the air compressor, since the compressor is oil lubricated and is the most likely source of an explosion. The device has been in operation, but there has been no evidence of either an explosion or an arrested explosion, so it is as yet unproved. A porous-

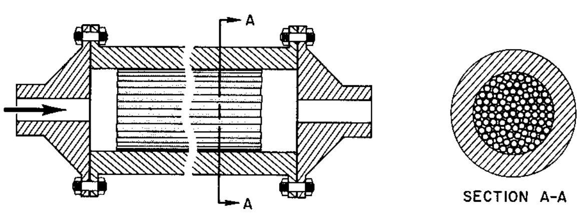


FIGURE 68.—Pipeline explosion arrester (thermal type).

bronze plug is sometimes used as a flame arrester and uses this same principle of operation. The plug is installed in a tubing fitting to prevent flame from leaving the case of an electrically operated hydraulic pump for control valve operation.

VALVES ACTUATED BY SURGE PRESSURE

Three examples of surge-pressure protection in the first category have been described, two of these associated with a gun-muzzle-blast problem. An appropriate way to introduce the second category—valves actuated by the surge pressure itself—is another muzzle-blast problem. A facility at Ames Research Center faced the problem of preventing gun-muzzle blast from entering an instrumented container. Expansion tanks as shown in figure 66 could not be used because of a space limitation; the guide rails shown in figure 67 could not be used because the gun muzzle and the instrumented container were required to be held at a low pressure (several millimeters of mercury absolute pressure) prior to model launch.

As can be seen in figure 69, the protection device consisted of a length of the largest diameter pipe that could be conveniently installed between the model launching gun and the instrumented tube. The pipe was perforated with a multitude of holes so that it would appear to the blast wave almost as if it were a screen, and hence vent the pressure surge. In preparation for the experiment, the pipe was covered with a plastic film about 1/100 of an inch thick; i.e., it was of sufficient strength to withstand the nominal 1-atmosphere pressure inward when the system was evacuated, but also worked as rapid-acting multiple rupture disks during the pressure-surge part of the cycle.

The pressure-surge device shown in figure 69 is an old idea. The effluent from an internal combustion engine consists of a steady flow on which are superimposed periodic pressure surges. A long perforated tube was enclosed in a larger outer pressure shell, and was furnished as original equipment called a "muffler" on Henry Ford's Model A's.

In the discussion of figure 68, the thermal pipeline explosion arrester, we noted that a large reduction of either temperature or pressure should arrest the propagating pressure front. Figure 70 illustrates how

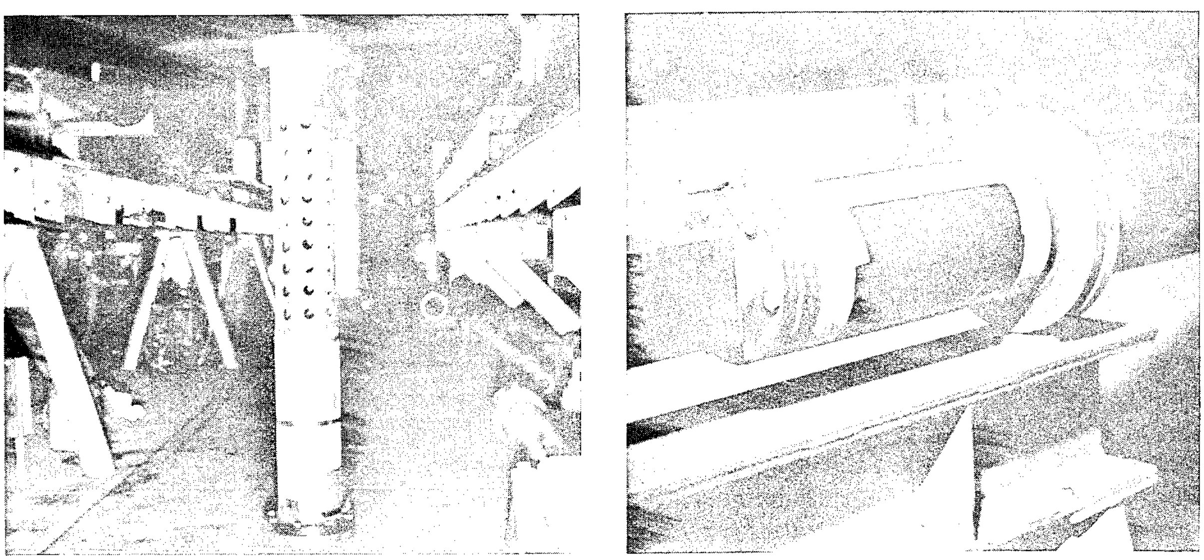


FIGURE 69.—Surge-pressure vent valve on gun tunnel.

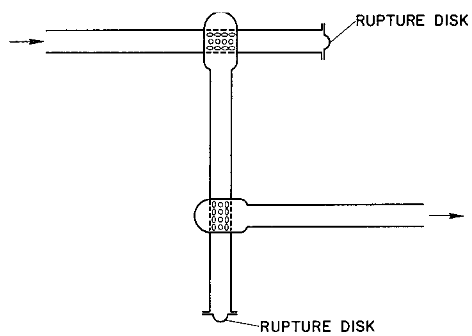


FIGURE 70.—Pipeline explosion arrester-pressure relief type.

this might be accomplished by reducing pressure rather than temperature. The pressure surge has a dynamic pressure which may be an order of magnitude larger than its static pressure; the perforated side outlet is presumed to pass only static pressure; the closed end has a rupture disk which would prevent pressure-front reflection with its corresponding approximately twofold augmentation of pressures and temperatures. Figure 70 shows a pair of side outlet and rupture disk combinations in series for additional insurance. This scheme apparently is conceptual and has not been tried.

The valve designer is well aware of the general failure of a rupture disk to possess a sharply defined burst pressure. For some conditions of physical size and pressure, the testing of a large number of apparently identical rupture disks will demonstrate that most diaphragms will burst within ±10 percent of a mean value of pressure. An occasional one will burst considerably under or over the 10-percent values. Careful design of the clamped edges and careful installation technique are imperative, but another approach to the problem of nonrepeatability of burst pressure, originated at Langley Research Center, is shown in figure 71. Here a 100-psi system must vent reliably if the pressure ever attains 105 psi, or only 5 percent above the ordinary working pressure. The technique consists of utilizing two rupture disks. Each disk is ordinarily held to 50 psi by a venting pressure regulator. Commercial relief valves which will open reliably at 105 psi are readily available; such a valve, sized to overwhelm the 50-psi pressure regulator, is installed as shown. Under ordinary conditions, each rupture diaphragm has a net pressure of only 50 psi across it, and hence should sustain this with little tendency to blow. Should the main system attain 105 psi, the relief valve would open and fill the cavity between the rupture diaphragms with fluid at 105 psi. This action would load diaphragm no. 2 to 105 psi, sufficiently in excess of its 75-psi burst rating, so that it would burst at once. As soon as diaphragm no. 2 burst, diaphragm no. 1 would have the full 105 psi across it. Hence it would burst reliably and vent the whole system.

Another device designed to overcome the undesirable large spread in the burst pressure of ruptured diaphragms is shown in figure 72. Personnel at Ames Research Center believed that much of the spread in the burst pressure of diaphragms is associated

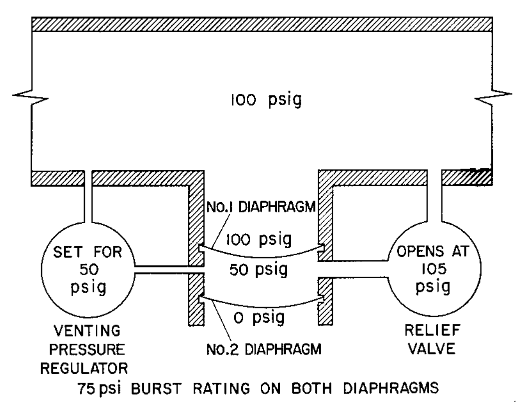


FIGURE 71.—Accurate, high-flow, overpressure relief.

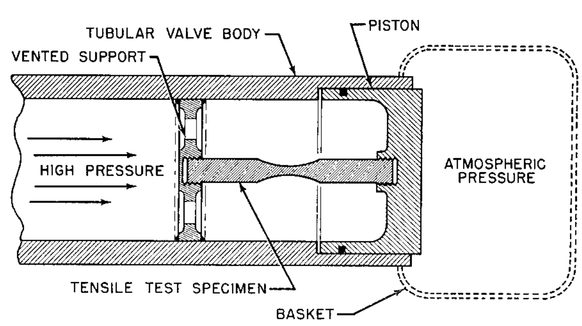


FIGURE 72.—Accurate release with high flow rates.

with the degree of clamping at the diaphragm edges. The clamping is likely to be nonrepeatable, for a number of reasons. Techniques for fabricating repeatable tensile test specimens, on the other hand, are well recognized. Moreover, the correct size to provide the desired force in a tensile-test specimen may be determined easily by use of a standard strength of materials-testing machine. Accordingly, a tensile-test specimen was utilized to restrain a piston in the particular application. Since rapid response was desired, the piston was designed to have a minimum mass. With the tensile-test specimen installed to minimize bending stresses at its ends, the technique for this being well developed in strength of materials-testing machines, scatter of only a few percent around the mean value was found to exist. Opening time, moreover, approached that of a rupture disk.

One disadvantage should be pointed out: A corrosive environment alters the burst pressure of a rupture disk in the safe (reduced burst pressure) direction. The opposite is true for a piston-cylinder combination.

The device shown in figure 73 can be considered a surge-pressure device only in that its initiation constitutes a pressure surge. But in addition to providing protection for the initial pressure surge, it must provide protection for continuous venting of high-pressure fluid. The sketch on the right of figure 73 describes the system and its problem. The large pipe is a portion of a 6600-psi system in which the pressure is reduced to 100 psi for transmission of a fluid through a heat-exchanger loop rated nominally at 100 psi. After sufficient heat exchange has occurred, the pair of gate valves shown in the left sketch are closed and the large pipe is then returned to 6600 psi. The possibility exists of a malfunction of one of the gate valves which would charge the heat-exchanger loop to the 6600 psi and thus present a potential surge and overpressure condition. The solution to the problem at NASA Langley Research Center is shown on the right of figure 73. A gate-valve malfunction would result in the rupture of the 150-psi rupture disk and, assuming a rather large leak through the gate valve, supersonic flow would be established in the Laval nozzle. Under these conditions, at the Laval nozzle exit of diameter D the static pressure of the effluent jet would be limited to a nominal value of 100 psi, thus providing the required protection for the 100-psi system. The point to be emphasized is that if the Laval nozzle were replaced with a pipe of constant diameter, the pipe-exit pressure would be about 3400 psi corresponding to choked flow. This pressure would be felt by the 100-psi system and would cause its failure. The plumbing shown constitutes an ejector. The design calculations are simple enough to be shown in their entirety in figure 73.

The device shown in figure 74 represents

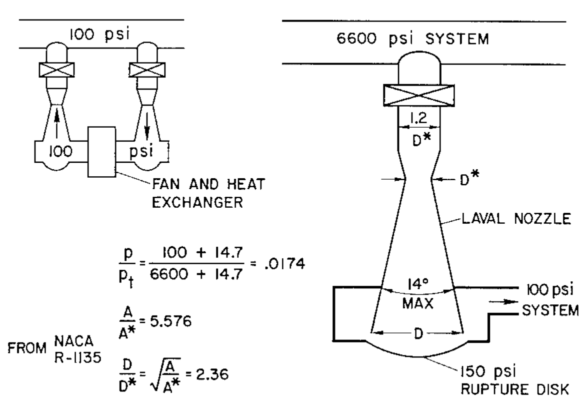


FIGURE 73.—Low-pressure system protection.

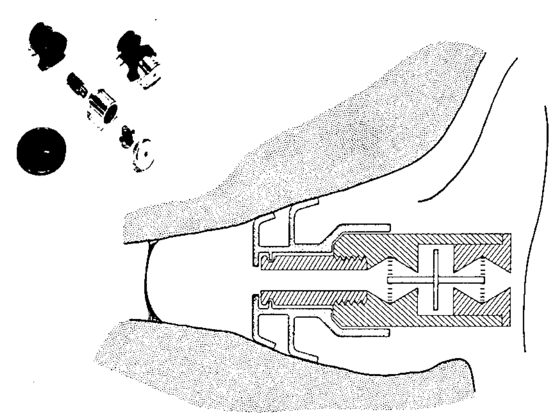


FIGURE 74.—Commercial ear surge-protection valve.

an unusual application of a valve actuated by a pressure surge. The purpose was to protect the human eardrum from rupture by a pressure surge. From the valve designer's standpoint, the eardrum can be considered a rupture disk, and the design problem consisted of designing a surge-operated valve with sufficiently rapid response to protect a ruptured disk. Success hinged on minimizing the moving mass of the valve and maximizing the actuating force.

The actuating force of a pressure surge acts on a circular piston approximately one-fourth of an inch in diameter. The backside of the piston, of course, forms the valve seat. The fact that the actuating piston has large side clearances does not detract from its operation under surge conditions, again illustrating that conventional fluid flow differs considerably from pressure surges. The mass of the piston valve seat and stem combination is about 30 milligrams—so small that if taken out and placed on a desk, a sneeze is likely to blow it out of sight.

The real question, of course, is, does it meet its design goals? No volunteers were found at Ames Research Center for a test of the device under rupture conditions. Instead, it was tested with a pressure surge large enough to make the unprotected ears ring. Ordinary sound levels were attenuated, as might be expected, since ordinary sound consists of weak pressure surges. The ear-ringing sound intensity was attenuated to a similar degree, which suggests that the device was not functioning as intended. The question of its operation under rupture conditions remains unanswered. A laboratory test could be set up easily if data were available on the burst pressure of the human eardrum. However, edge support and thickness may differ among individuals, and such variations would produce the large scatter typical of rupture disks and, therefore, complicate the problem.

Three different valves, designed and constructed by the Mosler Safe Co. to protect installations from airblast waves generated by nuclear explosions, were tested at the U.S. Army Engineer Waterways Experiment Station (ref. 1). These valves were a blast-actuated tubular type; a blast-actuated disk type; and a remote-controlled disk type. The first valve was designed to withstand a 100-psi blast pressure. It consists of two sets of 0.5-inch-diameter stainless-steel dowels, one set being movable and the other set stationary. Under the impulse of a shock wave, the movable dowels move against light springs, and seal when meshed with the stationary set. The valve weighs approximately 200 pounds and has overall dimensions of 18.125 by 15 by 7.25 inches. The design airflow is 550 ft^3/min, with a static pressure loss of 1 inch of water. The valve was designed to close at an air overpressure of 2 psi. For pressures ranging from 3 to 18 psi, the valve closing time is between 4 and 9 milliseconds.

The second valve was also designed to withstand 100 psi. It weighs about 200 pounds and is 16 inches in diameter and 36 inches long. It was designed for an airflow of 2750 ft^3/min, with a static pressure loss of 1 inch of water. The valve was designed to close when subjected to an air overpressure of 0.07 psi and has two major parts: the valve closure disk and the valve seat. The aluminum disk was designed to move 4 inches and slam shut against the steel seat with no gasket between the valve disk and seat. This valve is not effective in sealing off an airblast; it does not completely close when subjected to approximately 7-psi overpressure, but closes rather rapidly at the 70-psi level.

The third valve tested was a remote-controlled, pressure-operated, cylindrical, 36-inch diameter, disk-type valve designed to resist 100-psi blast pressure. This valve was designed to close upon command from a remotely located sensor which might be sensitive to pressure, thermal radiation, or nuclear radiation from a nuclear detonation. The valve weighs approximately 350 pounds and is 36 inches in diameter and 48 inches long. It was designed for an airflow of 13 000 ft^3/min with a static pressure loss through the valve of 0.5 inch of water. A 250-psi air supply and a 110-volt electrical source are required to operate this valve. In a series of 24 opening and closing cycles, the closing time averaged 73.1 milliseconds and the opening time averaged 1.24 seconds. No leakage was evident when the valve was subjected to pressures up to 80 psi, and no structural damage was observed.

A final example of a pressure-actuated valve is shown in figure 75; three such valves are installed at Ames Research Center. Circulating cooling water at 60 psi normally flows through this valve. A failure of a heat exchanger could subject the cooling water to a pressure of 2000 psi. Protection is accomplished by closure of the valve. The valve differs from an ordinary two-way spool valve in that the area of the spool at the location of the left O-ring is made larger than the area of the spool at the right O-ring. Hence, a net force toward the left is applied to the spool body by the fluid pressure. The spool body is prevented from moving under ordinary conditions by a shear pin. Should the pressure rise appreciably in excess of 60 psi, the pin shears and the valve is driven to the closed position by the pressure forces exerted on it. To offer full protection for the 60-psi cooling water system, a pneumatic surge tank should be installed just downstream of the valve to dissipate the pressure surge which would otherwise be transmitted through the valve during the time of its acceleration toward its closed position. Also, corrosion or deposits from the water could have a deleterious effect on the valve's reliability.

RAPID CONTROLLER-ACTUATOR SYSTEMS

The photograph on the left of figure 76 shows a section view of an explosive-actuated crushed tube valve. The tube, in this case a Laval nozzle, was fabricated from copper. A high-explosive charge was placed on the actuator shown (a kind of spherically ended punch press die) and detonated from a signal originating from a controller. Gas-tight closure is accomplished in several milliseconds at an instant of time prescribed to a fraction of a millisecond. This valve should find wide application outside NASA. It might be used as an economical ultimate protective closure for petroleum or natural gas wells, as illustrated in the sketch on the right of figure 76. In this example a loose-fitting ductal sleeve would be dropped over the well casing during the early stages of drilling and submerged sufficiently below ground level to protect it in event of gas release and fire. High explosive could be attached to the actuator and the wires run through suitable conduit to a battery box sufficiently remote so that it could be actuated in event of a natural-gas fire. Partial closure could be effected even in the event

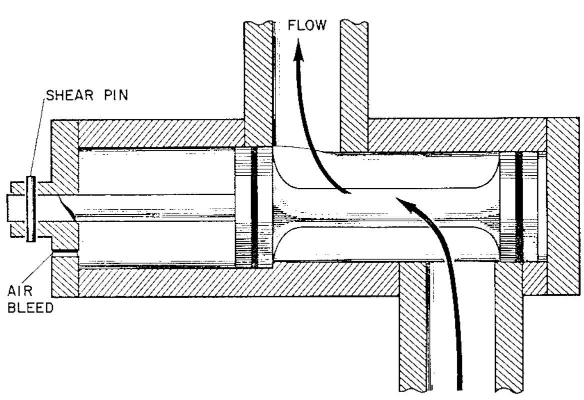


FIGURE 75.—Overpressure shutoff valve.

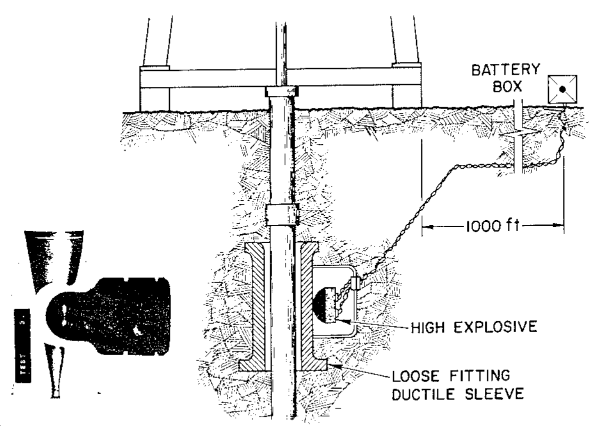


FIGURE 76.—Explosive-actuated crushed tube valve.

that the casing contained drill rod, a characteristic not possessed by other forms of valves.

The valve shown in figure 76 provided effective closure in a sufficiently precise and short-enough duration of time. Its only disadvantage is that it can be used only once. At Ames Research Center a new design has been developed, however, which possesses all the desirable characteristics of the valve of figure 76 and, in addition, is reusable. This Majeski valve is shown in figure 77. Basically, it consists of a two-way spool valve with an integral actuating piston, the force on which is supplied by a high explosive charge. The high explosive charge used in both figures 76 and 77 is a versatile and economical approach to the problem. Air or other gas compressed to pressures high enough to give short actuating times would be difficult to contain, valve, and actuate from an electrical signal. The high explosive and its detonator cost about $5 and can be actuated by a transistor or a 6-volt battery.

High explosive can be utilized to overcome the burst-pressure spread deficiency previously ascribed to the rupture disk. By locating a small high explosive charge on the rupture disk, precise actuation with respect to both pressure and time can be accomplished. Figure 78 shows a typical explosive-actuated rupture disk, which can be sized to fail of its own accord at a pressure far in excess of the maximum surge pressure expected in the system. Upon a signal from a suitable controller (which in this case may be a temperature, pressure, flame, or other sensor), the rupture disk can be made to open upon desired command. Figure 79 shows a section of steel plate on which various failure patterns have been made with suitably applied high explosive. This technique should find considerable industrial application where emergency venting or pressure relief is desired, and where the probability of occurrence is too small to justify installation of a more complicated and more costly valve, or in which relief of surge pressure can be attained only by anticipating its need at an upstream station sensor and actuating a downstream device

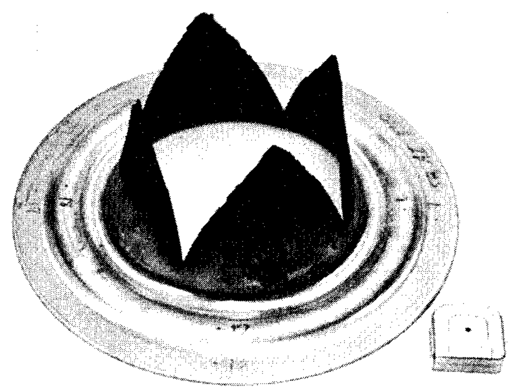


FIGURE 78.—Explosive-actuated rupture disk. (Material: ⅛ stainless sheet.)

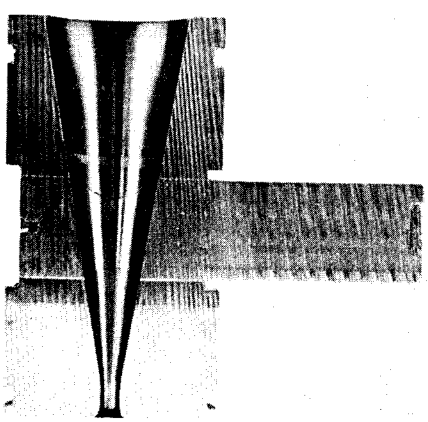


FIGURE 77.—Majeski valve.

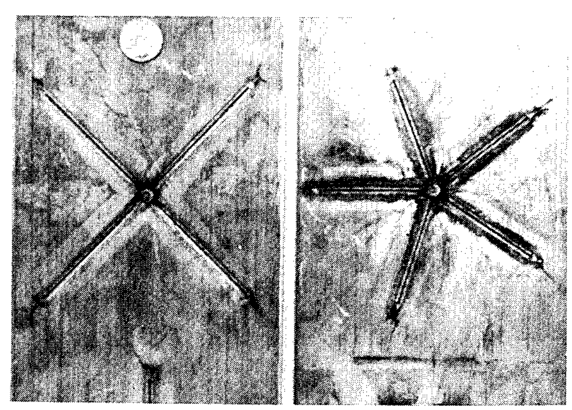


FIGURE 79.—Remote actuated rupture sections. (Material: ⅛ stainless sheet.)

prior to the arrival of the pressure surge. Also, if a system has a cyclic pressure variation, rupture disks tend to fatigue and leak. Similarly, under vacuum conditions, sealing is difficult with a rupture disk. A suitably located high explosive together with an appropriate sensor could overcome these deficiencies.

It is possible to procure the advantages of a remote actuated rupture disk without resorting to high explosives. At Langley Research Center, a magnetically actuated rupture disk has been successfully demonstrated, and development work is continuing on the technique shown schematically in figure 80. In the Langley device, energy is delivered from a momentary power source (a charged capacitor) and is caused to travel through a one-turn loop of wire which surrounds the rupture disk. Electrical currents induced in the rupture disk ultimately result in its mechanical failure. If the large amount of energy available from a high explosive is to be obtained from some other source such as capacitive storage, a large and expensive power source is necessary and this may be as hazardous as working with a high explosive.

Antiblast valves for protection of equipment and personnel of installations hardened against the effects of nuclear blast have been developed by Arthur D. Little, Inc. (ref. 2). These valves close in a few milliseconds, and may be described as disk-type valves. If the valves are in a protected structure and are provided with air-cylinder mechanisms, a source of compressed air, solenoid valves, and a triggering circuit

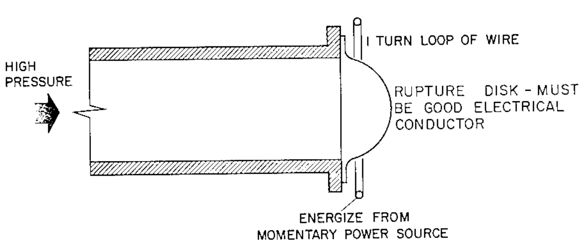


FIGURE 80.—Magnetically actuated rupture disk.

with appropriate remote sensing devices, closure can be effected before arrival of the shock from an explosion occurring 200 feet or more from the structure. Alternatively, if the trigger circuit should fail, the valves would be closed by the incident blast overpressures.

Many of the devices described in this chapter were developed as a consequence of deliberate efforts to produce and use surge pressures in the laboratory. The magnitude of surge overpressure which a structure or device can withstand has not been mentioned. A study conducted by the U.S. Naval Civil Engineering Laboratory on the blast resistance of some standard valves (ref. 3) is of interest in this regard. Commercially available 3-inch, 200-psi WOG (water, oil, or gas) bronze check and gate valves were subjected to transient overpressures of about 390 psi with air and 2000 psi with water. Subsequent visual examination, operational tests, and leakage tests revealed no damage to the valves and the pressure-surge test data indicated relatively low magnitudes of strain. The transient pressure rise time was about 1 millisecond compared to a natural period of the valves of about 0.1 millisecond. Therefore, the valves were not dynamically loaded. Although dynamic loading was not accomplished and the conditions of use must be considered before blast-resistance recommendations can be made, it was concluded that standard check and gate valves can withstand transient pressures far in excess of their rated capacity.

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3. BOCKMAN, K. R.; KING, J. C.; AND CHAPLER, R. S.: Blast Resistance of Check and Gate Valves. Report No. N–756, U.S. Naval Civil Engineering Lab., Port Hueneme, Calif., Nov. 1965.

CHAPTER 13

Unusual Designs and Applications

Unique valve and sealing designs have resulted from development programs within NASA. They represent solutions to problems that touch many areas of commercial valve design. As far as was practicable, these designs are grouped here in categories which characterize their salient features.

PACKINGS

Packing materials, as such, are not normally used in valves that must exhibit low leakage. Valve stems are sealed with Chevron or O-ring seals, or some combination of both.

In several specialized packing applications, the leaking properties of synthetic-rubber O-rings were found to be superior to those of plastic O-rings. Synthetic rubbers, however, often require lubrication and are frequently incompatible with the process fluid. A combination of the plastic and synthetic rubber O-rings was found to be of great utility in several applications. Plastic O-rings were used to contact the contained fluid to solve the compatibility problem. A synthetic rubber O-ring, with its superior sealing qualities, was installed behind the plastic O-ring. The space between the two O-rings was packed with a grease to further protect the synthetic rubber O-ring from the fluid. The grease also provides the lubrication necessary for the synthetic-rubber O-rings. This concept can be expanded by using a series of plastic O-rings followed by a series of synthetic-rubber O-rings with grease packing between all rings.

Marshall Space Flight Center has replaced many O-ring packings with Teflon-coated K seals. The K seal requires a smoother sealing surface than the O-ring, but a source of wear particles is eliminated. Further, elastomers such as rubber O-rings which have a limited life because of expiring cure dates are no longer a maintenance problem.

SEATS AND SEALS

Soft seats usually have lower leakage than metal-to-metal seats, particularly if repeated closures are unavoidable. Teflon and Kel-F are generally considered good seat materials, but some manufacturers recommend that Teflon not be used for pressure differentials above 200 psi. Above this range, Kel-F should be used. Although temperature is a selection factor (it is not certain whether Teflon or Kel-F is the better material at liquid oxygen temperatures), pressure should be the primary selection factor. In many cases, operating temperatures and process fluids do not permit the use of soft seats. Metal-to-metal seats are then necessary.

Metal-to-metal seats can be produced to give low leakage. The surface finish and dimensional accuracy of the seats must be carefully controlled. Unfortunately, contamination can quickly raise the leakage of even a carefully constructed valve to an unacceptably high level. Squib-operated valves and burst diaphragms give essentially zero leakage, but they are normally limited to a single actuation (squib units with six cycles of operation are available). Therefore, new sealing methods are being developed and old ones are being refined. The metal-to-metal seating study by Tellier at North American Aviation, which was reported in chapter 5, is an example of work in the latter area.

Wet Seals

Studies of "wet seals" are being conducted at TRW Systems. The goal is to retain the advantages of metal seating, while achieving the low leakage of soft seating, and preventing cold welding. The approach is to introduce a liquid metal interface between the mating metal parts to fill leakage paths, and to permit adequate sealing at seat stresses considerably below the yield strength of the material.

Tests were conducted to determine the materials that can be wet by liquid metals. The liquid metals used in this program were a gallium—13 percent tin alloy (40° F freezing temperature) and a mercury alloy with minor constituents of thallium and indium (−42° F freezing temperature).

The sealing capabilities of liquid metals applied to highly polished seat-poppet materials were tested in 1963–64. Helium leakage of 10^{-9} atm cc/sec at pressure differentials in excess of 2000 psi was obtained with static seals (ref. 1). The following conclusions were drawn from the work accomplished up to 1964:

(1) Finish at the sealing surfaces is very important. If nominal state-of-the-art finishes of less than 1 microinch can be maintained, both theory and static test results show that pressure differentials in excess of 2000 psi can be sustained in an essentially leak-free condition.

(2) For both static and dynamic seals, compatibilities between the liquid metal and the sealing surface must be exceptionally good to maintain long-term sealing ability. Tantalum, tungsten, or their alloys appear to be the most promising seat materials.

(3) The limited observations to date indicate that seal separation, as required by poppet action, is detrimental to seal effectiveness. This may be overcome through the use of porous media or other reservoir techniques to provide replenishment of liquid metal to the sealing surfaces. A valve using rotational motion of the primary seal surface, where the majority of the sealing area remains in contact, should not suffer loss of sealing properties in the liquid metal film.

(4) Long-term materials compatibility, possible enhancement of diffusion bonding, and long-term stability of the seal against pressure are potential problems.

A "zero-leak" rotational seal experiment, using burnished tungsten-tungsten seats lubricated with a liquid metal, was conducted. The results are stated to be satisfactory. However, breakaway friction increased after a significant amount of rotational motion; the device became squeaky. The liquid metal lubrication failed, permitting tungsten-tungsten contact to occur.

Wettability tests were performed in 1964–65 (ref. 2) on nonmetallic materials using a gallium-indium-tin liquid metal. The nonmetallics were high-density graphite, fused alumina (Al_2O_3), boron nitride, zirconium oxide, and magnesium silicate. Of these nonmetallic seal bearing materials, only Al_2O_3 is considered to warrant further study.

These rigidized polymer films were also considered for application to "wet seals" by TRW Systems (ref. 2). The polymer film is placed on a substrate metal and lies between it and the liquid metal. The thin polymer film shows promise of eliminating the long-term compatibility problem of loss of surface finish. Possibly the thin film will also reduce solution banding during lengthy static contacts of sealing surfaces.

The rigid polymer films were produced by electron bombardment of organic monomer vapors in vacuum. Dow-Corning silicone oil DC–705 and Convoil 20 were used to produce films on such substrates as beryllium-copper, steel, tungsten carbide, 440° C stainless steel, and 6061–T6 aluminum. While very low leakage was observed in seal tests with these material combinations and liquid gallium, attack of the surface finish of the seals after several days was noted. Other thin-film materials, including a mixture of gallium and powdered tungsten, also were tried. It appears that radiation polymerized films offer benefits in "wet seals," but additional techniques for film formation must be developed.

Labyrinth Seal

The cone labyrinth valve developed by the Smirra Development Co., Los Angeles, Calif., employs a novel concept of seat-surface contact. This concept promises improvements in contamination-insensitive, leaktight sealing, coupled with an unusual method for throttling many propellants. Figure 81 illustrates a prototype version of this device. The flow-control element of the valve accomplishes both throttling and shutoff by the use of two concentric sets of flexible metal blades. Throttling is effected by forcing capillary flow through the labyrinth created by valve closure; the sealing action results from the engagement of the resilient metallic sliding surfaces. The concentric blades approach each other with an intermeshing action which tends to be self-adjusting and which finally provides shearing contact.

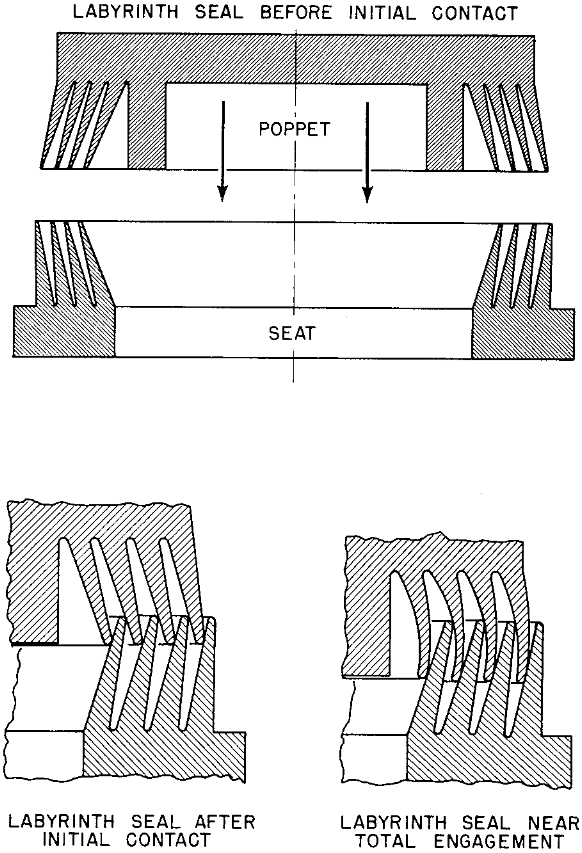

FIGURE 81.—Cone labyrinth valve—prototype version.

Some of the advantages of this design are:

(1) High corrosion resistance is afforded by the material choices possible with all-metal design. Corrosion resistance is further increased by minimizing the erosive cavitation normally associated with deep throttling but which is almost nonexistent in capillary throttling.

(2) Because of the all-metal design, extreme temperature capability is possible and service in cryogenic or in liquid metal systems is feasible. Radiation and vacuum problems associated with the use of elasttomers and plastics are similarly eliminated.

(3) The seats are insensitive to contamination, since impurities are scraped away instead of being crushed or embedded. This is particularly desirable when the valve is controlling metallized propellants.

(4) Seal life is extended by the self-adjusting feature which results in both cleaning and self-lapping of the seat-contact surfaces.

(5) Sealing is made redundant by the use of multiple seats.

(6) The pressure range which can be throttled is extended by the division and spread of the energy conversion process over several labyrinth stages.

(7) Flow-control characteristics are improved without the discontinuities associated with cavitation.

(8) The intermeshing action and flexibility of the multiple seat blades provide many of the advantages of soft seats without the incorporation of the less-durable elastomers and plastics.

Lewis Research Center developed a quick-disconnect coupling for cryogenic and hazardous fluids which utilizes the labyrinth seal principle. The leakage of this design is reduced by the additional use of elastomeric seals. As shown in figure 82, easily replaceable soft seals are positioned both inside and outside the labyrinth. Patents covering this design have been applied for.

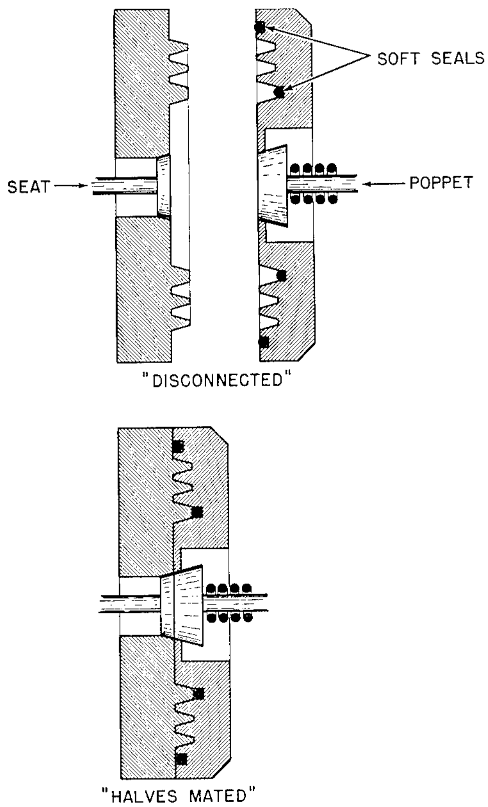

FIGURE 82.—Quick-disconnect coupling with labyrinth seal.

At Marshall Space Flight Center, labyrinth seals of Teflon have been developed and are being used to solve many longstanding leakage problems.

Inflatable and Large Valve Seals

Leak prevention usually requires a different approach in large valves than in small valves. At Lewis Research Center, a 10-foot-diameter, high-vacuum valve was required in test equipment for ion engines. A special gate valve using a unique sealing method was designed and fabricated. In operation, the gate is lowered to its bottom position without contacting the seats. Eight equally spaced pistons then are actuated to force the gate against rubber seals (double concentric O-rings). To open, the gate retracts from the seal before raising.

A problem with large wind tunnel diverter valves was encountered and solved at Ames Research Center. Twenty-foot- and twenty-four-foot-diameter valves in the unitary wind tunnel complex divert the flow through one of two passages. Personnel at this NASA center developed an inflatable rubber seal for use around the periphery of the valve disk. The inflatable rubber seal is deflated before valve actuation and reinflated after valve actuation to prevent leakage around the 24-foot-diameter seal.

Flange Seals

Considerable work has been done at Marshall Space Flight Center to stop leakage from flange seals. Tests of a number of materials indicated that Teflon is the most satisfactory material at cryogenic temperatures for flange seals. In addition, unique design developments have emerged for solving flange seal problems. Figure 83(a) illustrates the flat surface method in which Teflon seals (or gaskets) are normally clamped between two flat metal flanges. When leaks

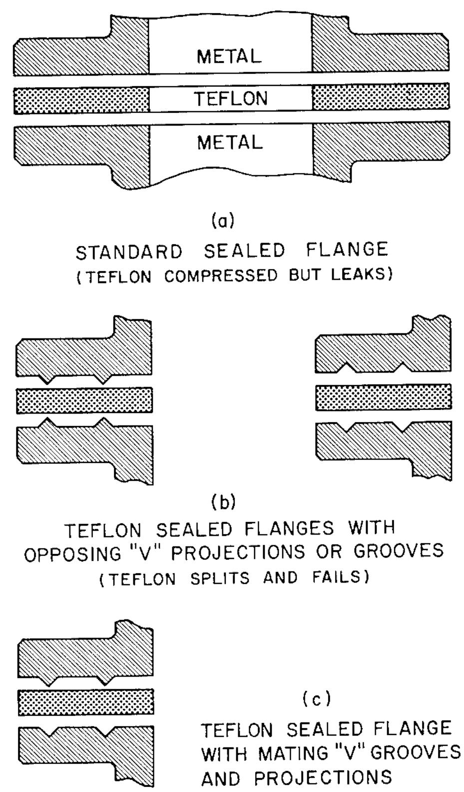

FIGURE 83.—Flange seal leakage prevention.

occurred, attempts were made to stop the leakage by machining projections or indentations on the metal surface such as are illustrated in figure 83(b). These approaches were unsuccessful because the metal cut into the Teflon and eventually sheared through the material. A simple solution is indicated in figure 83(c). Grooves and mating projections are cut into the metal parts, so that when these flanges are clamped together (within strict torque tolerances) the Teflon material provides the most effective seal.

An even more reliable seal can be made by the following procedure: After the Teflon seal has been placed between the flanges and torqued to the specified torque, it is placed in a preheated oven of 160° F for 3 hours, removed from the oven and allowed to return to ambient temperature, then retorqued. The heating process helps the Teflon flow into the flange grooves but also requires the flanges to be retorqued. The torquing of this seal is very important and should be accomplished in the most uniformly loaded sequence. Complete torque should not be obtained in the first operation. For example, if complete torque is to be 100 inch-pounds, torque in steps of 50, 70, 90, and 100 inch-pounds should be obtained.

Valve Seat With Elastic, Scrubbing Action

The Jet Propulsion Laboratory developed a simple, elastic metal tubelike seat which is formed in a valve to receive a ball member of pintle-type closure (ref. 3). As the ball is moved down on the valve seat in a closing action, the tubular seat is forced radially outward to create a scrubbing action on the closure surface. This action is illustrated in figure 84. The scrubbing action tends to clear away any particles which might hold the valve seat partially open. It is necessary to provide a backup ring around the tubular valve seat. The ring is sized to stop the radial expansion of the valve seat before the tube material reaches its elastic limit. Figure 84 also illustrates several different methods and designs for utilizing the expanding seat and backup ring concept.

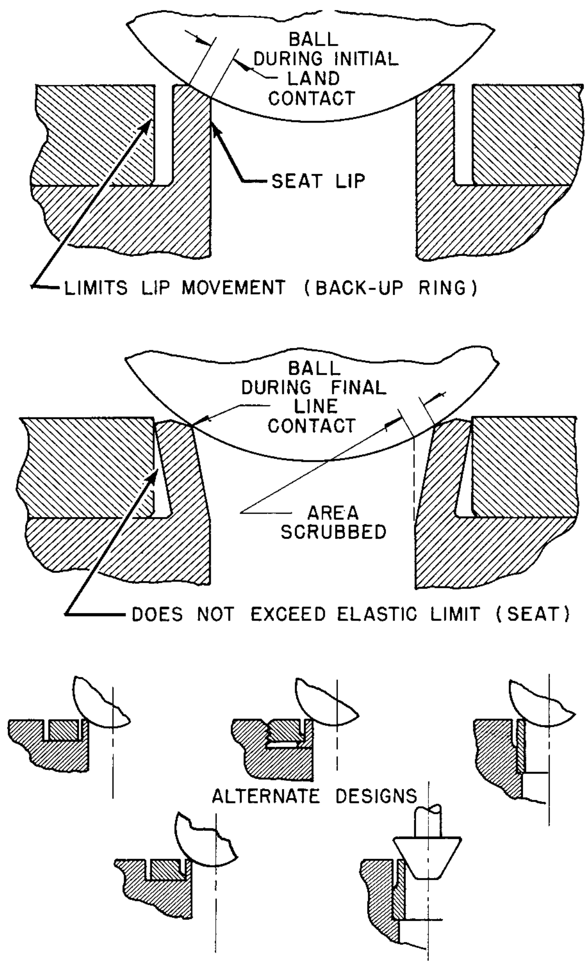

FIGURE 84.—Valve seat with expanding and scrubbing action.

Bull Nose O-Ring

A major improvement in reducing leakage, made at the Jet Propulsion Laboratory, is a "bull nose O-ring." It is designed to fit into a standard AN plumbing system using a conventional flare-tube configuration. A standard male fitting is provided with an O-ring groove cut into the external surface near the end. An O-ring is fitted over the groove as illustrated in figure 85. Finger-tight assemblies were found to have no leaks when tested with helium at 4000 psi. This modification is primarily for fittings, but might be incorporated into a valve design.

Floating, Nonrotating Poppet

Much work has been done at Jet Propulsion Laboratory to determine new design configurations for dead tight shutoff. Figure 86 illustrates several methods of achieving self-alinement of a ball or poppet in the valve seat. These designs reduce wear by not allowing the poppet to rotate against the valve seat. These particular valve designs were developed for advanced liquid propulsion systems—specifically, the Mariner "C" spacecraft. Leakage measurements with a mass spectrometer indicate helium leakage rates of the order of 1 atm cc/yr.

At one stage in the development of this valve, a 0.125-inch-diameter sapphire ball was used against a seat diameter of 0.085 inch. The 6061–T6 aluminum seat was diamond lapped to produce a 0.002-inch chamfer at an angle which would mate against the ball. The finish on the ball was 1 microinch rms or better. A further modification involved the use of a 1/4-inch-diameter aluminum oxide ball coated with molybdenum disulfide. Molykote Z is satisfactory, but Molykote in the 5-micron size range is better. To apply the coating, the aluminum oxide ball is rolled between a molybdenum disulfide powder-coated rubber pad and a hand-held Teflon block. The ceramic ball picks up a thin coating of molybdenum disulfide and a microscopic amount of Teflon.

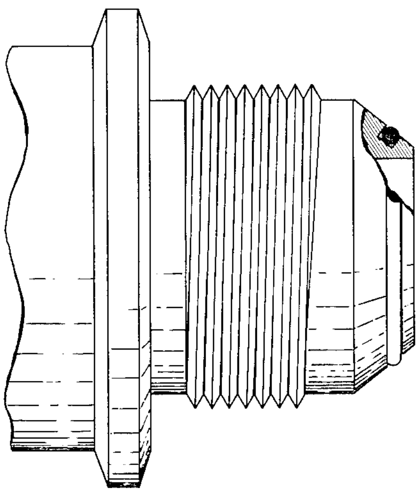

FIGURE 85.—Fingertight assembly is leakproof up to 4000 psi.

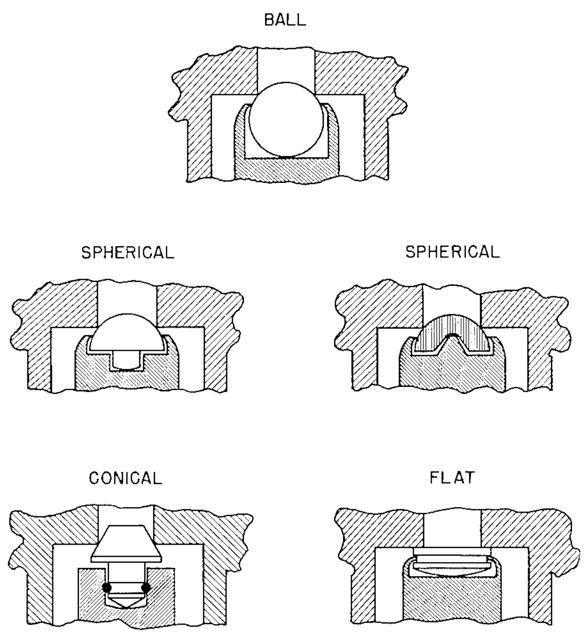

FIGURE 86.—Floating, nonrotating poppet for dead tight shutoff.

Valve Seats Despite Misalinement

A unique design was developed at Lewis Research Center to allow a valve to seat despite misalinement of the stem (ref. 4). It is illustrated in figure 87.

The sealing element is a conical plug mounted on the end of a valve stem. The cross section of the valve plug is the shape of a shallow cone. In closing, the outer edge

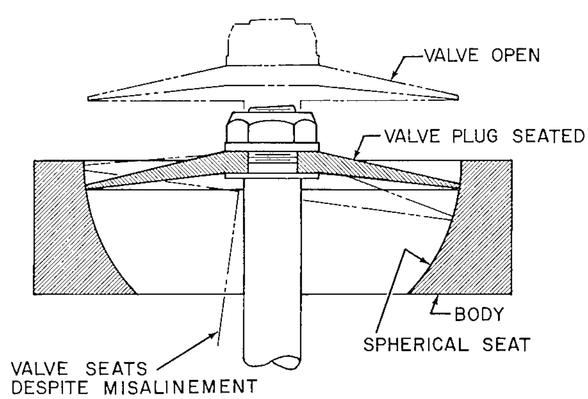

FIGURE 87.—Valve seats despite misalinement.

or circumference of the cone contacts the spherical valve seat. This arrangement permits the valve to seal effectively even though the valve stem is out of axial alinement. The conical-walled valve plug is always perpendicular to the tangent of the spherical valve seat at the point of contact, whether the stem is in its proper position or not. Uniformly diminishing thickness of the conical valve plug maintains a uniform level of stress in the plug. Tight sealing is possible without exceeding the elastic limit of either the plug or seat material, allowing many hundreds of operating cycles.

Diaphragm Backup Rings

A backup ring serves two primary functions: (1) it provides radial support to a central reciprocating member, and (2) it supplies low spring rate structural support to a diaphragm placed across the face of the reciprocating members (refs. 5 and 6). Figures 88(a) and (b) illustrate two possible backup-ring configurations.

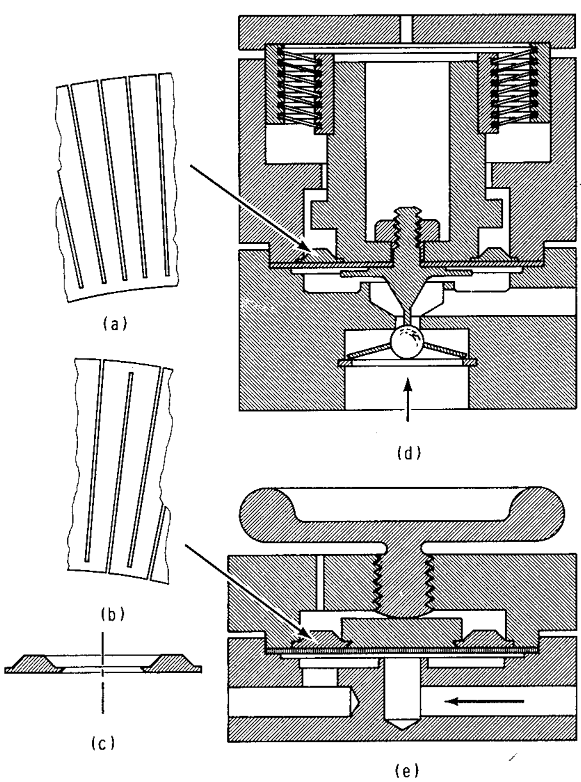

FIGURE 88.—Backup ring for flexure diaphragms.

In general, a backup ring consists of a ring slit radially to form a number of pie-shaped segments or beams. The segments are joined either along the outside or the inside periphery of the ring as shown in figure 88(a). Alternatively, the segments can be joined at both the inner and outer peripheries as illustrated in figure 88(b). The slitting enables the ring to assume a conical shape with the application of very little force. Thus, each segment of the backup ring is able to flex with the reciprocating member.

The backup-ring materials are contoured in thickness to obtain uniform stress distribution. The annuli of support (inner and outer peripheries) of the ring are thinned to carry shear stresses; the central section is thickened to sustain bending stresses.

For small displacements, a backup ring has a low spring rate and is nearly frictionless. Each backup-ring segment acts as an individual toggle to provide seizure-free radial support. Although not absolutely necessary, lubrication can be applied to the support annuli to minimize friction and wear.

A one-piece backup ring is sufficiently flexible if the outer diameter is large, but small-diameter rings are sometimes excessively stiff. To overcome this objectionable stiffness, the following procedure has been used: (1) each segment of the backup ring is scribed with an identifying letter and the web joining each segment is then broken; (2) the segments are reassembled in proper sequence in a jig; (3) self-vulcanizing rubber paste is applied to the back surface of the segments; and (4) after curing, the rubberbanded backup ring is removed as one piece from the jig. The rubberbanded backup ring that results is usually sufficiently flexible.

To seal the fluid, a metal or elastomer diaphragm is placed over the backup ring. Figure 88(c) illustrates a coned inner circumference to provide positive retention in a sealing groove, which was covered by a 0.003-inch-thick 1100–0 aluminum diaphragm and pressurized to 5400 psi without noticeable damage.

Backup rings have been used successfully in the Ranger, Mariner, and other spacecraft. A schematic of the Ranger and Mariner midcourse propulsion system pressure regulator is shown in figure 88(d). A possible valve design incorporating a backup ring is illustrated in figure 88(e).

Indium-Seated Valve

At the Jet Propulsion Laboratory, indium is used as a seat material for a special valve for high-vacuum, cryogenic-temperature applications. At low cryogenic temperatures, Teflon becomes brittle. Indium stays ductile at these extreme temperatures and is useful in systems for processing liquid helium.

In this valve design, the seats always remain in a horizontal position. The indium-seat material is soft, similar to lead, thereby providing a good sealing action. Each time a valve is opened, the indium seat is electrically heated, melted, and cooled to form a new surface for the reclosure of the poppet. The indium seat is contained so that it remains in place during the liquefying operation. In this manner, a fresh, soft, new seat is provided for the poppet each time it closes upon the seat.

Self-Sealing Disconnect

The Jet Propulsion Laboratory developed a special disconnect fitting that would automatically form a positive metal-to-metal seal in tubing when the fitting was broken by disconnecting forces. The fitting need only be used once, but must not leak during or after filling operations (ref. 7).

This design has application in industry for one-time, pressurized filling operations; e.g., refrigeration systems that are filled and sealed during manufacture.

Figure 89 illustrates the novel fitting in which the fill tube, during disconnect action, holds against a metal sleeve to form a positive metal seal. The fill tube is made so that the inner end has a shoulder extending beyond the outside diameter of the tube. Holes at that end permit the passage of liquids

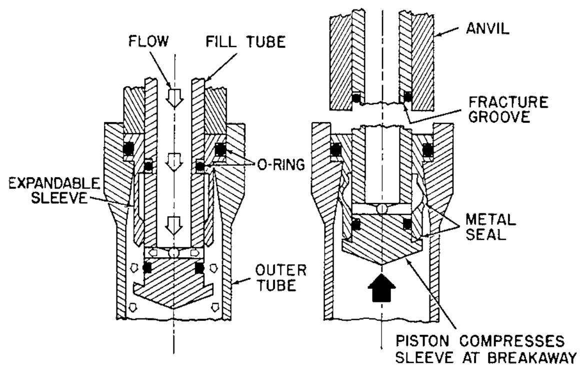

FIGURE 89.—Self-sealing disconnect forms metal seal after breakaway.

or gases. Surrounding the fill tube is a specially designed sleeve, also with a shoulder that drops into a recess in the main body of the fitting. During filling, O-rings in the shoulder of the sleeve and near the outer end of the fill tube seal against leakage. When the fitting is disconnected, as would occur during a rocket launching or when one part of a rocket separates from another in space, the fill tube breaks at the O-ring groove in the tube. Before it breaks, however, the disconnecting force pulls the shoulder on the inner end of the tube against the open end of the sleeve. The thin sleeve walls bulge out against the tapered inner wall of the main body of the fitting. This action establishes a metal-to-metal seal. An anvil that can be reused positions this sleeve until breakaway is completed. Two of the O-rings now act as backup for the metal seal.

Hermetically Sealed, One-Shot Device

The Hughes Aircraft Co. designed a hermetically sealed device consisting of a diaphragm and a spring-loaded plunger (ref. 8). Figure 90 shows the apparatus.

The bellows acts as a spring for the plunger and is held in a compressed position by a fuse link. The fuse link is melted by an electric current, releasing the compressed bellows which drives the plunger through the diaphragm. The bellows insures that the unit remains hermetically sealed after operation.

A plunger-pierced rupture diaphragm is

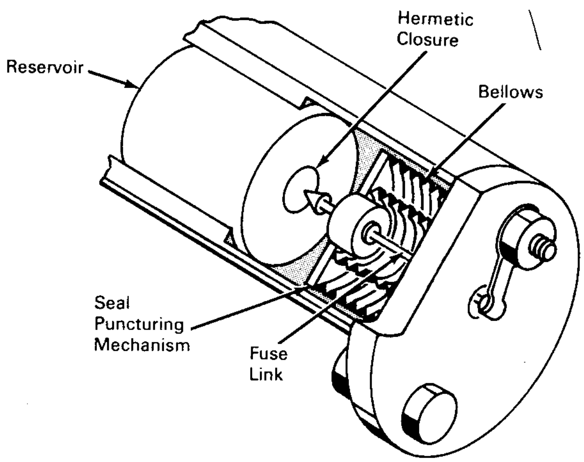

FIGURE 90.—Remotely actuated one-shot valve.

not unique. The combination of components incorporated in this device, however, may be useful for lowering fluid pressure in a container without losing the contained fluid. The actuating electric current could be triggered by a remote sensor. This is a one-shot device that must be replaced after each actuation.

Double-Seal Valve

Lewis Research Center developed a valve for use in transferring reactive fluids such as liquid fluorine (ref. 9). The valve has a central plug to block the bulk flow and a soft-seal outer seat to effect a final zero-leak sealing. This design is shown in figure 91.

The valve is made up of a trim group and a seat group. The trim group consists of a hemispherical plug, a loading spring that engages the plug rod, and a stainless-steel shutoff cap that is circular in form with a relatively sharp edge. The seat group consists of an inner stainless-steel seat that opens into a bellows flow passage and receives the hemispherical plug, plus a soft-seal circular seat that receives the stainless-steel shutoff cap.

In operation, fluid flow is directed past the trim group through the stainless-steel seat and bellows flow passage and seat group exhaust port. As the trim group is moved forward, the spring-loaded hemispherical plug contacts the stainless-steel seat and shuts off the bulk of the fluid flow. Contin-

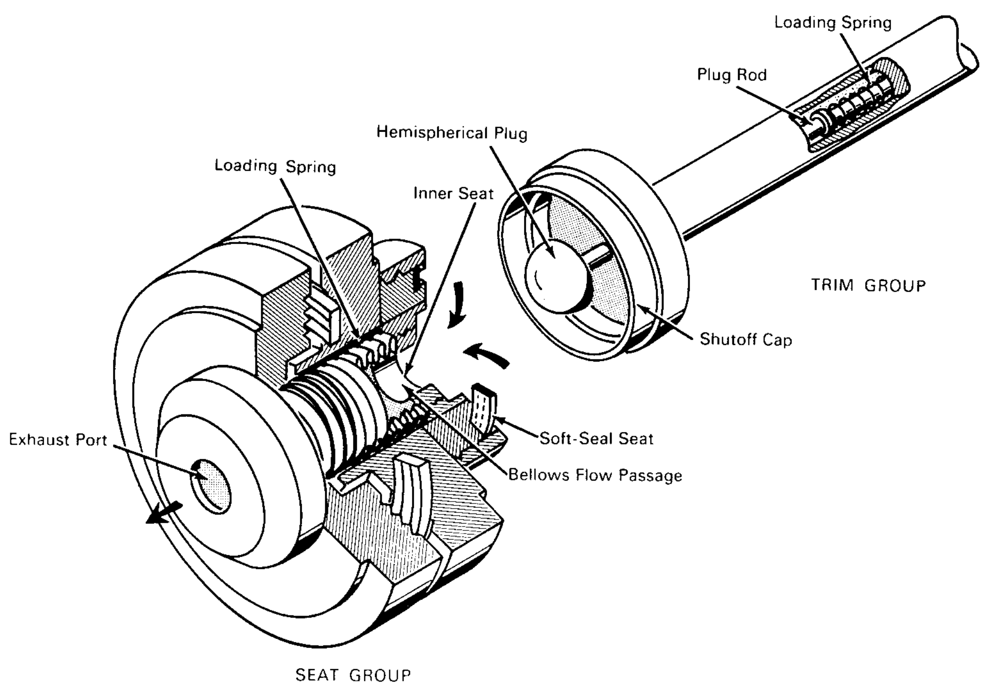

FIGURE 91.—Double-seal valve for hazardous fluids.

ued forward movement of the trim group brings the stainless-steel shutoff cap into contact with the soft-seal circular seat, completely closing the valve. Fluid trapped between the shutoff cap and hemispherical plug gradually gasifies and vents to the downstream side of the valve assembly through the round seal formed between the plug and stainless-steel seat.

The initial seal (which allows downstream venting) is accomplished by: (1) having the inner seat spring loaded, and (2) designing the annular surface area of the inner seat so that it is larger than the cross-sectional area of the bellows flow passage. Downstream venting of any fluid that may be trapped in the bellows spring cavity is accomplished through vent holes drilled through the inner seat.

Several soft-seal materials have been tested successfully. These consisted of tetrafluoroethylene combined with various metal fillers and with a glass filler. This device has been tested and found acceptable up to liquid fluorine flow rates of 2 pounds per second at pressure differentials up to 97 psi.

TWO-PART, QUICK-COUPLING VALVE

Filling large tanks from tanks of lesser capacity involves much handling of valves in both donor and recipient system. Fluid may be lost and a great deal of time expended. To solve this problem, the Jet Propulsion Laboratory developed a two-part valve, shown in figure 92, for use in the Mariner satellite program (refs. 10 and 11). One part remains integral to the recipient system, acting as a check valve when filling is not taking place, while the other part remains integral to the donor system.

The valve part is integral to the recipient system; it consists of assembly *A*, with feed port *A*, into which a ball stop is pressed by a threaded plug; the ball stop is captive. Feed port *A* is diamond lapped to provide a seating land 0.002 inch wide that follows the contour of the ball stop. The ball stop is constructed of ceramic material burnished with molybdenum disulfide. Movement of the plug and ball stop in and out is accomplished by means of a socket wrench drive engaging a square recess in the base of the threaded plug. When not used in a filling

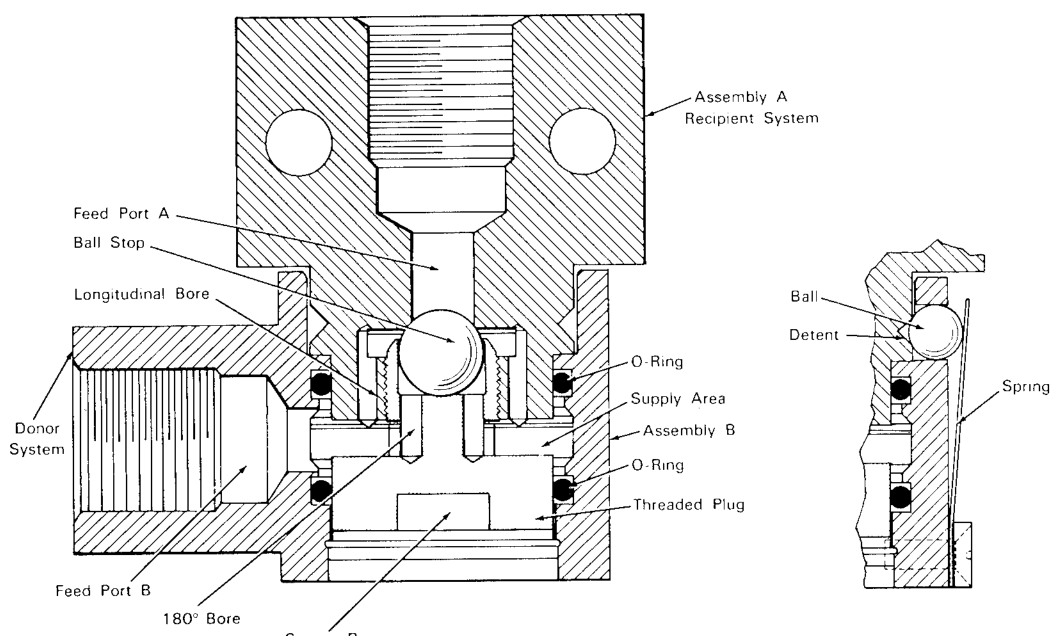

FIGURE 92.—Two-part, quick-coupling valve.

operation, the plug is drawn up so that the ball seals the feed port.

The valve part that is integral to the donor system consists of assembly *B* having a feed port *B* which mates with the supply area in the recipient system assembly *A* when in place. Two O-ring seals entrap all fluid under pressure within the supply areas so that the pressure is equalized to the two parts of the valve when it is assembled. Twelve longitudinal bores are arranged at 30° intervals around the periphery of the movable plug to direct the fluid being transferred from the supply area into the feed port *A*. Assembly *B* is retained in a position connected to assembly *A* by two phosphor-bronze spring and sapphire-ball devices which mate with detents on each side of assembly *A*. Two bores 180° apart within the threaded plug buoy up the ball stop to prevent shattering in the open position.

This valve has been opened and closed while handling pressures up to 3000 psi. Excellent sealing was obtained. Mass spectrometer tests with helium at 1700 psi revealed a leakage of the order of 1 atm cc/yr.

LOW-LEAKAGE SOLENOID VALVE

Another valve in the zero-leak nonrotating category was developed by the Marquardt Corp. for use in the Lockheed JF–104A. Figure 93 is a schematic illustration of this special-purpose valve. Rotation of the poppet against the Teflon seat is eliminated by use of a push-pull solenoid. Numerous Chevron seals are used along with an O-ring seal. No galling, leaks (either internal or external), corrosion, or other troubles have been experienced in this application. The valve action is relatively fast even though it is unbalanced.

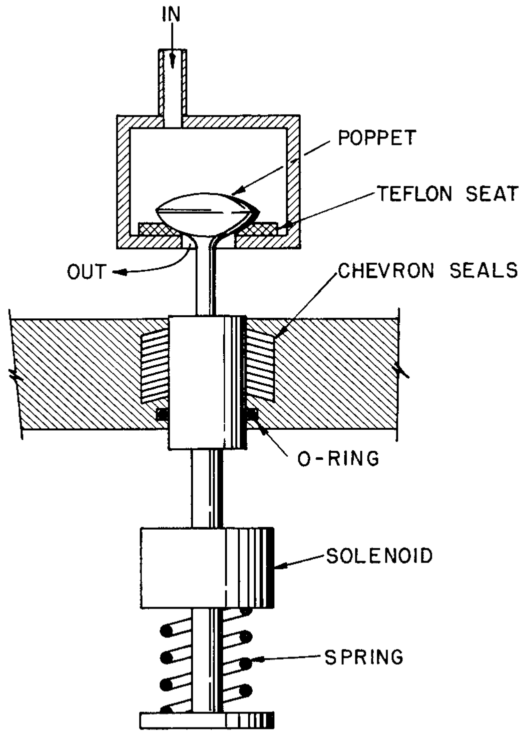

FIGURE 93.—Zero-leak valve.

EROSION PROTECTION

At Ames Research Center, a 6-inch throttling valve with positive shutoff was needed to handle the extremely dry air used in wind-tunnel tests. Pressure differentials across the valve ranged from 14.7 psi to 140 psi. Commercially available valves were satisfactory for only a short time; throttling of the dry air caused rapid galling of the nonlubricated seats. Stainless steel proved to be unsatisfactory for the disk and seat materials, as did chrome-plated steel. Finally, both the seat and the sealing areas were coated with Stellite, which was ground to a fine finish before the addition of a 0.020-inch-thick, hard-chrome plating. The chrome plating was then ground and polished to a fine finish. This combination of chrome plating over Stellite on the standard steel valve provided completely satisfactory dry operation.

SPECIAL COMMERCIAL VALVES

The Valve Division of Honeywell, Inc., designed a valve to handle a liquid lithium compound in a system in which a small air leakage could result in immediate solidification of the liquid in the process piping. An essentially zero-leak valve was necessary. Valve tests indicated an air leakage of less than 10^{-7} atm cc/sec, meeting the design requirement.

At Langley Research Center, a dead tight shutoff of helium at ambient temperature and at 6600-psi pressure was required. The Combination Pump Valve Co., Philadelphia, used a nylon insert in its standard plug valve to seat against a Monel seat. This design has proven satisfactory in a number of helium valves at Langley Research Center in sizes up to 2 inches.

At Lewis Research Center, valves in the zero-leak category were necessary for use in a helium system. Satisfactory valves of unique design were furnished by the P–K Paul Co.[1] Actual tests indicated that these 3-inch valves have a helium leakage of 5×10^{-6} atm cc/sec as measured with a mass spectrometer. This Hi-100 valve has a metal-to-metal seating surface, using a hollow ball of hard Stellite to seat on a softer Stellite. This valve is a caged-ball design with a stainless-steel body.

DIFFERENTIAL PRESSURE RING VALVE

North American Aviation designed a valve with but one moving part to sense a pressure change and to react accordingly (ref. 12). This concept is illustrated in figure 94. Its simplicity promised good reliability and its operation can be easily controlled by fluidic devices.

The mechanical feature of the valve is a pressure-riding annular ring that moves axially through its centroid. Controlled pressure differential across the top and bottom passages provides the force to move the ring either up or down, depending on the direction of the pressure differential. The ring itself is contained in an annular groove that provides guidance during motion and controls the amount of travel required. When P_1 is less than P_2, the valve will close as shown in figure 94(a). When P_2 is vented to a lower pressure level, P_1 will act against its annular area and provide the energy to move the valve to the open position, as shown in figure 94(b).

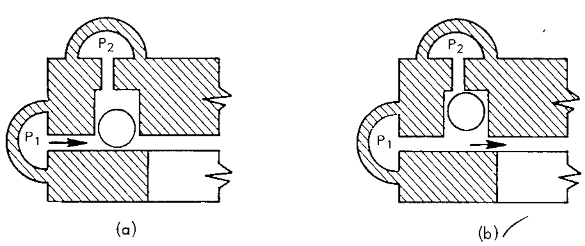

FIGURE 94.—Differential pressure ring valve.

[1] Now the Devar Kinetics Division of Consolidated Electrodynamics Corporation, Bridgeport, Conn.

FREEZE VALVE

A valve developed at Goddard Space Flight Center uses thermoelectric cooling to freeze flowing liquid. Fluid can be frozen and flow stopped in about 10 milliseconds. Reversing the current to the thermoelectric element melts the frozen fluid in about 1 millisecond. This valve has no moving parts and is a reliable design

ROTARY VALVE

At Langley Research Center, a unique rotary valve was developed that has potential use in industrial processes which require sequence timing of operations.

The valve is essentially a solid cylinder designed to rotate inside a barrel housing. Figure 95 illustrates the concept. A groove is machined on the outer surface of the cylinder so that it is filled with a fluid introduced through a port of the stationary barrel housing. Numerous passages are drilled in the cylinder to pass the fluid to slots machined around the cylinder. These slots match stationary ports in the barrel housing. In this manner, fluid flow from various outlet ports of this valve can be pulsed, se-

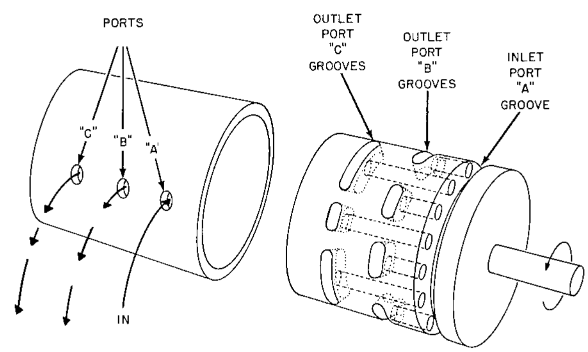

FIGURE 95.—Rotary valve.

quenced, overlapped, and timed to control various operations.

The problem of seals was overcome by using a very close fit between the inside of the barrel and the outside of the cylinder. However, some leakage may be experienced at high pressures. For leakage control at low rotational speeds, a T-connection can be used at the outlet ports *B* and *C* with one leg of the T containing an orifice which is sized to pass the leakage volume for return to a supply tank. When fluid flow, other than leakage, occurs at the outlet ports, this flow is then greater than the leakage flow through the orifice.

QUICK-RESPONSE VALVES

Check Valve

An extremely quick-response, low-inertia check valve was developed at the Jet Propulsion Laboratory. (See fig. 96.) The only moving part is an O-ring. The valve is only suitable for application in low- to medium-pressure systems (ref. 13). The normal flow of fluid passes over the outside diameter of the O-ring and compresses the O-ring radially inward. Should reverse flow occur, the higher pressure is applied around the inside diameter of the O-ring; thus, the O-ring is forced radially outward to seal off the reverse flow. Since only the lightweight O-ring moves, inertia is low and response is quick.

Plug Valve

Helium at pressures up to 2000 psi flows through one of the hypersonic tunnels at Ames Research Center. A custom-designed valve pulls a plug from the test-chamber nozzle in 2 to 3 milliseconds. Figure 97 illustrates the concept used in this custom design. The schematic only shows the method of opening the valve. An extension of the same concept can be used to close the valve in the same time of 2 to 3 milliseconds. Helium at 2000 psi exists around the plug, in the bleed port, inside the cylinder, and in the exhaust port. Upon electrical actuation of the solenoid, the small amount of helium contained in the cylinder is vented to the atmosphere. Because the bleed port has a much smaller diameter than the exhaust port, the pressure differential of 2000 psi is imposed across the piston. When the piston stroke has been completed, the cylinder is sealed from further helium leakage by the O-rings of the piston.

In this instance, the quick-response times are enhanced by an extra-heavy-duty solenoid and the use of helium rather than air. Helium molecules can accelerate more rapidly than air molecules because of their lower molecular weight and mass.

Bistable Valve

Figure 98(a) is a cutaway view of a valve actuator developed for NASA by the Marquardt Corp. The toroidal permanent magnet shown is arranged with one pole at the inner radius and the opposing pole at the outer radius. Magnetic lines of flux from this magnet, ϕ_1 and ϕ_2, flow through magnetic

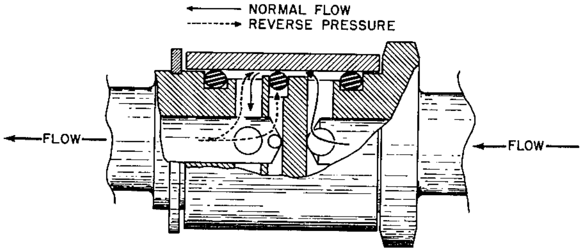

FIGURE 96.—Quick-response, low-inertia check valve.

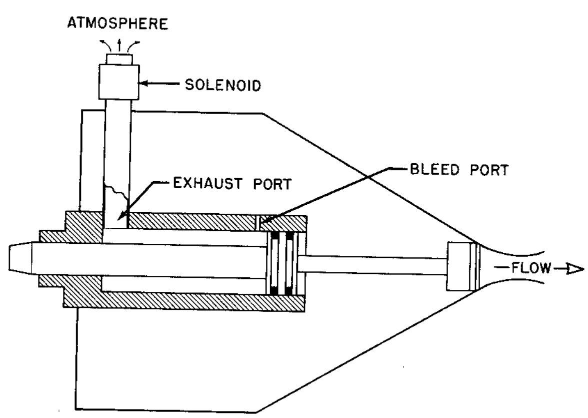

FIGURE 97.—Quick opening/closing plug valve.

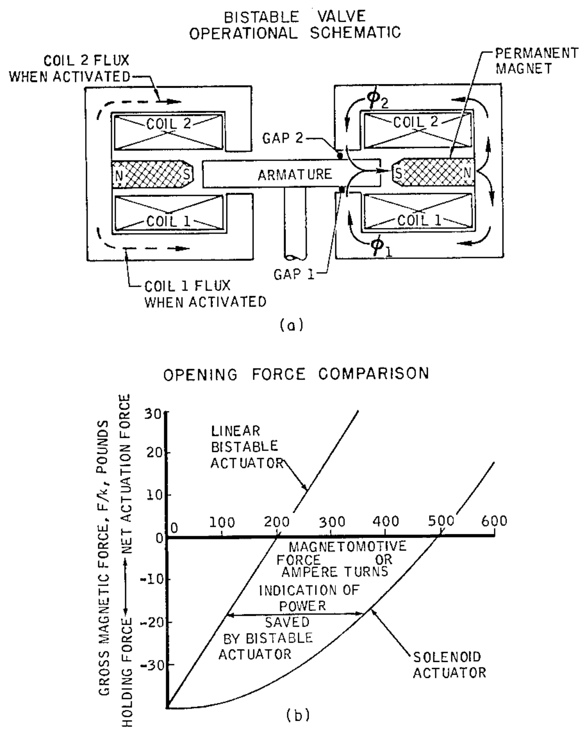

FIGURE 98.—Bistable quick-response valve.

reluctance gaps 1 and 2. The highest flux density, and consequently the largest attractive force, exists in the shortest magnetic reluctance gap when no current is flowing through actuator coils 1 and 2. Since the dimensions are such that gap 1 is shorter than gap 2 when the valve is closed (as shown), the flux produced by the permanent magnet acts to hold the valve closed. This condition is shown in the schematic where $\phi_1 > \phi_2$ because gap 1 is shorter than gap 2. Similarly, since gap 2 is shorter than gap 1 when the valve is open, the permanent magnet flux tends to hold the valve open. Consequently, the valve has two stable positions and is said to be bistable.

A pulse of current to the appropriate actuator coil serves to switch the higher magnitude of magnetic flux density from one gap to the other. If, when the valve is closed (armature is down, gap 1 is the shorter) as shown in the schematic, current is caused to flow in coil 2 in the proper direction, the flux in gap 2 will increase and that in gap 1 will decrease; when the magnetic force of attraction in gap 2 becomes large enough, the valve will open. Likewise, a pulse of current in coil 1 will close the valve.

Figure 98(b) shows that less electric power is required for a bistable actuator than for a solenoid actuator. Since actuator inductance delays the buildup of current, the bistable actuator with its smaller current requirement for a given inductance can respond much faster (approximately 3.5 to 4 milliseconds, nominal) than many other designs. An additional advantage of this design is that it does not require a spring-load return. This feature avoids the problem of poor repeatability of response time which is normally associated with the use of springs. The two limitations of this valving concept are (1) it does not incorporate a fail-safe feature in the closed position, and (2) special switching circuitry is required for operation. Design modification may decrease or eliminate these limitations.

Bipropellant Valves

A valve for a 100-pound-thrust engine developed by Parker Aircraft under a NASA contract has been tested extensively and is potentially very useful. A schematic of this design is shown in figure 99. This valve has been tested at 250 psi with nitrogen gas and repeatedly indicates a 2.5-millisecond opening time and a 0.8-millisecond closing time. These response times would probably increase slightly if an actual propellant were used. This valve draws 2.33 amps at 28 volts and weighs 1.1 pounds. Bellows are used to prevent internal mixing of propellants. If any leakage occurs, both bellows must fail before an explosion could occur. It is highly desirable to link both poppets mechanically to match response time and full-open and full-closure positions.

Rupture Diaphragms

Ames Research Center performs free-flight model tests at velocities up to 50 000 ft/sec (approximately 34 000 miles/hr). Figure 100 is a schematic illustration of the test facility. An explosive powder charge is detonated in a high-pressure vessel (1).

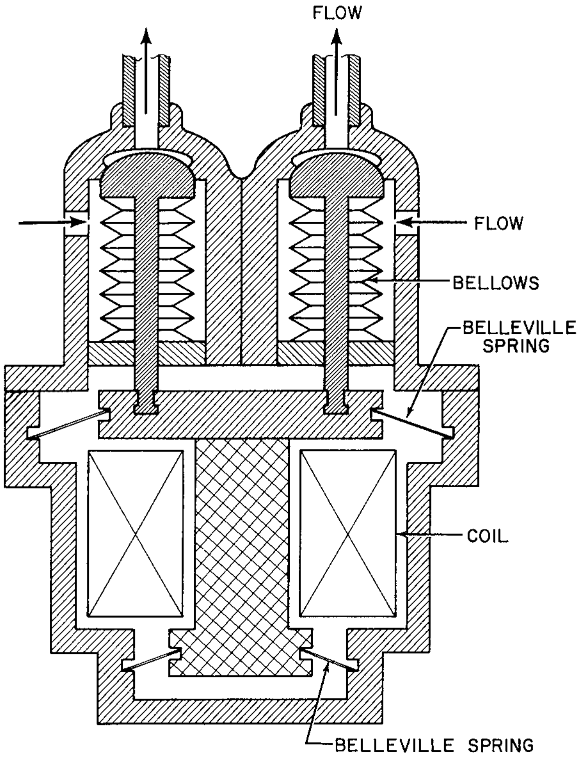

FIGURE 99.—Quick-response, bipropellant valve.

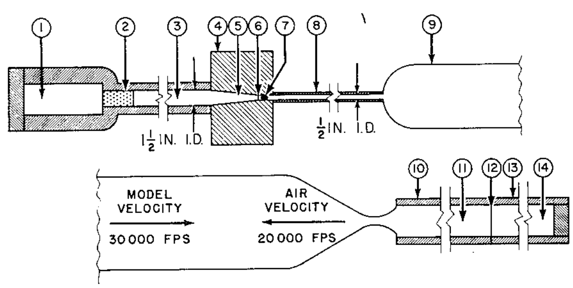

FIGURE 100.—High velocity test facility valves. (1) Powder charge; (2) Slug: Teflon or nylon; (3) Hydrogen (or helium); (4) High-pressure coupling; (5) 350 000 psi at 20 000° F; (6) Diaphragm (rupture type); (7) Model; (8) Launch tube; (9) Free-flight chamber; (10) 90-ft naval gun; (11) Dry air at 20 psi; (12) Explosive diaphragm; (13) 70-ft naval gun; (14) Helium, hydrogen, and oxygen.

The resulting products of combustion, under high pressure and temperature, move a 1½-inch-diameter Teflon or nylon slug down the tube.

The moving slug, acting like a piston, compresses hydrogen or helium in the high-pressure coupling (4). Area 5 in the high-pressure coupling then contains the helium or hydrogen gas at a temperature of 20 000° F and a pressure of 350 000 psi. A rupture diaphragm (6), designed to contain the gas momentarily at this high temperature and pressure, gives way, and the gas drives a sabot (7) down the launch tube (8) and into the free-flight chamber (9). The sabot, an incasement around the model being tested, separates from the model in the free-flight chamber.

At the other end of the system a mixture of helium, hydrogen, and oxygen is contained in a 70-foot-long naval gun (13). The hydrogen-helium-oxygen mixture is ignited by an exploding wire which is detonated by a high-potential electrical discharge. This explosive combustion heats the helium gas up to 5000° F and creates a pressure of 7000 psi. This high temperature and pressure gas is contained by a diaphragm (12). Upon signal, an explosive charge is detonated, rupturing the diaphragm. A shock wave is driven down the tube (10), heating the dry air in the tube. The pressure difference across the throat is sufficient to provide an air velocity of 20 000 fps.

Extensive research programs were conducted at Ames Research Center in developing these explosive diaphragms (12). Shaped explosive charges of many different temperatures were used. To protect the explosive charge from the high gas temperatures, an epoxy potting compound is used for a thermal insulation on the diaphragm.

Explosive rupture of the diaphragm can allow loose metallic pieces to enter the free-flight chamber. To prevent this, it is desirable in many instances to let the gas overpressure, rather than an explosive charge, rupture the diaphragm. This can be accomplished by selectively weakening the diaphragm in critical areas. Any desired shape can be cut or etched in the diaphragm, but close control on depth of penetration is required. For example, an X-configuration can be etched into the diaphragm to a depth of

75 percent of its thickness, plus or minus 5 percent. In other configurations, circular arcs of 300° are etched into the diaphragm so that when the pressure ruptures the diaphragm, the disk hinges out of the way without shedding any loose material.

Ring Valve

North American Aviation developed a simple, reliable, and quick-acting ring valve (ref. 14). As shown in figure 101, two porting rings, one within the other, control flow by using seal buttons as the sliding valve closes. Multiporting within the rings allows close control of the gas or liquid flow by the slight rotation of the outer porting ring. In this way, only a small actuator travel is required—one of the factors necessary for a fast-acting valve.

The outer or actuating ring contains plastic-seal buttons immediately adjacent to the ports and flush with the outside surface of the inner porting ring. The seal buttons ride against a plastic-coated surface of the inner ring, and are maintained by steel spring washers and the fuel flow pressure acting from behind them. The respective positions of the flow ports of the inner ring with the flow ports or seal buttons of the outer ring determine the amount of flow through the valve. The outer ring is hydraulically actuated by the gas or liquid of the operating medium and is controlled by a small solenoid valve.

Although no response time has been reported, tests at pressure differentials up to 1000 psi were conducted with no observable leakage after 275 actuation cycles. Flow rates through this valve were as high as 371 gallons per minute.

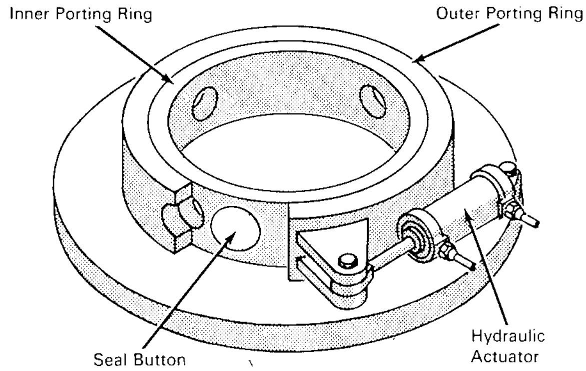

FIGURE 101.—Quick-acting ring valve.

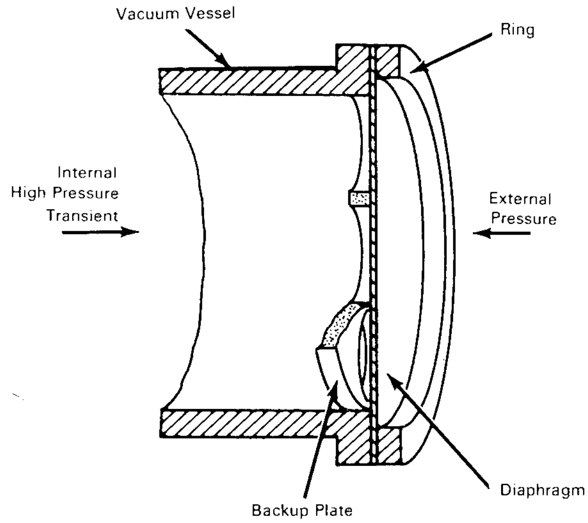

FIGURE 102.—Two-way burst diaphragm valve.

Two-Way Burst Diaphragm

Jet Propulsion Laboratory developed a burst diaphragm that will withstand a larger pressure differential in one direction than in the other (ref. 15). This diaphragm protects vacuum vessels from transient internal pressures. Figure 102 shows the design.

The diaphragm consists of a circular sheet of material stretched over a large-diameter ring. The side of the diaphragm facing the vacuum vessel is supported by a backup plate containing several holes of a diameter appreciably smaller than the inside diameter of the large ring.

If a sufficiently high-pressure transient arises in the vacuum vessel, the diaphragm will bulge outward and be supported only at the periphery of the large-diameter ring. An external pressure will cause the diaphragm to bulge inward through the small-diameter holes in the backup plate. The bursting pressure in this direction is greater because the backup plate supports the diaphragm over a greater area. The ratio of the bursting pressures is approximately inversely proportional to the ratio of the diameter of the holes in the backup support to the diameter of the large ring.

This diaphragm is gastight and is responsive to high-frequency pressure transients. The diaphragm can be used wherever it is desired to have different burst pressures in opposite directions.

Combustion Gas Sampling Valve

Texaco Research Center required a fast-response valve to withdraw samples of combustion gases from engine combustion chambers (ref. 16). The valve was machined from 304 stainless steel and is solenoid actuated. The sealing surfaces are faced with 24-carat gold bonded to the stainless-steel valve stem by epoxy cement.

This design was tested and operated for 2 years; its characteristics, therefore, were well established. The valve opening time is adjustable from 0.4 millisecond to 3 milliseconds, and exhibits excellent repeatability. The measured leakage was 10^{-5} cc/min with the valve installed in an operating engine (maximum engine pressure was 50 atmospheres) and connected to a collecting volume (collecting volume pressure was 10^{-2} torr).

Gate Valve

In some industrial research-and-development programs, a single-actuation quick opening valve may be useful. At Ames Research Center, the valve illustrated in figure 103 opens a lightweight gage in less than 1 millisecond. An electrical wire and helium at 300 psi are contained in the thin space between two Teflon membranes. Helium or hydrogen is used, since they can accelerate more rapidly than air. An electrical discharge from a bank of condensers is used to explode the wire, rupturing the Teflon, and releasing the gas on one side of a lightweight piston. The 300-psi gas on the other side of the piston moves the piston to open the gate in less than 1 millisecond. This particular design was utilized in a facility where free-flight test models are fired through the air, down an air-filled tube, and into a gas-filled chamber under higher pressure.

AIR PRESSURE HOLDS HELIUM AT 10 ATMOSPHERES

Figure 104 illustrates a valve design developed at Ames Research Center where wind-tunnel models are fired from a gun down a pipeline and into a 10-atmosphere-pressure helium tank. The 6-inch pipeline had to be essentially wide open, since any valve in the line could not open rapidly enough for successful tests to be performed.

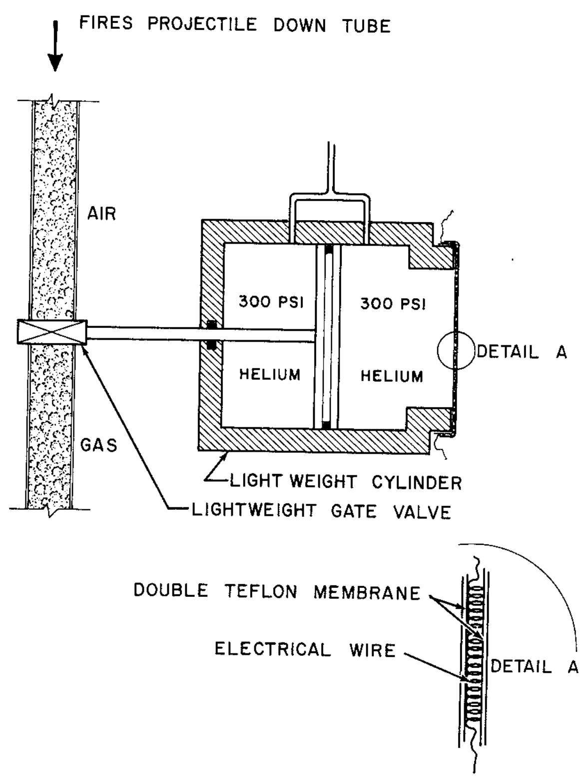

FIGURE 103.—Gate valve opened in less than 1 millisecond.

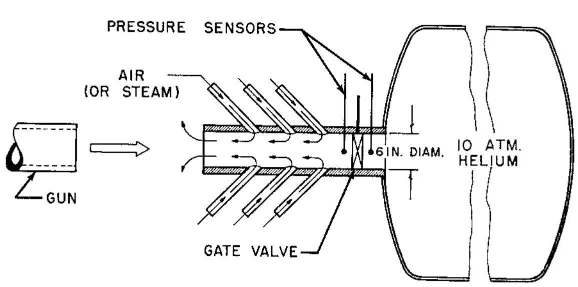

FIGURE 104.—Ames design for wind tunnel tube.

The scheme developed circumvents the lack of adequate valve-response time.

A standard gate valve contains the pressurized helium prior to the test run. Air is introduced into the 6-inch-diameter tube through rings. Each ring can build up approximately 3- to 5-atmosphere pressure on the gate valve. Three rings were used to build up 10 atmospheres of air pressure on the backside of the gate valve to balance the 10 atmospheres of helium pressure on the other side of the gate valve. When pressure sensors indicate matched pressures across the valve, it is opened. Little intermixing of air and helium occurs when the pipeline remains open for the gun to fire a free-flight model down the tube and into the pressurized helium. The nozzle arrangement and the injector shape are extremely important. This valve is in reality a coaxial multistage injector.

PROPORTIONAL FLOW CONTROL

Sliding Stem Valve

Flight Research Center engineers required truly proportional flow control valves in a sliding-stem proportional valve system for several research aircraft. Specifications were so exacting that they could not be met by commercial suppliers. The requirements were for a totally stainless-steel-and-Teflon valve with motorized operation. It was intended for operation at ambient temperature. The material problem was one of compatibility with hydrogen peroxide.

The temperature of the env... which this valve functions is... to 75° F, with heating a... required. (Because rocket re... rates when the H_2O_2 temperatu... below 60° F and the autodecomposi... increases as the temperature increases, a temperature of 75° F is a usable compromise.)

Figure 105 illustrates the unique design concept that provided truly proportional flow control in this application. A series of very small holes is drilled in a spiral pattern in a hollow valve stem. With linear stem travel, more holes are available for the transmission of hydrogen peroxide changes in proportion to the distance traveled. Good proportional flow characteristics are obtained with this design.

Curtain Flap Valve

A unique design for obtaining proportional flow was developed for NASA's Flight Research Center. Figure 106(a) illustrates the concept. One end of a Teflon curtain is secured to a rotatable valve stem. The other

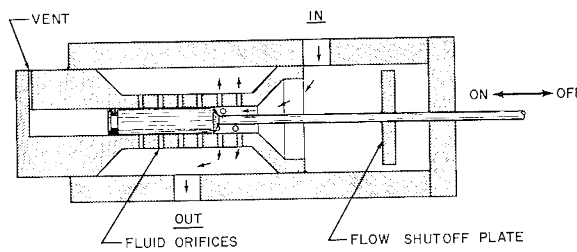

FIGURE 105.—Sliding stem proportional valve.

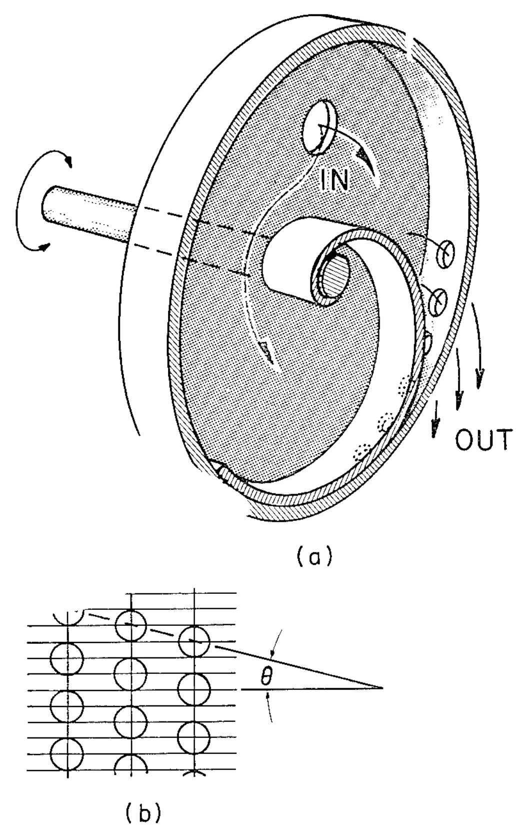

FIGURE 106.—Curtain flap proportional valve.

end of the curtain is secured to the inside of a stationary hollow cylinder. Fluid under pressure is introduced into the hollow cylinder, exerting fluid pressure against the curtain flap which covers all holes to prevent flow. The holes in the hollow cylinder should be small enough to prevent extrusion of the Teflon curtain through the holes. As the valve stem is rotated, the curtain flap is either wound on or unwound from the rotating stem, thereby covering more or fewer holes. Fluid flow is, therefore, proportional to the amount of stem rotation.

Figure 106(b) illustrates the geometric layout of the holes around the hollow cylinder. The angle O and the separation are chosen so that the holes are in a uniform angular-step pattern. In this way the number of holes covered by the curtain is proportional to the amount of stem rotation and proportional flow results.

SHUTOFF AND TIME-DELAY VALVE

North American Aviation designed a valve that increased fluid-flow area at a uniform rate and incorporates shutoff and time-delay features (ref. 17). Figure 107 illustrates this design in which a metering spool moves at constant velocity under pneumatic pressure and spring compression to achieve uniform flow area increase.

As the fluid flows through the main section of the valve from inlet to outlet, the rate of flow is controlled by changing the flow area. This is done by stroking the metering spool at a uniform rate.

A spring holds the spool in the initial position. Pneumatic pressure is applied to the closing port, causing air to flow through the check valve and into the control cavity. The pressure builds up rapidly in the control cavity, exerting a force on diaphragm 1, which moves the spool into its shuttled position. In this position the flow area is minimal. If the desired minimum flow area is greater than zero, a mechanical stop within the valve will be required to hold the spool away from its seat. If the minimum flow desired is zero, the seat may act as the stop.

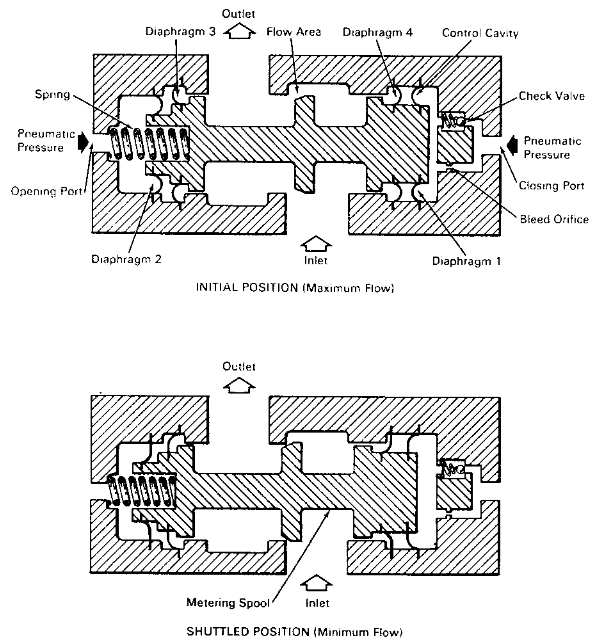

FIGURE 107.—Pneumatic shutoff and time-delay valve.

To stroke the spool through its metering cycle, pressure is applied at the opening port at the same time that pressure is vented through the bleed orifice at the closing port. Pressure at the opening port exerts a force on diaphragm 2, causing the spool to move to the open position.

The pressure in the cavity around diaphragm 2 is essentially constant as is the force exerted by the spring over the displacement of the spool. The force on the spool is constant as a result, and therefore produces a constant gas pressure in the control cavity. Thus gas flows at a uniform rate through the bleed orifice, and the spool moves at a constant speed.

It is also possible with this arrangement to achieve a time delay prior to the motion of the spool. If the effective area of diaphragm 2 is sufficiently smaller than that of diaphragm 1, and equal pressures are applied in the cavities around the two diaphragms, then the force on diaphragm 1 will be greater than the force (due to the pressure acting on diaphragm 2 and the spring compression) acting on the left end of the spool. The excess force will hold the spool against its seat or stop in the shuttled position. The pressure in the control cavity must be bled down to the point where the opposing forces are equal before the spool can begin to move to the right. The time required to bleed the control cavity to this lower pressure is the time delay desired. Diaphragms 3 and 4 are used to isolate the controlled fluid and to balance the fluid-line-pressure forces on the spool. Diaphragm areas, control cavity volume, and bleed-orifice size may be varied to give any desired combination of time delay and spool traveltime.

VALVES FOR HIGH TEMPERATURE

The need for special valve designs often can be avoided by simply placing valves some distance from very hot or very cold parts of the system. In original designs for the X–15 rocket aircraft, pressure-loaded check valves were put immediately adjacent to rocket engines to control flow into these engines without delays in starting as well as dribble at cutoff. The heat developed in the engine was conducted back to the valves and caused severe operating problems.

Before undertaking an extensive program to redesign the valves, the possibility of merely relocating them was investigated. This study resulted in relocation of the valves at a point 8 inches from the rocket engine. This distance was enough to provide thermal isolation from the rocket engine without introducing serious problems in the timing of engine ignition shutoff.

At Langley Research Center, this same type of analysis resulted in the relocation of a fluid-control valve from the outlet to the inlet connection of a heat exchanger. Since the heat exchanger is the source of heat for the controlled fluid, the placement of the control valve in the cool inlet location permitted the use of a standard, commercially available valve.

At Langley, too, heat conduction from attitude-control motors to valves was encountered in a system where valve relocation was not possible. These valves handle a 90-percent concentration of hydrogen peroxide that will decompose at a temperature of 1364° F when placed in contact with a silver catalyst. The catalytic silver screen must be located immediately adjacent to the control valve. The flow of hydrogen peroxide through the valves normally keeps them cool. When the motor is intermittently fired, however, the off periods allow heat to be conducted back to the valve.

A special development program was undertaken to provide a conduction heat barrier at the valve flanges (ref. 18). Two special materials were incorporated in a design that provided a satisfactory solution. They were Armalon and Fluorogold.[2] A combination of them overcame a leakage problem associated with the laminated Armalon and cold-flow problem associated with the nonlaminated Fluorogold. The Armalon will stand compression without the associated cold flow, and the Fluorogold will prevent leakage associated with laminated structures. Figure 108 illustrates the design heat shield of a valve flange which was used for this application. The combination of these two materials has passed temperature-cycling tests from ambient to 600° F where other materials have failed. This design is good for liquid pressures up to 600 psi; it can withstand cycling through extreme temperature ranges where expansion and contraction occur; it prevents deformation of insulating materials; it is compatible with hydrogen peroxide; it has excellent thermal insulation characteristics; and it will withstand vibration loads greater than 10 g's.

Industrial High-Temperature Valves

Arde-Portland, Inc., in a company-funded program, developed a valve for the Air Force (ref. 19) to handle erosive and corrosive gases at temperatures up to 6500° F. The design employs an unusual self-cooling princi-

[2] Armalon: Trademark for TFE-Fluorocarbon resin-coated glass fabrics and laminates, E. I. du Pont de Nemours & Co., Inc.

Fluorogold: Trade name by Fluoro Carbon Co., Anaheim, Calif.

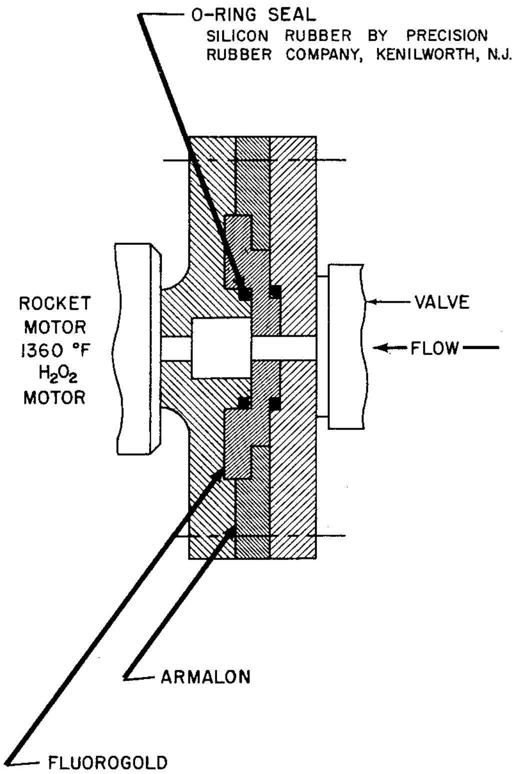

FIGURE 108.—Valve flange heat shield.

ple. Plugs, made of tungsten embedded with silver, are installed on the end of valve stems. In high-temperature operation the silver boils off into the atmosphere, thereby absorbing heat and cooling the valves. Successful operation was achieved in a 1-minute static firing at Edwards Air Force Base in a rocket motor burning highly aluminized solid propellant at temperatures above 5000° F.

Honeywell Research Center, Hopkins, Minn., developed a short-length, in-line, all-metal valve (ref. 20) for ultra-high-vacuum systems that has a leakage rate measured at less than 10^{-10} standard cc/sec with atmospheric pressure on one side. The valve is constructed of nonmagnetic stainless steel and copper and is a bakable assembly.

Flomatics, Inc., Natoma, Calif., furnished a 1500-psi valve to Ames Research Center for operation at temperatures up to 1540° F. A Stellite-on-Stellite design was used for this needle valve seat and stem.

NASA-Developed High-Temperature Valves

Bakable Vacuum Valve

Valves to provide barriers to molecular flow are often required for use in vacuum systems. Finely finished mating surfaces of poppet and seat are required. A valve for vacuum service (ref. 21) developed at Lewis Research Center utilizes Pyrex glass in a construction using a highly polished glass ball and seat, and has a magnetic slug embedded in the ball-type poppet to permit external operation. This valve is bakable to about 840° F.

All-Metal Valve

The Jet Propulsion Laboratory developed an all-metal valve to handle wide temperature and pressure ranges (ref. 22). This valve has been pressure-tested at 5000 psi and is capable of operating at temperatures from −459° to +1000° F. (This temperature limitation is due to Molykote lubrication of the actuation screw.) The construction of this packless valve is shown in figure 109.

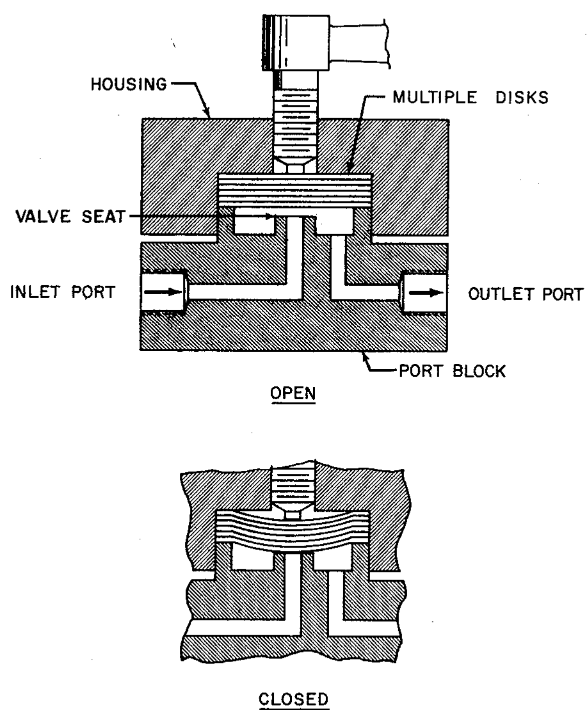

FIGURE 109.—All-metal valve.

The valve body is No. 347 stainless steel for corrosion resistance. The design uses a stack or spring disk. In the open position the flat form of the disk provides an opening for fluid flow past the ⅛-inch-diameter seat. For closing, the actuating screw deflects the disks to cover the valve seat. The bottom disk is plated with 0.002 inch of soft gold, which is left with a dull finish. This plating provides a soft seat for better sealing. This design should be excellent for throttling service.

Some nonmetallic valves are excluded from high-temperature applications because of excessive softening of O-ring seals fabricated from annealed-copper tubing. The tubing is fitted into a rectangular groove formed in one or both of two mating surfaces (such as pipe flanges). The cross-sectional perimeter of the groove is approximately the same as the cross-sectional perimeter of the tube. Hence, when compressed, the tube deforms and tends to fill the groove. Continuous lines of high-pressure contact are produced which are evenly distributed around the sealed areas. For a completely leakproof seal against high pressure, it is necessary that the two ends of the tube be joined together by welding, brazing, or soldering. This development successfully provides seals between high-temperature ovens and large pipe flanges.

Isolation Valve

A 22-inch-diameter valve was required at Ames Research Center to isolate air at 2000° F. This valve is used as an isolation valve in a wind tunnel when it is necessary to close off the heated section for access to other parts. No industrial supplier could be found to supply this size of valve for this temperature and pressure requirement. Figure 110 illustrates the custom design (ref. 23). The valve is water cooled and is made of nickel-plated steel; however, should a second valve be needed, stainless steel would be used. The valve is normally operated across a zero-pressure difference, but was designed to open at pressure drops up to 400 psig. Since the line pressure is 2000 psi, special procedures must be employed to operate the valve.

High-Temperature Gate Valve

At Ames Research Center, a gate valve operating in a 4000°–5000° F environment was troubled by welding of the gate and its base surface. The rapid sliding motion (approximately 5-millisecond duration) during valve closure aggravated the already serious materials problem. Conventional gate valves, which incorporate metal (e.g., brass, copper, stainless steel) gates and bases, required replacement after each closure.

The solution, illustrated in figure 111, was

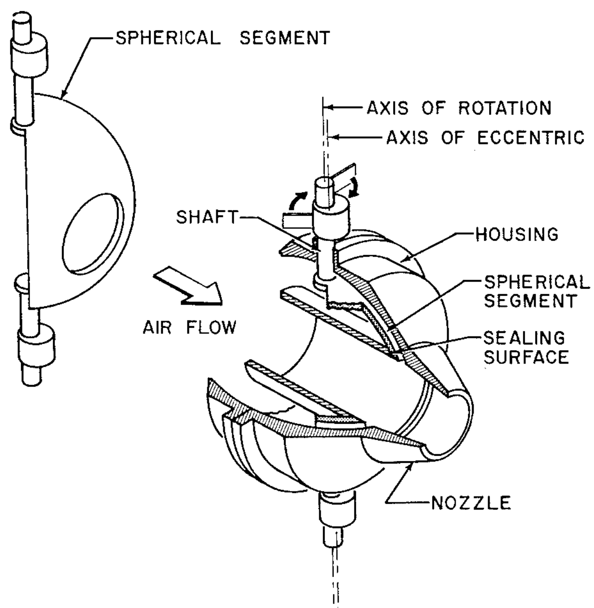

FIGURE 110.—High-temperature, high-pressure isolation valve.

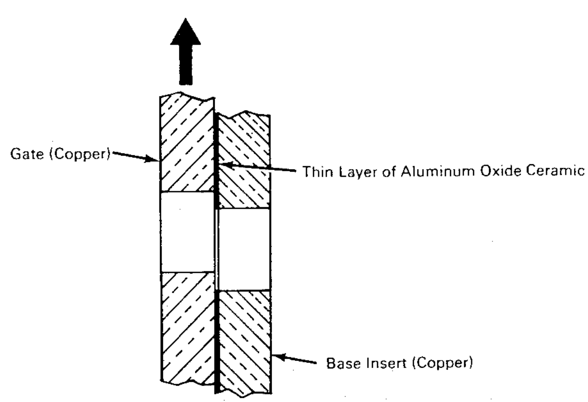

FIGURE 111.—High-temperature gate valve.

a renewable base insert of copper coated with a thin layer of aluminum oxide ceramic on the surface that comes into contact with the gate (ref. 24). The base insert is coated by spraying with a preparation of alumina (activated, powdered, catalyst grade AL–0102 P–98 percent Al_2O_3) to a depth of about 0.008 inch. The coating is then ground down to 0.005 inch. This insert is positioned with the aluminum oxide surface facing the surface of a copper gate in the valve.

The thin film of aluminum oxide ceramic has a negligible effect on the heat-sink characteristics of the copper base and gate, and prevents welding of the surfaces during sliding action in several 5-millisecond closures of the valve at air temperatures of 4000° to 5000° F. The thin ceramic coating is highly resistant to cracking and chipping, and requires replacement only after it has deteriorated as the result of ablation. To replace the coating on the insert, the base insert is removed from the valve, the worn-out surface is ground away, and the insert is resurfaced with aluminum oxide. This coating can be advantageously applied to any metal surfaces which come into intimate sliding contact in high-temperature environments.

SPRINGS

A spring is a common valve component that can be used to introduce damping into a system. Vibration can result in the malfunction of a valve or a valve control system. For example: an unsupported ball (as in a ball check valve) tends to flutter when off its seat during flow. The flutter causes squeal and impact damage to the seat. The design illustrated in figure 112 prevents flutter by the use of a star spring to hold the ball on its seat (ref. 25).

Any tendency of the ball to flutter is damped by the friction of the fingers on the inner face. A baffle is used to direct the flow around the spring fingers in cases where the dynamic pressure would be great enough to damage the fingers. The pressure drops through the baffle exert an additional force on the spring finger contact. Thus, additional damping is automatically provided and is proportional to the flow.

Belleville springs were used in the Mariner Mars pneumatic controller, and also have proved quite useful in other valve designs. Basically, a Belleville spring is a ring that is stamped from sheet material and is dished to the shape of a shallow cone. It is used where deflections are small. By stacking these springs in parallel, high load capacity can be attained (ref. 6).

A Belleville spring parallel assembly is shown in figure 113. The springs are retained in grooves machined in tubes. The inside tube is axially sliced into three segments, and the outside tube is sliced into two segments to permit the installation of the springs. The segmented retainers are held in place on the springs by an inner shaft and an outer tube. Some of the springs can be removed to reduce the spring force without changing the spring package configuration. The spacing of the springs, so that each carries its own load only, results in a spring package that has minimum hysteresis.

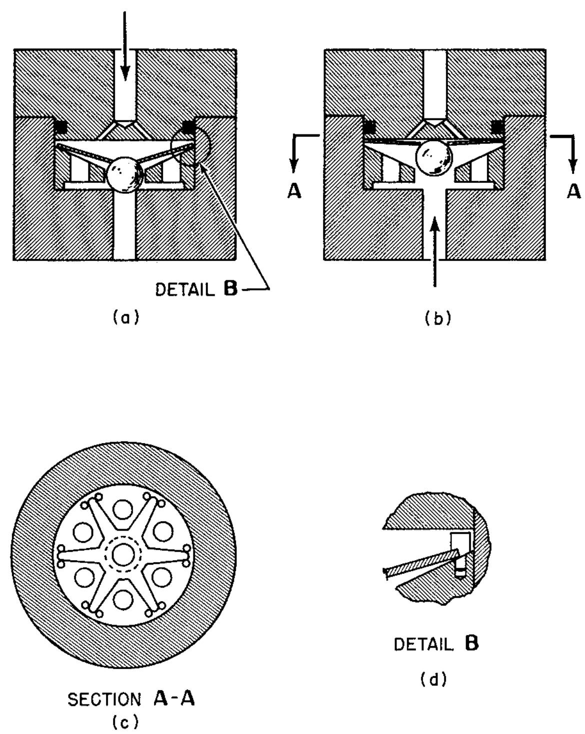

FIGURE 112.—Antiflutter valve design.

The load-deflection characteristics of a Belleville spring are, of course, important design factors. Figure 114 illustrates typical behavior. The spring is deflected by pressing through the flat position to a conical position in the opposite direction while being supported at the inner and outer peripheries. The load-deflection curve is dependent upon the cone angle. When the H/T ratio (axial height of dish/material thickness) is 1.5, there is a flat spot in the upward sweep of the load-versus-deflection curve so that the load remains constant while deflection continues. When the H/T ratio is greater than 1.5, the load-deflection curve rounds over and proceeds downward before resuming the upward trend. The negative, or back slope, portion of the curve (see fig. 114) is

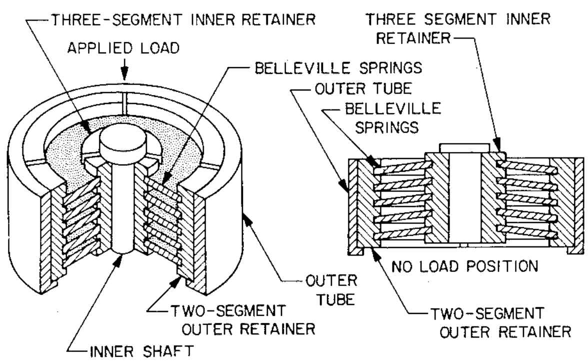

FIGURE 113.—Belleville spring package.

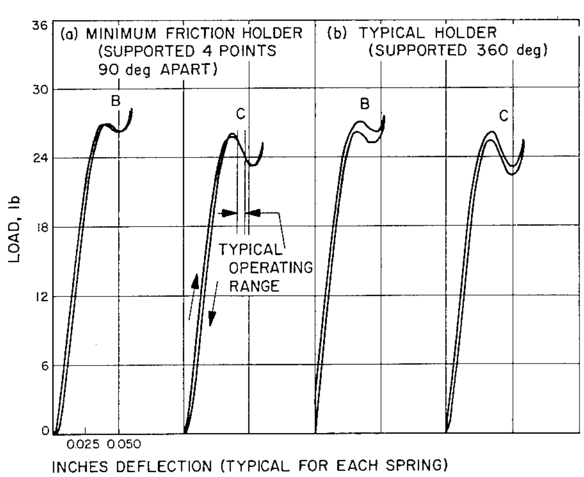

FIGURE 114.—Influence of Belleville spring support on the hysteresis curve.

the most desirable operating range for controllers. By actual measurement of a Belleville spring $1\frac{9}{16}$-inch OD $\times$ 1-inch ID $\times$ 0.020 inch thick, dished 0.030 inch, the OD increases 0.0033 inch and the ID decreases 0.0033 inch when the spring is deflected through its flat position. This radial growth produces sliding friction between the spring and the holder and accounts for much of the hysteresis shown in figure 114. If the spring is supported on long flexible columns (every 90° on both OD and ID), the friction is removed elastically. Figure 114 shows the hysteresis reduction when the same two springs are elastically supported. This Belleville spring was made of AISI 6150 spring steel, heat treated to Rockwell C 45–47 hardness, and silver plated for corrosion resistance and dry lubrication.

Even though the springs from a batch appear to be identical, the hysteresis (load-deflection) curves differ as shown in figure 115. The springs were supported in the flexible holder mentioned in figure 114(a) and were deflected in a Baldwin Mark G press. The hysteresis curves were recorded on a modified Moseley X–Y plotter.

A spring package prepared from a matched set of individual springs will have the same load-deflection curve as the individual springs if the load scale on the plotter is changed to give the same height. The steepness of the back slope can be reduced by random mixing of the springs comprising the package. Alternatively, the steepness can be reduced by using an inside or outside spacer between two matched sets in parallel so that one set picks up the load before the other. Figure 116 shows the effect on the hysteresis curve when a group of matched Belleville springs is placed in two packages and deflected, while one package is out of phase with the other by amounts of 0 to 0.020 inch in 0.005-inch increments. An out-of-phase increase produces a decrease in the back-slope spring rate (decrease of steepness). Both back-slope load extremes tend toward the mean back-slope load, which remains essentially unchanged.

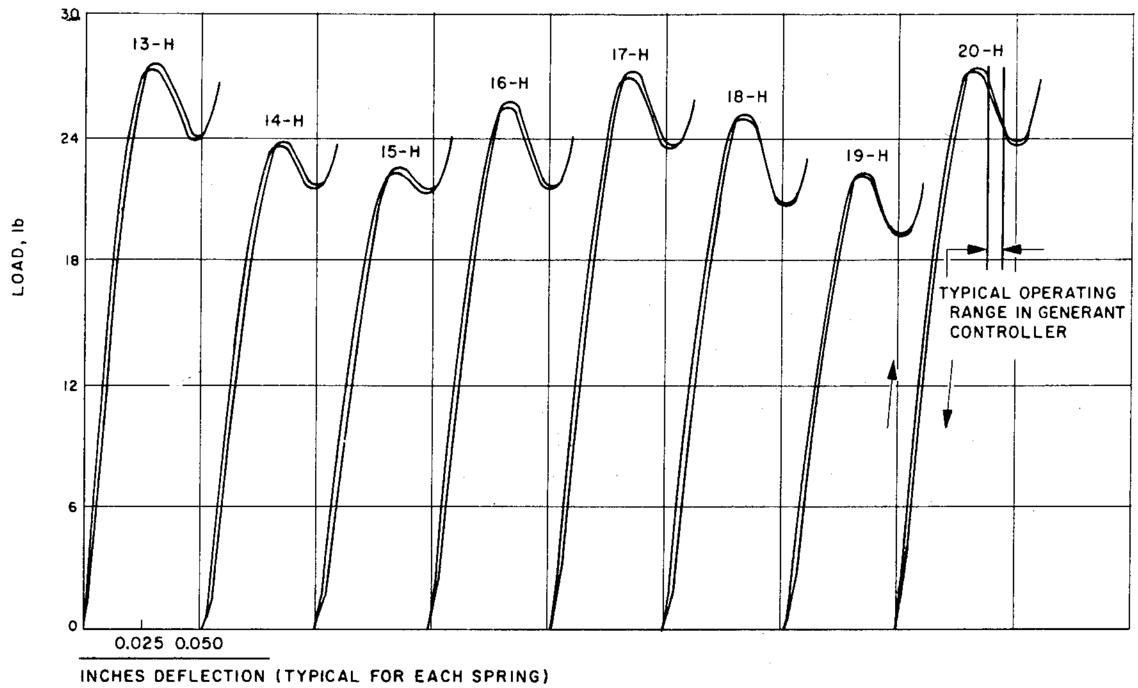

FIGURE 115.—Typical variation of load versus deflection for Belleville springs. (If a dual parallel package consisting of matched springs (such as 13-H, 17-H, and 20-H in one unit and as 16-H and 18-H in the other) are placed out of phase, the back slope of the composite load versus deflection curve will be reduced (slope similar to that of 15-H).

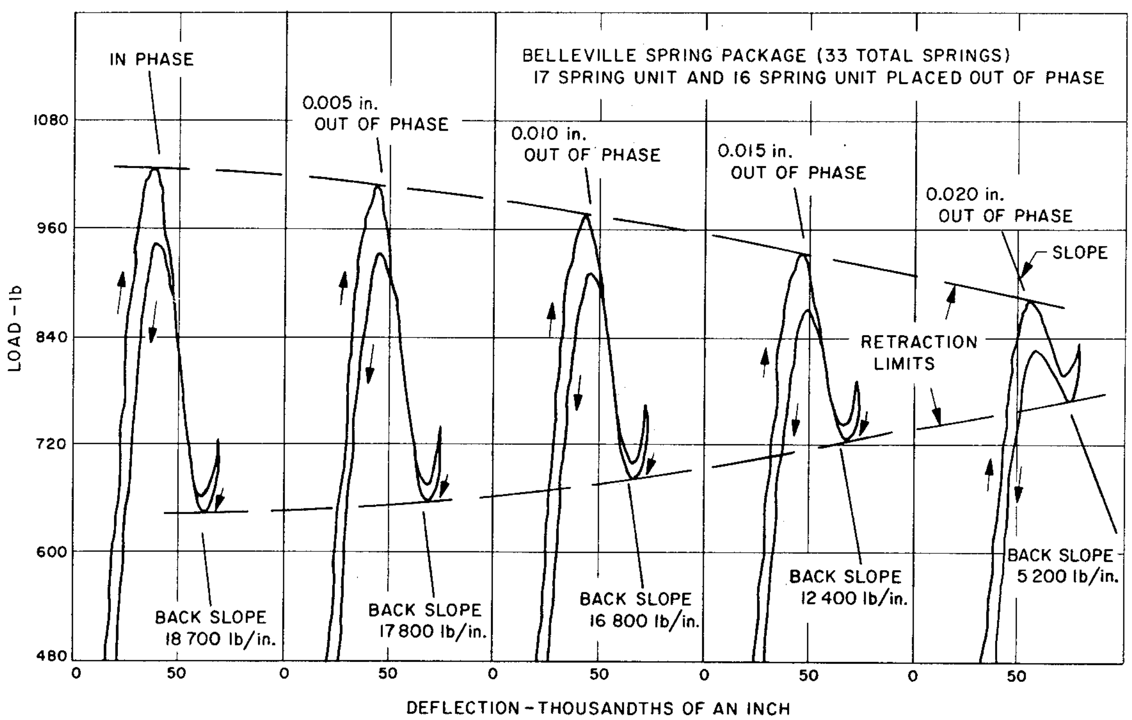

FIGURE 116.—Effect on slope and retraction limits upon Belleville springs are out of phase.

Materials suitable for Belleville springs are, in addition to the AISI 6150 spring steel already mentioned, stainless steels (e.g., AISI 302 and AISI 34), AISI 1095, AISI 420, 17–4PH, and 17–7PH.

FLUIDIC DEVICES

Fluidic devices are receiving increased attention for the performance of valving functions and as components for valve-control systems because they have no moving parts and, therefore, offer advantages in reliability and maintenance. A fluidic device is also tolerant of severe environments such as nuclear radiation, high temperature, and shock and vibration.

Flueric Servo Valve

The Bendix Corp. is developing a high-performance, pneumatic-input, four-way flueric servo valve with dynamic load pressure feedback (ref. 26). It is intended to function as a position servo for the control drum of a nuclear-reactor-powered rocket.

Design conditions call for operation with hydrogen at temperatures ranging from 100° to 600° R, functioning with a supply pressure at 215 psia and an exhaust pressure of 50 psia, and having a maximum control pressure of 70.4 psia. One of the components, a Venjet amplifier, has been built, and a small vortex valve has been tested. Development work is still required.

Hot Gas Jet Reaction Valve

Fluid amplifier techniques for the control of hot gases were explored by the Army Inertial Guidance and Control Laboratory (ref. 27). Valves based on fluid-controlled, bistable amplifier principles were successfully tested with gases at pressures up to 1300 psi and temperatures up to 2350° F; flow rates up to 1 pound per second of cold air were also used. The input to the valves was a pulse-duration modulated signal with 25-cps carrier frequency.

QUIET VALVES

Throttling valves are known to contribute to submarine noise. The U.S. Navy Marine Engineering Laboratory conducted a study of throttling designs in an effort to reduce valve-generated noise (ref. 28). The report explains methods of fluid-flow regulation and noise characteristics and gives experimental results. Considerable attention is given to frictional throttling. In one design, the fluid is passed through a large number of parallel passages. In another design, flow is directed through a porous plug. An experimental valve with a sintered metal plug, providing a variable-length throttling section, exhibited excellent noise and flow characteristics.

EXPLOSIVE-ACTUATED VALVES

Explosive devices are widely used in the aerospace industry. They have a high power-to-weight ratio, are very reliable, and can deliver energy more quickly than any other controllable source. The energy can be released by very modest electric currents. A review of commercial and aerospace explosives compiled by Romaine describes many valves (ref. 29).

An explosive-actuated valve, referred to as the Majeski valve in chapter 12, was developed at Ames Research Center (ref. 30). This valve shuts off a high-pressure (4600 psi), high-temperature (10 000° F) gas flow in 6 to 8 milliseconds.

Figure 117 shows this design. The valve has a piston that is inserted with a light

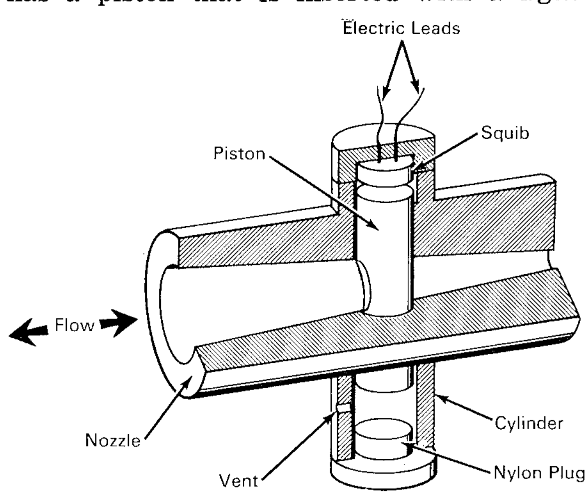

FIGURE 117.—Majeski valve schematic.

interference fit into a cylindrical bore extending transversely through the flow nozzle. The piston has a radial hole which is concentrically alined with the axis of the nozzle to provide unobstructed flow when the valve is in the open position. The squib is mounted at a small standoff in a cap above the top of the piston. When the flow is to be shut off, the squib is initiated from a remote voltage source. The detonation drives the piston down the cylinder until it is stopped by the nylon plug. In this closed position, the piston provides a tight seal against the gas flow.

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CHAPTER 14

Industrail Valves: The Future

Trends revealed by a review of developments up to the present time can serve as the basis for extrapolation into the future.

REVIEW OF DEVELOPMENTS

Valves are becoming more specialized because of the increasingly extreme conditions under which they must operate. Internal pressures of interest now range from the low values of an extreme vacuum to the high values of 40 000 psi or more that are encountered in the petrochemical industry. Fluid temperatures currently of interest range from cryogenic temperatures as low as $-452°$ F for liquid helium to greater than $2000°$ F for exhaust gases from rocket engines and jet engines for the supersonic transport (SST). Response time and repeatability have become more important than was previously the case because of increased attention to the control of mixing, shutoff, and injection operations. These diverse, and often contradictory, specifications have led to specialization of design.

Materials previously unknown or unavailable have been incorporated into valves. Teflon, Kel-F, and other plastics and elastomers are now commonly used and, for example, have improved the practicality of the ball-valve design. Metals such as titanium and aluminum have become available and are being used in valves. Fiber reinforcement and dispersion strengthening are being explored as techniques to increase the elevated temperature strength and stability of metals and alloys. Material properties have been more accurately and quantitatively determined. Structures now can be developed by design more than by test, but the desired property information still appears to be incomplete even for many common materials.

Leakage in the shutoff position has become a major concern in recent years because fluids and processes are becoming more costly and more critical with respect to contamination. Detailed studies of valve-seating phenomenon are yielding information pertinent to leakage mechanisms. These studies show promise of generating design information, particularly for metal-to-metal seats. Leakage-testing procedures and instruments, too, have been refined and standardized to a greater extent. Hence it is now possible to evaluate a valve more completely.

Valve *reliability* has become of increasing importance. The temperatures and pressures encountered in the process and aerospace industries challenge the structural integrity of valve assemblies. Procedure and techniques to determine component reliability by testing samples and to evaluate a design's potential reliability are under development. The prime contribution has been one of organization. The process of failure-mode identification, identification of consequences of a given failure mode, and data collection has been fruitful for electrical components such as resistors. Some recent attempts at "super-component" valve development have attained modest success.

New concepts for control of fluid flow have emerged. Fluidic devices utilizing, for example, the Coanda effect have no moving mechanical parts and are tolerant of severe environments. A diverting valve using fluidic principles is now commercially available. The effects of a nonuniform electric field on an uncharged flowing fluid or of a uniform electric field on a charged fluid have been investigated. Fluidic devices using the first of these effects appear to have merit. The electrokinetic effect and other unconventional means of influencing a fluid have been studied.

Automation has developed great potential for reducing plant operating costs and for process control. Many of the computers necessary for this automation are digital, and actuators capable of converting a digital signal from a computer into a suitable valve positional command are being developed.

THE FUTURE

Valves, like wheels, have a long history. They are likely to be improved by refinement rather than by drastic innovation. Design information will be generated from detailed seating studies which will lead to improvements in leakage characteristics. A further result of these studies will be seating designs that are remarkable for their ability to "eat contaminants"; i.e., to seal despite contamination. Knowledge of material properties and increased use of high-speed computers will permit valve designs to be evaluated more extensively and such critical items as tolerances to be specified accurately. Valve reliability will improve both in terms of structural integrity and from a functional viewpoint as methods now evolving are adapted and perfected. Further, data on such pertinent topics as mean time to failure will become readily available.

Specialization of valve design will increase. Specialization will be encouraged by the availability of materials and manufacturing processes that, although expensive, will offer unusual properties. Composite materials, fiber reinforcement, and dispersion strengthening will permit better use of presently known materials. Single crystals may be grown from which valve bodies will be machined. Detailed design information, increased application of analytical techniques and computers, and development of unconventional means, perhaps electrical in nature, for fluid-flow control will further encourage specialization.

Valves for light duty (bathroom faucets, for instance) will be replaced at the first sign of trouble. The advent of plastics or similar materials plus a sound understanding of their sealing capabilities will permit metals to be eliminated from these valves and permit their economical manufacture.

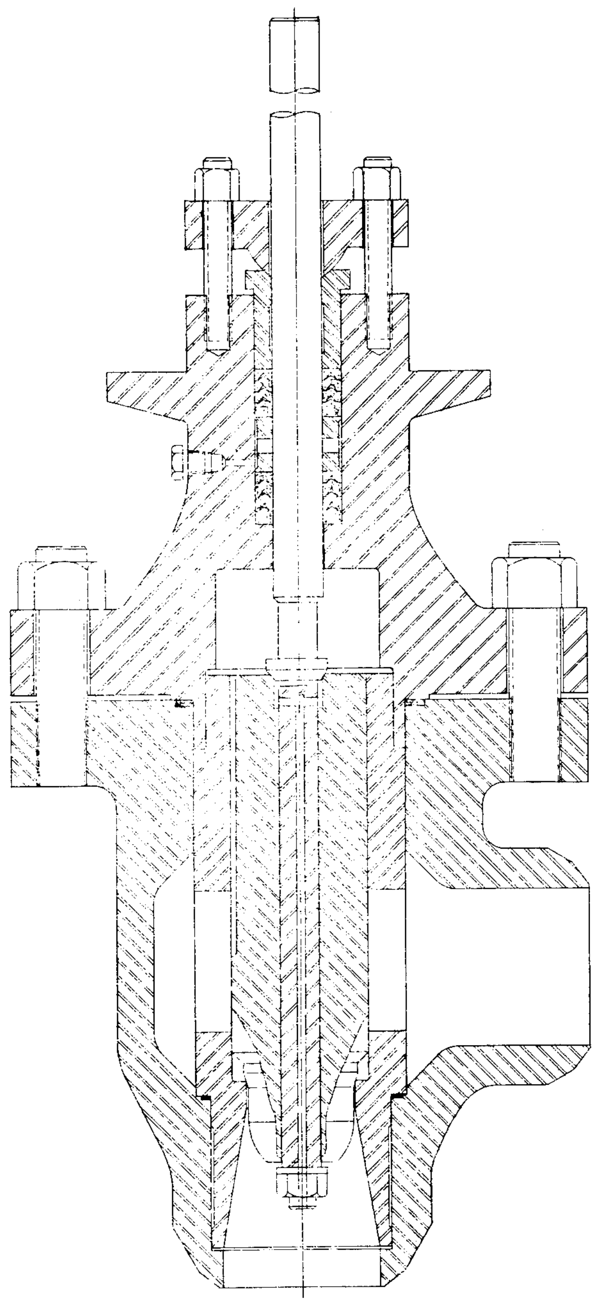

FIGURE 118.—A valve of the future.

Control valves for service in automated processes will be highly developed. The actuators are likely to be integral with the valve in contrast to many present patched-on approaches and will accept digital signals. Electrical and fluidic devices will be commonly used in the actuators and instrumentation; fluidics will be of great utility in high-temperature or radiation environments.

While specific predictions are dangerous, they are particularly interesting. In this vein, a forecast by Ralph Renouf (ref. 1) of a control valve design for the near future merits attention.

Requirements of a low maintenance cost, a wide choice of materials for seats and inner valves, and a capability for flushing the lines for many processes have stimulated widespread interest in the cage-type valves. The valve of the future is possibly the cage-type valve, with modifications not now conceived. Figure 118 is a composite of a valve to combat the abuse the control processes will administer. The valve body will be a forging with weld end connections to guarantee soundness of material and to eliminate leakage. The angle design will protect the valve body from impingement and erosion, while providing a maximum flow rate for a given valve size and a given set of operating conditions. The cage-type trim will be used with pressure-sealing gaskets to allow easy removal of all valve parts for rapid maintenance or replacement. A single-seat design will be used for minimum-seat leakage. The seating surfaces will be protected from "wire drawing," and the inner valve will be balanced for easy operation. The cage-type trim will consist of a "clearance flow" design on the inner valve further protected by a "clearance flow" design of the inner valve in the seat cage. As the inner valve enters the "clearance flow" portion of its travel, the guide surface of the inner valve will begin to eliminate flow to the inner valve. Thus, the seating surfaces will be protected from erosion. The cage trim will be deliberately designed to allow leakage around the inner valve to the chamber in the closure above the inner valve. A poppet-type relief valve will be designed into the upper end of the inner valve. As a signal is received from the instrumentation to open the valve, the valve operator must only overcome the unbalanced force on the poppet valve to relieve the pressure holding the process inner valve closed. As long as the relief flow exceeds the leakage around the inner valve, the unbalanced forces will be eliminated, and the inner valve may be opened easily. There are many possible derivatives of this basic design principle, and many of them will be prominent in industrial valves of the future.

REFERENCE

1. Renouf, R.: "Industrial Valves—The Future." Paper presented at the Valve Technology Seminar, Midwest Research Institute, Kansas City, Mo., Oct. 12–22, 1965.

Appendix A. Glossary of Valve Terms

Accumulator—A fluid-pressure storage chamber in which fluid pressure energy may be accumulated and from which it may be withdrawn.

Actuator—A device to convert control energy into mechanical motion.

Air bleeder—A device used to remove air from the high point in a circuit. It may be a needle valve, capillary tubing to the tank, or a bleed plug.

Ambient temperature—The temperature of the surrounding environment.

Amplifier—A device used to increase volume rather than pressure. The opposite of a pressure intensifier.

Asperities—Rough places.

Choke—A restriction which is relatively long with respect to its cross-section dimension.

Clarifier—A device for removing deleterious solids and assisting in maintaining the chemical stability of hydraulic fluid.

Cold welding—The mechanical bonding of two similar or dissimilar materials in a vacuum environment.

Control—A device used to regulate the functions of a machine.

Control, liquid-level—A device which controls the liquid level by a float switch or other means.

Control, mechanical—A control actuated by linkages, gears, screws, cams, or other mechanical elements.

Control, pneumatic—A control actuated by air pressure.

Control, servo—A control actuated by a feedback system which compares the output with the reference signal and makes corrections to reduce the difference.

Cushion, hydraulic—A cushion in which a hydraulic cylinder provides the resistance. Pressure in the cylinder is developed by the mainram movement. The cushion is returned to its normal position hydraulically.

Cushion, hydropneumatic—A cushion in which a hydraulic cylinder provides the resistance. Pressure in the cylinder is developed by the mainram movement. The cushion is returned to its normal position by air pressure acting on the hydraulic fluid in the reservoir.

Cushion, pneumatic—A cushion in which an air cylinder provides the resistance.

Cycle, automatic—A cycle of operation which, once started, is repeated indefinitely in a predetermined sequence until stopped.

Cycle, semiautomatic—A cycle which is started upon a given signal, proceeds through a predetermined sequence, and then stops with all the elements in their initial position.

Cylinder—A linear-motion device for converting fluid energy into mechanical energy (or vice versa) in which the thrust or force is proportional to the effective cross-sectional area.

Cylinder, double-acting—A cylinder in which fluid force can be applied in either direction.

Cylinder, double-end-rod—A cylinder with two rods, one extending from each end.

Cylinder, piston-type—A cylinder in which the internal element is of one or more diameters and the seal is of the expanding type.

Cylinder, plunger type—A cylinder in which the internal element is of a single diameter and upon which the seal applied is of the contracting type.

Cylinder, single acting—A cylinder in which the fluid force is applied in only one direction.

Damper—A device used to restrict the amplitude of a shock wave.

Dwell—The portion of the cycle in which feed or pressure stroke is stopped.

Face-centered metals—An arrangement of atoms in crystals which may be initiated by packing spheres. The atomic centers are disposed in space in such a way that they may be supposed to be situated at the corners and the middle of the faces of a set of cubic cells.

Feed—The portion of the cycle in which the work is performed on the workpiece.

Filter—A device for the removal of solids from a fluid wherein the resistance to motion of such solids is in a tortuous path.

Fluid—A substance which yields and suffers indefinite distortion due to any pressure tending to alter its shape. Fluids include both liquids and gases.

Fluid absolute viscosity—The force in dynes required to move a plane surface of 1 sq cm over another plane surface at the rate of 1 cm/sec when the surfaces are separated by a layer of fluid 1 cm in thickness. The unit is known as the poise. Since absolute viscosity is difficult to determine, the viscosity usually is expressed as Saybolt Universal Seconds (SSU), which is the time in seconds for 60 cc of oil to flow through a standard orifice at a given temperature.

Fluid flash point—The temperature at which a fluid first gives off sufficient flammable vapor to ignite when approached by a small flame or spark.

Fluid oxidation—A chemical breakdown of a fluid, causing the formation of oxidation products, which in turn cause emulsification, foaming, and the deposition of sludge.

Fluid SAE viscosity numbers—The arbitrary numbers for classifying fluids according to their viscosities. The numbers in no way indicate the viscosity index of the fluids.

Fluid specific gravity—The ratio of the weight of a given volume of fluid to the weight of an equal volume of water.

Intensifier—A device which increases the working pressure over that delivered by a primary source.

Joule-Thompson effect—Simply stated, the effect of decreasing temperature accompanying the expansion of a fluid.

Joule-Thompson effect, reverse—The effect of increasing temperature accompanying the expansion of a fluid.

Langmuir equation—$G = P/17.14 \sqrt{M/T}$, where G = rate of loss per unit area of exposed surface in g/sec/cm^2; P = vapor pressure at temperature T in mm Hg; M = molecular weight of the metal in the gas phase; T = absolute temperature (°K).

Line—A tube, pipe, or hose which acts as a conductor of fluid.

Line, exhaust—A return line which carries power or control-actuating fluid back to the reservoir.

Line, pilot—A line which acts as a conductor of control-actuating fluid.

Line, working—A line which acts as a conductor of power-actuating fluid.

Orifice—A restriction which is relatively short with respect to its cross-section dimension.

Outgassing—The evolution of gas from a solid in a vacuum environment.

Passage—A machine or cored connection which lies within or passes through a hydraulic component and which acts as a conductor of hydraulic fluid.

Poppet—A mushroom or tulip-shaped valve consisting of a circular head with a conical face.

Port—An opening at a surface of the component; e.g., the terminus of a passage. It may be internal or external.

Port, valve—A controllable opening between passages; i.e., one which can be closed, opened, or modulated.

Positive position stop—A structural member which definitely limits the working motion at a desired position.

Positive safety stop—A fixed structural member which confines maximum travel within the design limits of the machine or equipment.

Pressure, back—The pressure encountered on the return side of a system.

Pressure, operating—The pressure at which the system is operated.

Pressure, suction—The absolute pressure of the fluid at the inlet of a pump.

Pressure head—The pressure resulting from the height of a column or body of fluid, expressed in feet.

Pump, centrifugal—A power-driven device for converting mechanical energy into fluid energy, having an impeller rotating into a volute housing with liquid carried around the periphery of a housing and discharged by means of centrifugal force.

Pump, gear—A power-driven pump having two or more intermeshing gears or lobed members enclosed in a suitably shaped housing.

Pump, screw—A power-driven pump consisting of one or more screws rotating in a housing.

Reservoir—A chamber used to store fluids. Pumps, motors, and valves may be mounted on the reservoir.

Restriction—A device which produces a deliberate pressure drop or resistance in a line or passage by means of a reduced cross-sectional area.

Reverse Joule-Thompson effect—The effect of increasing temperature accompanying the expansion of a fluid.

rms—A measure of surface roughness; the root-mean-square variation.

Sabot—A fixture for temporarily holding an object in a particular attitude.

Soluble oil—A substance added to water to provide lubricating qualities and inhibit corrosion.

Specific impulse—The total impulse of thrust provided by the combustion of a unit mass of propellant. It is proportional to the combustion temperature of the fuel and inversely proportional to its molecular weight. Since hydrogen has the lowest molecular weight, it has a correspondingly high specific impulse.

Spool—A device within a valve for changing the flow from one port to another port.

Squib—An explosive device used to initiate an event or action.

Surge—A transient rise of hydraulic pressure in the circuit.

Trip device—A mechanical element for the actuation of a position control.

Ullage—The amount which a vessel lacks of being full.

Valve, cam-operated—A valve on which the spool is positioned mechanically by a cam.

Valve, center-pressure—A valve which in the center position connects the supply to working ports only.

Valve, check—A valve which permits flow of fluid in one direction only and self-closes to prevent any flow in the opposite direction.

Valve, counterbalance—A valve which maintains resistance against flow in one direction but permits free flow in the other. It is usually connected to the outlet of a double-acting cylinder to support its weight or to prevent uncontrolled movements.

Valve, directional—A valve which selectively directs or prevents fluid flow through desired channels.

Valve, flow-dividing—A valve which divides the flow from a single source into two separate branches of a circuit at a constant ratio regardless of the difference in pressure between the two branches.

Valve, four-way—A valve having four controlled working passages, usually ending in four external ports. A four-way valve usually has one inlet port and three outlet ports.

Valve, gate—A valve used to start, stop, or limit the flow of fluid. It controls the flow by means of a gate, which is raised or lowered by the action of a screw or other means to close or open the passage.

Valve, globe—A valve used to start, stop, or limit the flow of fluid. It controls the flow by means of a plug, a ball, or a disk, which by action of a screw or other means is pulled away from or lowered into a corresponding seat.

Valve, needle—A valve used to start, stop, or limit the flow of fluid. It controls the flow by a tapered needle, which is pulled away from or forced into a corresponding seat. The tapered needle permits gradual opening or closing of the passage.

Valve, pilot—A small directional-control valve generally used for operating other valves.

Valve, pilot-check—A check valve provided with a piston to unseat the check poppet when pilot pressure is applied.

Valve, pilot-operated—A valve which is positioned by pilot fluid pressure.

Valve, poppet-type—A valve construction which closes off flow by a poppet seating against a suitable seating material. Normally considered a dead-tight seal. The poppet may be a ball, a cone, or a flat disk.

Valve, pressure-reducing—A valve which maintains a reduced pressure at its outlet regardless of the higher inlet pressure.

Valve, relief—A valve which limits the maximum pressure which can be applied to the portion of the circuit to which it is connected.

Valve, safety—A poppet-type two-way valve intended to release fluid to a secondary area when pressures approach the maximum set value.

Valve, shuttle—A valve with three ports and a floating piston between two of the ports moving in a horizontal plane. Pressure entering either of these two ports will shift the piston blocking the opposite port and directing fluid to the third port.

Valve, spool-type—A valve construction using a spool consisting of machined undercuts or recesses on a cylinder of metal. The spool is fitted to a bore containing annular undercuts. Movement of the spool in the bore connects ports uncovered by the spool undercuts. Clearance flow is usually necessary to insure free spool movement.

Valve, standard-action—A valve which is positioned by manual, mechanical, or pilot means without springs or detents.

Valve, surge-damping—A valve which prevents shock by controlling the rate of acceleration of fluid flow.

Valve, three-way—A valve having three controlled working passages, usually ending in three external ports. A three-way valve usually has one inlet port and two outlet ports.

Valve, time-delay—A valve in which the change of fluid occurs only after a desired time interval has elapsed.

Valve, two-way—A valve having two controlled working passages, usually ending in two external ports. A two-way valve has one inlet and one outlet port.

Appendix B. Bibliography

In addition to references given at the ends of some chapters, readers may find the following works helpful.

CHAPTER 1. INTRODUCTION

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BRADY, B. P.; AND SALVINSKI, R. J.: Advanced Valve Technology for Spacecraft Engines. Final Report, Report No. 8651-6016-RU-000, TRW Space Technology Labs., Inc., Redondo Beach, Calif., Mar. 1963.

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CHAPTER 5. CONTAMINATION

HOWELL, G. W., ED.: Aerospace Fluid Component Designers' Handbook, Vol. 1. RPL TDR-64-25, TRW Space Technology Labs., Redondo Beach, Calif., May 1964.

BRADY, B. P.; AND SALVINSKI, R. J.: Advanced Valve Technology for Spacecraft Engines Report no. 8651-6016-RU-000, TRW Space Technology Labs., Inc., Redondo Beach, Calif., Mar. 1963.

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CHAPTER 10. VALVES FOR EXTREME PRESSURES AND TEMPERATURES

BLOW, C. M.: The Development and Testing of Elastomeric Materials for Fluid Sealing Applications. Second International Conference on Fluid Sealing, British Hydromechanics Research Association, Cranfield, England, Apr. 1964, p. H1–1.

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CHAPTER 14. INDUSTRIAL VALVES: THE FUTURE

RHODES, A. F.: Trends in Valving. Pipeline Engineer, vol. 37, no. 10, Sept. 1965, p. 43.

Appendix C. Abstracts of Recently Published Valve Guides

by JOHN B. LOSER

This appendix is a series of abstracts of recently published guides that valve engineering specialists consider extremely useful. The guides, representing hundreds of pages, are not known to all valve designers, application engineers, and users, since distribution of the parent publications was generally limited to specific industries.

A Course in Hydraulic Valves (Parts 1, 2, and 3), Hydraulics and Pneumatics, Feb., Apr., and July, 1961, pp. 55–63, 69–77, and 73–78, respectively.

This series of articles explains basic design and performance characteristics and presents important factors in selecting the proper valve for a particular application. Each part contains symbols and schematics of the various types of valves discussed.

Part 1—"Directional Control Valves," includes porting, spool valves, packed plunger, rotary, poppet, sliding plate, check valves, and valve actuators.

Part 2—"Pressure Control Valves," includes relief, pressure reducing, sequence, counterbalance, unloading, and pre-fill-sequence valves.

Part 3—"Flow Control Valves," includes needle and globe valves, fixed flow and bypass flow regulators, adjustable flow controls, integral check valves, built-in overload relief valves, temperature compensation, decelerating valves, and metering circuits.

BRADY, B. P.; AND SALVINSKI, R. J.: Advanced Valve Technology for Spacecraft Engines, Space Technology Laboratories, Inc., Contract no. NAS7–107, Final Report, Mar. 1963 (357 p.). (Available from Clearinghouse for Federal Scientific and Technical Information, Springfield Va.—request document no. N63–15032)

A valve design study was made to determine what elements or design features should be included or omitted in the design of many valve types. Each type and the elements within it was analyzed for performance with propellants, space environments, and functional reliability. Technological advances were reported in detail. Tables C–1 and C–2 are reproductions from the report.

Design Criteria for Zero-Leakage Connectors for Launch Vehicles, Advanced Technology Laboratories, General Electric Co. for NASA, Marshall Space Flight Center, Contract no. NAS8–4012.

This detailed study, in five volumes, is slanted toward connectors, but contains information on advances applicable to valves and to zero leakage.

DIBARTOLO, E. A.: How To Select Flow-Control Valves. Fluid Controls, Inc., Machine Design, July 18, 1963, pp. 167–184.

The author discusses principles of operation, types of valves, valve capabilities, and application circuits. Major topics are: Mobile vs. Industrial, Flow-Control Principles, Circuit Types, Metering Methods, Pressure-Compensated Capabilities, and Proportional Flow Divider.

DODGE, L.: Fluid Throttling Devices for Controlled Flow Resistance. Products Engineering, Mar. 30, 1964, pp. 81–87.

This article deals with planned pressure drops, not unintentional pressure drops. In the relationship $H_L = KV^2 2_g$, for use in the basic formula $Q = AC\ 2_g\ H_L$, the author gives values of K for relief, gate, butterfly, ball, disk, needle, simple-ball, and four-way valves. Also included are K values for typical parts, notches in plungers, partial orifices, ramp slots, slotted sleeves, intersecting holes, and rotary notches, slots and wedges, and the formula for converting the flow coefficient C_V to K.

EDWARDS, T. W.: Valves. Power, June 1961, pp. 69–92.

This is a special report on factors in selection, recent advances in materials and fluid-loss technology, a study of fluid-end types, valve actuators available, and practical ideas on maintenance. The article is well illustrated and contains tables on material selection, ASTM materials specifications for pressure-temperature extremes, pressure-temperature ratings, and American Standards Association valve dimensions.

EVANS, F. L.: Valves. Hydrocarbon Processing and Petroleum Refiner, Oct. 1961, vol. 40, no. 10, pp. 121–136.

A discussion of valves used in industry, with emphasis on how to select the right valve for performance and economy. This is a general, yet thorough, article about industrial valves and includes hints for users.

HOWELL, G. W., ED.: Aerospace Fluid Component Designers' Handbook, vols. I and II. Technical Documentary Report no. RPL–TDR–64–25, TRW Systems, One Space Park, Redondo Beach, Calif., Contract No. AF04(611)–8385, Oct. 1965. (Available by request from AF Rocket Propulsion Laboratory, ATTN: Aerospace Fluid Component Designers' Handbook, Edwards Air Force Base, Calif., 93523; or from Defense Documentation Center, Cameron Station, Alexandria, Va.—request document no. AD 447 995).

This handbook is a useful reference for the valve designer, application engineer, and user. Revisions are prepared periodically.

Quick Guide to Product Selection. Hydraulics and Pneumatics, Jan. 1964, Vol. 17, no. 1, The Industrial Publishing Corp., 812 Huron Rd., Cleveland, Ohio, 44115 (16 pp. on valves).

This is an availability guide for types of fluid power products. Directional control, flow control, pressure control, and servo valves are listed by manufacturer. Information on type, design, use, operation, ports, maximum flow rating, and maximum pressure rating is given.

SALVINSKI, R.; FIET, O.; AND MERRITT, F.: Advanced Valve Technology for Spacecraft Engines. TRW Systems, One Space Park, Redondo Beach, Calif. Contract no. NAS7–107, Final Report No. 8651–6042–SU–000, Aug. 1965.

An updated valve chart was constructed for defining and evaluating problems related to propulsion valves in the space environment. On this chart a "reliability rating" is assigned to each valve and element in the valve assembly in relation to functional parameters and performance with propellants in space planetary environments. The chart (tables C–3 and C–4) has been reviewed and updated to reflect the latest information.

Special Valve Report. The Petroleum Engineer Publishing Co. ran a special report on valves in all five of its July 1963 publications. All five carried the articles, "Fundamentals of Valves" (8 p.) and a "Directory of Valve Products and Valve Manufacturers" (32 p.). Other articles were distributed among the publications as follows:

Petroleum Engineer
"What To Look for in Oilfield Valves" (10 p.)
"Helpful Hints for Valve Usage in Waterflooding" (5 p.)

Petro/Chem Engineer
"Valve Makers Face up to User Needs" (1 p.)
"Stainless Valves Success with Sludge Acids" (2 p.)
"When Does a Control Valve Cost Too Much?" (1 p.)
"Valve Design Overcomes Early Seat Failure" (1 p.)
"Nomographs To Speed Valve Selection" (5 p.)
"R&D Facility Aids Valve Design" (1 p.)
"New Patents Pertaining to Valves" (2 p.)

Pipeline Engineer
"How To Select Crude and Products Line Valves" (7 p.)
"Which Valve in Gas Pipelining?" (8 p.)
"The Good Business Role of Valve Standards" (3 p.)
"LPG Pipelining Has Its Special Valve Demands" (2 p.)

American Gas Journal
"Distribution Valving Today" (5 p.)

Standards. The Fluid Controls Institute, Inc., Post Office Box 1485, Pompano Beach, Fla., 33060, has a series of five "standards" available:

FCI 55–1: Standard Classification and Terminology for Power Actuated Valves (8 p.). 20 cents.

FCI 58–1: Definitions of Regulator Capacities (8 p.). 20 cents.

FCI 58–2: Recommended Voluntary Standards for Measurement Procedure for Determining Control Valve Flow Capacity (4 p.). 10 cents.

FCI 61–1: Recommended Voluntary Standards for Procedure in Rating Flow and Pressure Characteristics of Solenoid Valves (12 p.). 20 cents.

FCI 62–1: Recommended Voluntary Standard Formulas for Sizing Control Valves (8 p.). 20 cents.

The Commercial Application of Missile/Space Technology, Pts. 1 and 2, Denver Research Institute, University of Denver, Sept. 1963.

This study presents much information concerning aerospace-developed technology. A considerable amount is pertinent to valves.

WING, JR., PAUL: Limitations of Valve-Sizing Formulas. Instruments and Control Systems, Mar. 1963, vol. 36, pp. 131–135.

This article discusses several basic limitations of the standard working formulas for valve sizing.

Reliability ratings assigned to various combinations, components, and parameters have the following definitions:

Rating	*Definition*
1	A serious problem exists for which there is no satisfactory solution.
2	A problem exists, but a remedy may be available.
3	Satisfactory; i.e., within the state of the art.
U	Necessary information upon which to base a judgment was unavailable.
N	Parameter is not applicable.

In some cases, as indicated on the chart, a differentiation in values was made between manned and unmanned vehicles.

TABLE C-1.—*Liquid Propellant Valves for Spacecraft Rocket Engines*

Idealized Design Selection Chart

	Performance with propellant				Performance in environments							Functional reliability							Remarks
	Pressurants to 5000 psia	Liquid propellants	Metallized gels	Nonmetallized gels	Hi-temperature 1000°F	Low temperature −159°F	Radiation	Vacuum	Zero-g	Meteoroids	Sterilization	Operating life 1000 cycles	High response	Sticking	Manufacturing tolerances	Zero leak	Contamination sensitive	Space maintenance	
Valve closure:																			
Polymeric seal	X	O	O	O	O	X	O	O	X	NA	O	X	NA	O	X	X	X	U	Design techniques or new concepts required
Metal to metal seal	O	X	O	U	X	X	X	O	X	NA	X	X	NA	X	O	O	O	U	
Static seals:																			
Polymeric	X	O	O	O	O	X	O	O	X	NA	O	NA	NA	NA	X	X	X	U	Use metal seals only
Metal	X	X	X	X	X	X	X	X	X	NA	X	NA	NA	NA	X	X	X	U	
Dynamic seals:																			
Polymeric	X	O	U	O	O	X	O	O	X	NA	O	X	O	O	X	X	X	U	Use no dynamic seals
Metal	O	X	U	X	X	X	X	O	X	NA	X	X	X	X	O	O	O	U	
Hermetic seals:																			
Bellows	O	X	X	X	U	X	X	X	X	O	X	X	X	NA	X	X	X	U	Use: (1) Metal diaphrams for small displacements)
Metal diaphragms	X	X	X	X	X	X	X	X	X	·O	X	X	X	NA	X	X	X	U	(2) Use metal bellows (for large displacements and low pressure gradients)
Polymeric diaphragms	O	O	O	O	O	X	O	O	X	O	O	X	X	NA	X	X	X	U	Provide meteoroidal shielding

Lubes:																			
Dry	X	O	O	O	X	X	U	U	X	X	U	X	X	X	X	NA	O	U	Use no lubricants
Liquid and grease	X	O	O	O	O	O	O	O	O	X	O	X	X	X	X	NA	X	U	
Moving parts:																			
Small clearances	O	O	O	O	O	O	X	O	X	NA	O	O	O	O	O	NA	O	NA	Use: (1) Large clearances only (2) Flexure pivots
Large clearances	X	X	X	X	X	X	X	X	X	NA	X	X	X	X	X	NA	X	NA	
Ball bearings	X	X	O	X	O	X	X	O	X	NA	X	X	X	X	O	NA	X	U	
Linear ball drives	X	X	O	X	O	X	X	O	X	NA	X	X	X	X	O	NA	X	U	Ball drives and bearings may be considered except in vacuum or hi-temperature service in lieu of bushings with small clearances.
Ball screw drives	X	X	O	X	O	X	X	O	X	NA	X	X	X	X	O	NA	X	U	
Flexure pivots	X	X	X	X	X	X	X	X	X	U	X	X	X	X	X	NA	X	U	
Gears	X	X	O	X	O	O	X	O	X	NA	X	X	O	X	O	NA	O	U	
Fittings and fasteners:																			
AN and MS connections	X	X	X	X	O	O	X	X	X	U	X	NA	NA	NA	O	O	O	O	(1) (Use welded construction and bolts and screws where possible and where space maintenance is not required.
Welded	X	X	X	X	X	X	X	X	X	X	X	NA	NA	NA	X	X	X	O	
Bolts and screws	X	X	X	X	X	X	X	X	X	X	X	NA	NA	NA	X	NA	NA	O	(2) Design concepts required for space maintenance.

X—Use
O—Do not use
NA—Not applicable
U—Unknown

TABLE C-2.—*Functional and Environmental Valve Parameters*

	Functional Parameters													Space Flight Environmental Parameters												
	Media													Vacuum			Radiation									
	Liquid	Gas	Sterilization	Operating Temperature	Operating Pressure	Weight	Power Requirements	Leakage	Operating Life	Contamination	Space Maintenance	Response	Vibration and Shock	Atmospheric to 10^{-9} mm Hg	10^{-9} mm Hg and higher	Cosmic	Ultraviolet	Geomagnetic (Van Allen belt)	Auroral soft	Solar Flares	Induced (Bremsstrahlung)	Zero g	Time	Temperature	Meteoroids	Products of Combustion (Turn Around)
Type of valve:																										
Disconnect	2	2	2	3	3	3	NA	1	3	1	1	NA	3	2	2	3	3	2	3	U	2	3	2	3	2/1M	1
Shutoff	3	3	2	2	3	3	2	1	2	1	1	2	3	1	1	3	3	2	3	U	2	3	2	3	2/1M	2
Flow Metering	3	3	2	3	3	3	3	NA	3	3	1	3	3	3	3	3	3	3	3	3	3	3	3	3	2/1M	3
Vent	2	2	2	3	3	3	NA	1	3	3	1	2	3	2	2	3	3	2	3	U	2	1	2	3	2/1M	2
Regulator	NA	3	2	3	3	3	NA	2	2	1	1	2	3	3	3	3	3	2	3	U	2	3	2	3	2/1M	3
Closure:																										
Ball	1	1	3	1	3	3	2	3	3	3	NA	2	3	2	2	3	3	1	3	2	2	NA	2	3	NA	3
Poppet	3	3	3	3	3	3	3	3	3	2	NA	3	2	3	3	3	3	3	3	2, 3	3	3	3	3	NA	3
Butterfly	3	3	3	3	2	3	2	3	3	3	NA	2	3	3	3	3	3	3	3	2, 3	3	3	3	3	NA	3
Burst Diaphragm	3	3	3	3	3	3	3	3	1	3	NA	3	3	3	3	3	3	3	3	3	3	3	3	3	NA	3
Seals:																										
Metal	3	3	2	2	3	3	NA	1,2	3	1	NA	NA	3	1	1	3	3	3	3	3	3	NA	3	3	NA	1
Polymeric	1	1	2	1	3	3	NA	3	3	3	NA	NA	3	2	2	3	3	1	3	2	2	3	2	2	NA	3

Actuator:																										
Solenoid	NA	3	3	2	NA	2	2	NA	3	3	1	3	3	2	2	3	3	2	3	2	2	3	3	2	2/1M	3
Pneumatic	NA	3	3	3	3	3	3	3	3	3	1	3	3	3	3	3	3	2	3	3	3	3	3	3	2/1M	3
Hydraulic	3	NA	3	3	3	3	3	3	3	3	1	3	3	3	3	3	3	3	3	3	3	3	3	3	2/1M	3
Elect. Motor	NA	NA	3	NA	NA	2	2	NA	3	NA	1	2	3	2	2	3	3	2	3	2	2	3	3	2	2/1M	3
Squib	NA	NA	U	NA	NA	3	3	NA	1	NA	1	3	3	3	3	3	3	U	3	U	U	3	U	U	2/1M	3
Lubricant:																										
Dry	1, 3	2	3	2, 3	NA	3	NA	NA	3	3	1	NA	3	2	2	3	U	U	U	U	U	3	3	3	NA	3
Liquid	1, 3	1, 3	3	1	NA	3	NA	1	3	3	1	NA	3	2	2	3	3	2	3	U	2	2	2	3	NA	3
Fittings and Fasteners:																										
AN flare	3	3	3	3	3	2	NA	2	3	3	1	NA	3	3	3	3	3	3	3	3	3	3	3	3	3	3
MS Flareless	3	3	3	3	3	2	NA	2	3	3	1	NA	3	3	3	3	3	3	3	3	3	3	3	3	3	3
Welded	3	3	3	3	3	3	NA	3	3	3	1	NA	3	3	3	3	3	3	3	3	3	3	3	3	3	3
Bolts and Screws	3	3	3	NA	NA	3	NA	NA	3	NA	1	NA	3	3	3	3	3	3	3	3	3	3	3	3	3	3
Filters	3	3	3	1, 3	3	3	NA	NA	3	2	1	NA	2	3	3	3	3	3	3	3	3	3	3	3	NA	3

Rating characteristics:

1—Poor

2—Fair

3—Good

U—Unavailable information

NA—Not applicable

Example:

Unmanned (digit only)

2/1M

Manned (subscript)

$\boxed{3}$ = Same value for manned and unmanned (digit only)

TABLE C-3.—*Valve Analysis Chart—Part A*

Applicability		Performance With Propellants																										
		Propellant Gases or Pressurants to 5000 psia							Combustion Gases to 1500° F to 5000 psia	Liquid Propellants to 1000 psia													Gels—metallized			Nonmetal-lized gels		
		Hydrogen	Helium	Nitrogen	Oxygen	Argon	CO_2	Propellant Boiloff gas		Nitrogen Tetroxide	Hydrazine	UDMH	Monomethyl Hydrazine-MMH	Aerozine 50	Pentaborane—9	Chlorine Trifluoride	Perchloryl Fluoride	Oxygen Difluoride	LOX	Liquid Hydrogen	Liquid Fluorine	Hybaline AS	N_2H_4	UDMH	MMH	N_2O_4	ClF_3	MAF—mixed amine fuels
	Valve type:																											
FM	Flow metering	3	3	3	3	3	3	NA	2-3	3	3	3	3	3	3	3	3[a]	3[a]	3[a]	3[a]	3[a]	3	U	U	U	U	U	U
S	Shutoff	3	2[b]	3	3	3	3	U	2-3	3	3	3	3	3	2[e]	1-2[b]	2[b]	1[b]	2[b]	1-2[b]	1[b]	3	1	1	1	2[c]	2[c]	2[c]
V	Vent or relief	3	2[b]	3	3	3	3	2-3	3	3	3	3	3	3	3	2	2	2	2	2	2	3	NA	NA	NA	NA	NA	NA
CR	Cold gas regulator	3	2[b]	3	3	3	3	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA
HR	Hot gas regulator	NA	NA	3	U	NA	3	NA	2	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA
LF	Liquid fill or disconnect	NA	NA	NA	NA	NA	NA	2-3	NA	2[d]	2[d]	2[d]	2[d]	2[d]	2[d]	2[d]	2[d]	2[d]	1[e] 2[d]	1[e] 2[d]	1[e] 2[d]	3	1	1	1	1	1	1
PF	Pneumatic fill or disconnect	2[d]	2[d]	2[d]	2[d]	2[d]	2[d]	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA
	Closure:																											
S	Ball	3	3	3	3	3	3	3	1	2	3	3	3	3	3	1-2	1	1-2	2	2	1	3	3	3	3	3	3	3
S,V,CR,HR,LF,PF	Poppet	3	3	3	3	3	3	3	2	3	3	3	3	3	3	2-3	3	2	3	3	2-3	3	2	2	2	2	2	2
FM,S	Butterfly	3	3	3	3	3	3	3	2	2	2	2	2	3	3	1-2	1-2	1-2	3	3	1-2	3	1	1	1	1	1	1
S	Burst diaphragm	3	3	3	3	3	3	NA	NA	3	3	3	3	3	3	U	U	U	3	3	U	3	3	3	3	3	3	3
	Polymeric seals:																											
S,V,CR,HR,LF,PF	Valve closure	3	1-2[b]	3	3	3	3	1-2	1	2	2	2	2	2	2	1	1	1	1-2	1-2[b]	1-2[b]	U	1	1	1	2	1	2
All	Static seals	3	1-3	3	3	3	3	1-2	1	2	3	3	3	3	2	1-2	1-2	1-2	2	2[b]	1-2[b]	3	2	3	3	2	3	3
All	Dynamic seals	3	1-2	3	3	3	3	1-2	1	1	3	3	2	3	3	1	1	1	1-2	1-2[b]	1[b]	U	1	1	1	2	2	2
	Metal seals:																											
S,V,CR,HR,LF,PF	Valve closure	1-2[b]	1[b]	1-2[b]	1-2[b]	1-2[b]	1-2[b]	1-2[b]	1-2[b]	3	3	3	3	3	3	2-3	U	U	1[b]	1[b]	1[b]	3	2	2	2	2[c]	2[c]	2[c]
All	Static seals	3	3	3	3	3	3	3	3	3	3	3	3	3	3	2-3	2	2	3	3	2	3	3	3	3	3	3	3
All	Dynamic seals	3	1[b]	3	3	3	3	3	3	3	3	3	3	3	3	2-3	2	2	1[b]	1[b]	1[b]	3	U	U	U	3	3	3
	Actuators:																											
S	Solenoid	3	3	3	3	3	3	1-3	1	3	3	3	3	3	3	3	U	U	U	U	1	3	3	3	3	3	3	3
FM,S	Pneumatic	3	3	3	3	3	3	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA
FM,S	Hydraulic	NA	NA	NA	NA	NA	NA	NA	NA	3	2	2	2	2	3	3	U	U	U	U	U	U	U	U	U	U	U	U

FM,S	Elect. motor	1	3	3	3	3	3	3	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA
S	Squib	3	3	3	3	3	3	3	1	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA
	Drive mechanisms:																											
All	Linear push-pull	U	2f	2f	2f	2f	2f	1-2f	1	U	1f	1f	1f	1f	1f	U	U	U	2f	2f	U	U	1	1	1	3f	U	1f
All	Ball screw drives	2f	2f	2f	2f	2f	2f	1-2f	1	3f	1f	1f	1f	1f	1f	U	U	U	2f	2f	U	U	1	1	1	3f	U	1f
All	Linear ball drives	2f	2f	2f	2f	2f	2f	1-2f	1	3f	1f	1f	1f	1f	1f	U	U	U	2f	2f	U	U	1	1	1	3f	U	1f
All	Geared	U	2f	2f	2f	2f	2f	1-2f	1	3f	1f	1f	1f	1f	1f	U	U	U	U	U	U	U	U	U	U	U	U	U
All	Piston	U	U	2f	2f	2f	2f	1-2f	2	3f	1f	1f	1f	1f	1f	U	U	U	2f	U	U	U	U	U	U	1f	1f	1f
	Bearings and bushings																											
All	Ball-rotary	2f	2f	2f	2f	2f	2f	1-2f	1	3f	1f	1f	1f	1f	1f	2-3	U	U	2f	2f	U	2f	1	1	1	3f	1f	1f
All	Polymeric	3	3	3	3	3	3	1-3	1	1	U	U	U	U	U	1	1	1	2-3	2-3	1	U	1	1	1	U	U	U
All	Metal	2f	2f	2f	2f	2	2f	1-2f	1	3f	1f	1f	1f	1f	1f	2-3	U	U	2f	2f	U	2f	1	1	1	1f	1f	1f
All	Flexure pivots	3	3	3	3	3	3	3	3	3	3	3	3	3	3	U	U	U	3	3	U	3	3	3	3	3	3	3
	Hermetic seals:																											
All	Bellows	2	3	3	3	3	3	3	3	3	3	3	3	3	3	2	3	3	3	3	U	3	3	3	3	3	3	3
All	Diaphragms	2	3	3	3	3	3	3	3	3	3	3	3	3	3	U	3	3	3	3	U	3	3	3	3	3	3	3
	Lubes:																											
All	Dry film	3	3	3	3	3	3	1-2	1	2	2	2	2	2-3	2	1	1	1	1	1	1	U	U	U	U	U	U	U
All	Liquid	3	3	3	3	3	3	1-2	1	1	2	2	1	2	1-2	1	1	1	1	1	1	U	1	1	1	1	1	1
	Springs:																											
All	Coil	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3
All	Belleville	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3
All	Liquid	1-2	3	3	2	3	3	1-2	1	2g	2g	3g	3g	3g	3g	2g	U	U	U	U	U	U	U	U	U	U	U	U
	Fittings and fasteners:																											
FM,S,CR,HR	AN flare and MS flareless	3	3	3	3	3	3	3	3	3	3	3	3	3	3	1h	2	2	3	3	1h	3	3	3	3	3	3	3
All	Welded	3	3	3	3	3	3	3	3	3	3	3	3	3	3	2	3	3	3	3	2	3	3	3	3	3	3	3
All	Bolts and screws	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3
All	Filters	3	3	3	3	3	3	3	3	3	3	3	3	3	3	2	2	3	3	3	1-2	3	NA	NA	NA	U	U	U

[a] Noncavitating flow control.

[b] Rating based on leakage control.

[c] Rating for shutoff valve exposed to space vacuum.

[d] Rating for service in vacuum environment.

[e] Freezing of condensed moisture at interface of disconnect.

[f] Rating based on propellant lubrication data at low loads, short duration.

[g] Propellant used as compressible fluid.

[h] Rating based on poor reliability associated with passivation.

[i] Contamination generator.

Rating characteristics:

1—Poor
2—Fair
3—Good
U—Unavailable information
NA—Not applicable

Unmanned (digit only)

2 / 1M

Manned

Valve designation:

FM—Flow metering
S—Shutoff
V—Vent or relief
CR—Cold gas regulator
HR—Hot gas regulator
LF—Liquid fill or disconnect
PF—Pneumatic fill or disconnect

TABLE C-3.—*Valve Analysis Chart—Part B*

Applicability		Functional parameters									Space environmental parameters							
													Radiation					
		Temperature high to 2000° F	Temperature low	Sterilization	Power requirements	Operating life 1000 cycles	Contamination	Response	Vibration and shock	Leakage	Space maintenance	Space vacuum	Van Allen belt	Solar flare	Induced bremsstrahlung	Zero g	Meteoroids	Products of combustion (turn around)
	Valve type:																	
FM	Flow metering	U	3	2	3	3	3	3	3	NA	U	3	3	3	2	3	2/1M	2
S	Shutoff	1	3	2	2	2	1	2	3	1	U	1	2	2	2	3	2/1M	2
V	Vent or relief	2	3	2	3	3	3	2	3	1	U	2	2	2	2	1	2/1M	2
CR	Cold gas regulator	NA	2–3	2	3	2	1–2	2	2	2	U	3	3	3	2	3	2/1M	3
HR	Hot gas regulator	2	NA	2	3	2	1–2	2	2	2	U	3	3	3	2	3	2/1M	3
LF	Liquid fill or disconnect	NA	2	2	NA	3	2	NA	3	1	U	2	2	2	2	3	2/1M	1
PF	Pneumatic fill or disconnect	NA	3	2	NA	3	2	NA	3	1	U	2	2	2	2	3	2/1M	1
	Closure:																	
S	Ball	1	2	3	2	3	3	2	3	3	NA	1	3	3	3	3	NA	3
S,V,CR,HR,LF,PF	Poppet	2	3	3	3	3	2	3	2	3	NA	2	3	3	3	3	NA	3
FM,S	Butterfly	U	3	3	2	3	3	2	3	3	NA	2	3	3	3	3	NA	3
S	Burst diaphragm	3	3	3	3	1	1–2	3	3	3	NA	3	3	3	3	3	NA	3
	Polymeric seals:																	
S,V,CR,HR,LF,PF	Valve closure	1	2	2	NA	3	3	NA	3	3	U	2	1–2	1–2	1–2	3	NA	2
All	Static seals	1	2	2	NA	2	3	NA	3	3	U	2	1–2	2	1–2	3	3	3
All	Dynamic seals	1	2	2	1–2	3	3	2	3	3	U	2	1–2	2	1–2	3	U	2
	Metal seals:																	
S,V,CR,HR,LF,PF	Valve closure	2	3	3	NA	3	1	NA	3	1–2	U	1	3	3	3	3	NA	1
All	Static seals	2	3	3	NA	1	3	NA	3	3	U	2	3	3	3	3	3	1
All	Dynamic seals	2	3	3	2	2	1	2	3	1–2	U	1	3	3	3	3	U	1

	Actuator:																	
S	Solenoid	1	3	3	2	3	3	3	3	NA	U	1–2	2	2	3	3	2/1M	3
FM,S	Pneumatic	U	NA	3	3	3	3	3	3	3	U	3	2	3	3	3	2/1M	3
FM,S	Hydraulic	1	1	3	3	3	3	3	3	3	U	3	3	3	3	3	2/1M	3
FM,S	Elect. motor	1	1	3	2	3	NA	2	3	NA	U	1–2	2	2	2	3	2/1M	3
S	Squib	1	3	U	3	1	3	3	3	1	1	U	U	U	U	3	U	3
	Drive mechanisms:																	
All	Linear push-pull	1	U	3	2	3	2	3	3	NA	1	1	3	3	3	3	2/1M	2
All	Ball screw drives	1	2	3	3	3	3	3	3	NA	1	1	3	3	3	3	2/1M	2
All	Linear ball drives	1	2	3	3	3	3	3	3	NA	1	1	3	3	3	3	2/1M	2
All	Geared	U	2	3	2	3	3	2	3	NA	1	1	3	3	3	3	2/1M	2
All	Piston	3	2	3	2–3	3	2	3	3	2	1	1–2	3	3	3	3	2/1M	2
	Bushings and bearings:																	
All	Ball-rotary	1	2	3	3	3	2	2	3	NA	U	1–2	3	3	3	3	2/1M	1
All	Polymeric	1	2	2	2	2	3	1	3	NA	U	2	1–2	2	2	3	2/1M	2
All	Metal	1	2	2	2–3	2	1	1–2	3	NA	U	1	3	3	3	3	2/1M	1
All	Flexure pivots	U	U	3	3	3	3	3	3	NA	U	3	3	3	3	3	2/1M	3
	Hermetic seals:																	
All	Bellows	1	3	3	1	3	NA	3	3	3	1	3	3	3	3	3	2/1M	3
All	Diaphragm metal	1	3	3	3	3	NA	3	3	3	1	3	3	3	3	3	2/1M	3
	Lubes:																	
All	Dry film	1	1	3	NA	3	2	NA	3	3	1	2	U	U	U	3	NA	1–2
All	Liquid	1	1	3	NA	3	3	NA	3	1	1	2	2	2	U	1–2	NA	2–3
	Springs:																	
All	Coil	1–2	3	3	NA	3	3	2–3	3	NA	U	3	3	3	3	3	2/1M	3
All	Belleville	1–2	3	3	NA	3	3	2	3	NA	U	3	3	3	3	3	2/1M	3
All	Liquid	1	1	3	NA	3	3	3	3	2	U	2	U	U	U	3	2/1M	3
	Fittings and fasteners:																	
FM,S,CR,HR	AN flare and MS flareless	1	3	3	NA	1	3	NA	2	2	U	2	3	3	3	3	3	3
All	Welded	3	3	3	NA	NA	NA	NA	3	3	1	3	3	3	3	3	3	3
All	Bolts and screws	3	3	3	NA	NA	NA	NA	3	NA	U	3	3	3	3	3	3	3
All	Filters	U	3	3	NA	3	1[1]	NA	3	NA	U	3	3	3	3	3	NA	3

TABLE C-4.—*Valve Component Rating Analysis Chart*

Applicability		Earth Moon: Atmospheric pressure ≈ 10^{-14} mm Hg	Earth Moon: Temperature —243° F to 260° F	Earth Moon: Mission time ≈ 3–6 days	Earth Moon: Lunar dust	Earth Moon: Lunar maintenance	Venus: Atmospheric composition CO_2 and N_2	Venus: Atmospheric pressure ≈ 20 Earth atm	Venus: Temperature ≈ 800° F	Venus: Mission time ≈ 4 months	Venus: Venus dust	Venus: Venus maintenance	Mars: Atmospheric composition 95% N_2 5% CO_2	Mars: Atmospheric pressure << 1 atm.[a]	Mars: Temperature —180° F to 90° F	Mars: Mission time ≈ 1 year	Mars: Radiation belt (Unknown)[b]	Mars: Martian dust	Mars: Martian maintenance	Jupiter: Atmospheric composition CH_4, NH_3	Jupiter: Atmospheric pressure >> 1 atm.[a]	Jupiter: Temperature[a] ≈ —200 ° F	Jupiter: Mission time ≈ 3 years	Jupiter: Radiation belt (Unknown)[c]	Jupiter: Jovian dust	Jupiter: Jovian maintenance
	Valve type:																									
FM	Flow metering	3	2	3	3	U	3	3	U	3	3	U	3	3	3	2	U	3	U	3	3	3	2	U	3	U
S	Shutoff	1	2	3	2	U	3	3	1–2	3	2	U	3	3	3	2	U	2	U	3	3	3	2	U	2	U
V	Vent or relief	2	2	3	2	U	3	3	2	3	2	U	3	3	3	2	U	2	U	3	3	3	2	U	2	U
CR	Cold gas regulator	3	2	3	3	U	3	3	NA	3	3	U	3	3	3	2	U	3	U	3	3	2–3	2	U	3	U
HR	Hot gas regulator	3	2	3	3	U	3	3	3	3	3	U	3	3	3	2	U	3	U	3	3	2–3	2	U	3	U
LF	Liquid fill or disconnect	2	2	3	1	U	3	3	U	3	1	U	3	3	3	2	U	1	U	3	3	2	2	U	1	U
PF	Pneumatic fill or disconnect	2	3	3	1	U	3	3	U	3	1	U	3	3	3	2	U	1	U	3	3	3	2	U	1	U
	Closure:																									
S	Ball (polymeric seal)	1	1–2	3	2–3	NA	3	3	1–2	3	2–3	NA	3	3	2	2–3	U	2–3	NA	3	3	2	U	U	2–3	NA
S,V,CR,HR,LF,PF	Poppet	2	3	3	2–3	NA	3	3	1–2	3	2–3	NA	3	3	3	3	3	2–3	NA	3	3	3	3	3	2–3	NA
FM,S	Butterfly	2	3	3	2–3	NA	3	3	3	3	2–3	NA	3	3	3	3	3	2–3	NA	3	3	3	3	3	2–3	NA
S	Burst diaphragm	3	3	3	3	NA	3	3	3	3	3	NA	3	3	3	3	3	3	NA	3	3	3	3	U	3	NA
	Polymeric seals:																									
S,V,CR,HR,LF,PF	Valve closure	2	1–2	3	2	U	3	3	1	3	2	U	3	3	2	2–3	U	2	U	3	3	2	2	1–2	2	U
All	Static seals	2	1–2	3	3	U	3	3	1	3	3	U	3	3	2	3	U	3	U	3	3	2	3	1–2	3	U
All	Dynamic seals	2	1–2	3	2	U	3	3	1	3	2	U	3	3	2	2–3	U	2	U	3	3	2	2	1–2	2	U
	Metal seals:																									
S,V,CR,HR,LF,PF	Valve closure	1	2	3	1	NA	3	3	2	3.	1	NA	3	3	3	U	3	1	NA	3	3	3	U	3	1	NA
All	Static seals	2	2–3	3	3	U	3	3	2	3	3	U	3	3	3	3	3	3	U	3	3	3	3	3	3	U
All	Dynamic seals	1	2	3	1	U	3	3	1	3	1	U	3	3	3	U	3	1	U	3	3	3	U	3	1	U
	Actuators:																									
S	Solenoid	1–2	3	3	3	U	3	3	2	3	3	U	3	3	3	2–3	U	3	U	3	3	3	2	1–2	3	U
FM,S	Pneumatic	3	3	3	3	U	3	3	2	3	3	U	3	3	3	3	3	3	U	3	3	3	3	2	3	U
FM,S	Hydraulic	3	1–2	3	3	U	3	3	1	3	3	U	3	3	1–2	3	3	3	U	3	3	1–2	3	2	3	U

FM,S	Elect. motor	1–2	1	U	3	U	U	U	1	U	3	U	3	3	3	U	U	3	U	U	U	1–2	U	1–2	3	U
S	Squib	U	2–3	3	U	1	3	U	1	3	U	1	3	3	3	U	U	U	1	3	U	3	U	U	U	1
	Drive mechanisms:																									
All	Screw drives	1	2	3	1	U	2	3	2	3	1	U	3	3	3	U	3	1	U	3	3	2	U	3	1	U
All	Ball screw drives	1	2	3	1	U	2	3	2	3	1	U	3	3	2–3	U	3	1	U	3	3	2	U	3	1	U
All	Linear ball drives	1	2	3	1	U	2	3	2	3	1	U	3	3	2–3	U	3	1	U	3	3	2	U	3	1	U
All	Geared	1	2	3	2	U	2	3	2	3	2	U	3	3	2–3	U	3	2	U	3	3	2	U	3	2	U
All	Piston	1–2	2–3	3	2	U	3	3	2	3	2	U	3	3	3	3	3	2	U	3	3	3	3	3	2	U
	Bearing and bushings:																									
All	Ball-rotary	1–2	2	3	1	U	2	3	2	3	1	U	3	3	2–3	U	3	1	U	3	3	3	U	3	1	U
All	Polymeric	2	2	3	2	U	3	3	1	3	2	U	3	3	2–3	U	U	2	U	3	3	2	U	1–2	2	U
All	Metal	1	2	3	1	U	2	3	1	3	1	U	3	3	2–3	U	3	1	U	3	3	2	U	3	1	U
All	Flexure pivots	3	U	3	3	U	3	3	U	3	3	U	3	3	U	U	3	3	U	3	3	U	U	U	1	U
	Hermetic seals:																									
All	Bellows	3	3	3	3	1	3	3	2	3	3	1	3	3	3	U	3	3	1	3	3	3	U	U	3	1
All	Diaphragm metals	3	3	3	3	1	3	3	2	3	3	1	3	3	3	U	3	3	1	3	3	3	U	U	3	1
	Lubes:																									
All	Dry film	2	1	3	1–2	1	3	3	2	3	1–2	1	3	3	1	2–3	U	1–2	1	3	3	1	2	U	1–2	1
All	Liquid	2	1	3	2–3	1	3	3	1	3	2–3	1	3	3	1	U	U	2–3	1	3	3	1	U	U	2–3	1
	Springs:																									
All	Coil	3	3	3	3	U	3	3	2	3	3	U	3	3	3	3	3	3	U	3	3	3	3	3	3	U
All	Belleville	3	3	3	3	U	3	3	2	3	3	U	3	3	3	3	3	3	U	3	3	3	3	3	3	U
All	Liquid	2	1–2	3	3	U	3	3	1	3	3	U	3	3	2	U	U	3	U	3	3	1–2	U	U	3	U
	Fittings and fasteners:																									
FM,S,CR,HR	AN flare and MS flareless	3	3	3	3	U	3	3	1	3	3	U	3	3	3	3	3	3	U	3	3	3	3	3	3	U
All	Welded	3	3	3	3	1	3	3	3	3	3	1	3	3	3	3	3	3	1	3	3	3	3	3	3	1
All	Bolts and screws	3	3	3	3	U	3	3	3	3	3	U	3	3	3	3	3	3	U	3	3	3	3	3	3	U
All	Filters	3	3	3	NA	U	3	3	3	3	NA	U	3	3	3	3	3	NA	U	3	3	3	3	3	NA	U

[a] Estimated.
[b] Assumed to be small.
[c] Assumed much greater than Van Allen level.

Rating characteristics:

1—Poor
2—Fair
3—Good
U—Unavailable information
NA—Not applicable

Valve designation:

FM—Flow metering
S—Shutoff
V—Vent or relief
CR—Cold gas regulator
HR—Hot gas regulator
LF—Liquid fill or disconnect
PF—Pneumatic fill or disconnect

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